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Parametric identification of the mathematical model of the functioning of tribosystems in the conditions of boundary lubrication

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Abstract

The parametric identification of the tribosystem as an object of modeling the functioning of tribosystems in the conditions of boundary lubrication is performed in the work. Using the analysis of the dimensions of significant factors, expressions are obtained to calculate the gain and time constants.

It is established that the coefficient K_1 takes into account the degree of influence of the load, sliding speed, tribological characteristics of the lubricating medium on the quality factor of the tribosystem. It is shown that the increase in the coefficient K_1 will have a positive effect on the processes inherent in tribosystems during operation. Coefficient K_2 – characterizes the magnitude of the change in volumetric wear rate and friction coefficient when changing the magnitude of the load, sliding speed, quality factor of the tribosystem. Coefficient K_3 – characterizes the ability of the tribosystem to self-organize when changing the values of the input parameters by rearranging the surface layers of materials from which the triboelements are made during secondary running-in. It is shown that the value of the coefficient is large K_3 will contribute to the rapid change in the roughness of the friction surfaces, the restructuring of the structure of the surface layers, the appearance of oxidizing films on the friction surfaces (secondary structures).

It is proved that the time constant T_1 – this is the time required to change the roughness of the friction surfaces and rearrange the structure of the materials of the surface layers when changing external conditions. Time constant T_2 characterizes the time during which there is a stabilization of the temperature gradient by volume of triboelements, taking into account the thermal conductivity of materials when changing external conditions. Time constant T_3 characterizes the time during which the tribosystem returns to a steady state of operation after the cessation of the outrageous force, or the time to stabilize the parameters in the new mode of operation. It is proved that the value T_3 will be optimal for the process of self-organization. It is shown that one of the factors that can control the value T_3, this is the sliding speed υ_{sl}.

Keywords: tribosystem; mathematical model; differential equations; parametric identification; coefficient of gain; time constant; boundary lubrication; quality factor of the tribosystem; dissipation speed

Introduction

Analysis of the current state of mathematical models of friction and wear, as well as methods for calculating wear and predicting the resource, shows the increasing attention of researchers to this problem. This is due to high material costs in the design, testing and refinement of new models of equipment before they are put into production. One of the ways to reduce material costs is to replace laboratory and bench tests with mathematical modeling and resource prediction at the design stage.

For the development of mathematical models, both analytical and numerical methods are used. In this case, the task is posed that mathematical models should be multifactorial and take into account the processes of deformation and destruction of the surface layers of tribosystems, the formation of wear-resistant structures on friction surfaces and the exchange of friction surfaces with matter and energy. Such multi-level and multifactorial tasks are solved by various methods and different approaches, while they have one goal - resource forecasting.

This work is a continuation of work [1], where a third-order differential equation is obtained for modeling the boundaries of stable operation of tribosystems under boundary lubrication conditions. To use this approach, it is necessary to obtain expressions for calculating all the coefficients and time constants that are included in the left and right sides of the equation. The presence of such coefficients will make it possible to simulate the processes of friction and wear in tribosystems.

Literature review

The authors of the work [2] studies on modeling the processes of friction and wear at the mesoscale level. As a methodological approach, the authors use the finite element method, where surface roughness is represented by the combined law of friction. In the developed model, the relations of wear of the friction surfaces and the corresponding mechanisms of energy dissipation on the spots of actual contact are considered. According to the authors, such a methodological approach makes it possible to simulate the wear process with sufficient accuracy. A similar approach is used by the authors of the work [3]. The concepts developed in this article are based on statistical analysis, which is based on the mechanisms of energy dissipation during friction of contacting sliding surfaces.

In work [4] it is shown that the dynamics of the tribosystem is well reproduced by simplified models obtained using the Markov process, even in the presence of several minima of the investigated functions. After evaluating the parameters of the tribosystem by numerical simulation, the authors calculate the average wear rate and friction losses when external forces and temperature change.

In work [5] the physical mechanisms of formation and transformation of corpuscular-vortex perturbations in the contact of the tribosystem, which are based on the quantum-mechanical exchange mechanism of interaction, are considered. The presence of a contact gap determines the generation of pairs of quasiparticles-perturbations, stabilized by wavelength and frequency. It is established in the work that the internal instability and collapse processes in such a system of perturbations lead to defect formation in the material of the tribosystem and underlie the emergency modes of friction.

In the works [6, 7] performed analysis of the strength and durability of the surface layer material by friction. The authors propose to take into account the presence of two areas of accumulation of damage and the type of mechanism of destruction: the area of multicycle fatigue and a layer of debris. Methods for estimating the parameters of the durability model for the region of multicycle fatigue are proposed. The connection between the stress-strain state and the fatigue strength characteristics of the material with the characteristics of the material fracture model is obtained. The analysis of the received relations showed that any physical action on a surface, leads to decrease in structural inhomogeneity and prevents development of cracks, promotes increase of wear resistance.

In work [8] the analysis of various methods for calculating wear and predicting the resource is given and it is concluded that analytical methods do not allow taking into account the dynamics of changing the operating modes of the contact, and numerical methods seem to be promising. The author of the work proposed to describe wear by an array of probability vectors of wear values of discrete points of the surface, which are modeled by non-stationary random functions of the Markov type, and wear is estimated by the mathematical expectation of the probability of finding surface elements in a certain state.

In work [9] theoretical studies on the substantiation of the methodology for modeling stationary processes of friction and wear in tribosystems under conditions of boundary lubrication are presented. The authors have developed a technique for modeling the characteristics of the actual contact patch and a mathematical model of the rate of dissipation in the tribosystem, which allow modeling the rate of volumetric wear and the coefficient of friction in stationary modes.

The analysis of the above works shows the versatility in the approaches to the construction of mathematical models of the processes of friction and wear in tribosystems. In our opinion, the most promising are numerical methods based on differential equations, which make it possible to simulate the dynamics of the transient process. Obtaining such models is associated with the stages of structural and parametric identification. Structural identification of the tribosystem is carried out in the work [1], where the third-order differential equation is obtained. The obtained equation can be used provided that all the parameters that are included in the left and right sides of the equation are determined. Obtaining such expressions is called parametric identification of a mathematical model.

Purpose

The purpose of this work is to perform parametric identification of the tribosystem and obtain expressions for the gains and time constants, which are included in the differential equation for modeling the processes of friction and wear in tribosystems.

Methods

To substantiate the methodological approach in research, we use the equation of the dynamics of the functioning of the tribosystem, which is given in [1]. The third-order differential equation is written in operator form:

$$\begin{aligned} &\left(T_1T_2T_3\right)p^3+\left(T_1T_2+T_1T_3+T_2T_3\right)p^2+\left(T_1+T_2+T_3+K_2K_3T_1\right)p+ \\ &+K_2K_3+1=\left(K_1K_2T_3\right)p+K_1K_2. \end{aligned} \tag{1}$$

p – a differentiation operator that is equivalent to a record d/dt;

T_1, T_2, T_3 – time constants, dimension s;

K_1, K_2, K_3 – coefficients of gain, dimensionless quantities.

The right part of the differential equation (1) contains the first derivative of the input signal, which is represented as the product of the coefficients $K_1\,K_2$ and time constant T_3. The dynamics of the tribosystem is influenced not only by the magnitude of the values K_1, K_2, T_3, as well as the rate of change over time (the first derivative).

The left side of the equation is the reaction of the tribosystem to the input signal. Time constants of the tribosystem T_1, T_2, T_3 have the dimension of time and characterize the inertia of the processes occurring in the tribosystem, during running-in, or during changes in operating modes.

The purpose of parametric identification is to determine the expressions for the calculation of the above coefficients and time constants, so that when substituting them in equation (1), the right and left parts differ the least.

When solving the problems of friction and wear, a methodical approach to the theory of similarity and modeling is often used, where dimensional analysis methods are used to obtain dimensionless criteria. Analyzing the dimensional factors that affect the process, but do not depend on each other, you can get dimensionless criteria (coefficients) that adequately describe the process in similar (different in size and design) physical objects.

The procedure of parametric identification or finding expressions for calculation K_1-K_3, T_1-T_3, which characterize the dynamics of the functioning of tribosystems, there is an experimental material that allows you to choose the most significant factors.

Such factors include.

1. The diameter of the actual contact spot d_{acs}, m, the number of contact spots on the friction surface n_{acs}, pc, and stress on the actual contact spots σ_{acs}, Pa. Depends on the load on the tribosystem N, N, modulus of elasticity and roughness of contact materials of triboelements. Calculated according to the formulas given in the work [9].

2. Deformation rate in movable $\acute{\varepsilon}_{mov}$ and fixed $\acute{\varepsilon}_{fix}$, triboelements, s^{-1}. Depends on the load N, N, sliding speed υ_{sl}, m/s, modulus of elasticity and Poisson's ratio of contact materials of triboelements. Calculated according to the formulas given in the work [9].

3. Tribosystem shape factor K_{ϕ}, m^{-1}, takes into account the areas of friction and the volumes located under the areas of friction in the movable and fixed triboelements. Calculated by the formula [10].

4. Coefficient of thermal conductivity a, m^2/s, takes into account the thermal conductivity of movable materials a_{mov} and fixed a_{fix} triboelements. Reference value.

5. Tribological properties of the lubricating medium E_u, J/m^3, are determined on a four-ball friction machine and take into account the anti-wear and anti-emergency properties of lubricants, calculated by the formula [11].

6. Rheological properties of the structure of movable materials δ_{mov} and fixed δ_{fix} triboelements, dimension dB/m. Takes into account the internal friction of the material structure. The values of internal friction for steels, cast irons, bronzes are presented in the paper [12].

7. Smaller area of friction of one of the triboelements, $F_{\min}$, m^2.

8. Volumes of material which are located under the areas of friction at the movable V_{mov}, m^3 and fixed V_{fix}, m^3 triboelements.

9. The volume of movable material V_{dmov} and fixed V_{dfix} triboelements, m^3, involved in the deformation during friction, is calculated by formulas:

$$V_{dmov} = F_{\max} \cdot h_{dmov}, \text{ m}^3, \tag{2}$$

$$V_{dfix} = F_{\min} \cdot h_{dfix}, \text{ m}^3, \tag{3}$$

where $F_{\max}$ – large area of friction of one of the triboelements;

h_{dmov} and h_{dfix} – depth of propagation of deformation in movable and fixed triboelements, m.

In work [13] the proposed physical quantity is the quality factor of the tribosystem Q, and the dependences of the change in the quality factor on the initial value are given Q_0 to the maximum value $Q_{\max}$, which increases during running-in.

Results

Coefficient of gain K_1, included in the differential equations and their solutions, in the theory of identification of dynamic objects is called the coefficient that takes into account the degree of influence of the input signal (load, sliding speed, tribological characteristics of the lubricating medium), the magnitude of the output signal (quality of the tribosystem). Based on this physical concept and using the methods of dimensionality of similarity theory and modeling, we obtain the expression:

$$K_1 = \frac{Q_{\max}}{Q_0}, \tag{4}$$

where Q_0 and $Q_{\max}$ – the initial value of the quality factor of the tribosystem and the value of the quality factor formed during the running-in of the tribosystem. Determined by the formulas given in the work [13].

As follows from expression (4), the ratio of the maximum value of the quality factor, which is characteristic of the tribosystem after completion of running-in, to the quality factor of the tribosystem Q_0 before starting, evaluates the possibility of the tribosystem to change the structure of the surface layers of the materials of the triboelements under load. The increase in the quality factor of the tribosystem is based on the concept of compatibility of materials in the tribosystem. Increasing the ratio K_1 will have a positive effect on the processes inherent in tribosystems during operation.

Coefficient K_2 – characterizes the magnitude of the change of the output parameters (volumetric wear rate and friction coefficient) when changing the values of the input parameters (load, slip speed, quality factor of the tribosystem).

Based on the analysis of dimensions, we write an expression to determine the coefficient K_2:

$$K_2 = \frac{W_{FR} \cdot K_f}{Q_{\max} \cdot a_g}, \tag{5}$$

where W_{FR} – the rate of dissipation in the tribosystem, J/s, is calculated by the formulas given in the work [9];

a_g – the coefficient of thermal conductivity of materials of movable a_{mov} and fixed a_{fix} triboelements, dimension m^2/s, calculated by expression:

$$a_g = \frac{2 \cdot a_{mov} \cdot a_{fix}}{a_{mov} + a_{fix}}, \text{ m}^2/\text{s}. \tag{6}$$

The physical meaning of the coefficient K_2 – this is the sensitivity of the tribosystem to changes in external influences (load, slip speed, quality factor of the tribosystem). Great value of the coefficient K_2 will promote the appearance of fluctuations in wear rate and friction coefficient during the operation of the tribosystem, especially in transient modes. Conversely, small value K_2 will positively affect the operation of the tribosystem.

From the analysis of formula (5) it is possible to develop recommendations for reducing the value of the coefficient K_2. To do this, increase the maximum value of the quality factor of the tribosystem and the value of the coefficients of thermal conductivity of movable materials a_{mov} and fixed a_{fix} triboelements.

Coefficient K_3 – characterizes the ability of the tribosystem to self-organize when changing the values of the input parameters (load, slip speed, quality factor of the tribosystem).

Based on the analysis of dimensions, we write an expression to determine the coefficient K_3:

$$K_3 = \frac{RS^2_{TS(\max)} \cdot a_g}{\dot{\varepsilon}_g}, \tag{7}$$

where $RS_{TS(max)}$ – maximum value of rheological properties of connected materials in the tribosystem after completion of running-in, dimension m^{-1}, is calculated by the formulas given in the work [12];

$\dot{\varepsilon}_g$ – the value of the rate of deformation of the surface layers of the materials of movable and fixed triboelements, the dimension s^{-1}, is calculated by the formulas given in the work [9].

$$\dot{\varepsilon}_g = \frac{2 \cdot \dot{\varepsilon}_{mov} \cdot \dot{\varepsilon}_{ftx}}{\dot{\varepsilon}_{mov} + \dot{\varepsilon}_{ftx}}, \quad 1/s. \tag{8}$$

Deformation rates in the surface layers of the movable $\acute{\varepsilon}_{mov}$ and fixed $\acute{\varepsilon}_{fix}$ triboelements are determined by the expressions given in the work [9]:

$$\dot{\varepsilon}_{mov} = 75\left(1+\mu_{mov}\right)\left(0,86-1,05\mu_{mov}\right)\frac{\sigma_{acs} \cdot v_{sl}}{E_{mov} \cdot d_{acs}}, \ 1/s; \tag{9}$$

$$\dot{\varepsilon}_{fix} = 75\left(1+\mu_{fix}\right)\left(0,86-1,05\mu_{fix}\right)\frac{\sigma_{acs} \cdot v_{sl}}{E_{fix} \cdot d_{acs}}, \ 1/s, \tag{10}$$

where μ_{mov} and μ_{fix} – Poisson's ratios of materials of movable and fixed triboelements, reference value;

v_{sl} – sliding speed, m/s;

E_{mov} and E_{fix} – modulus of elasticity of materials of movable and fixed triboelements, Pa.

The physical meaning of the coefficient K_3 – it is the ability of the tribosystem to return to the conditions of stable functioning after the cessation of action on the tribosystem of factors that outrage. According to the principle of Le Chatelier - Brown, any physical system that was in equilibrium is exposed to an outrageous factor, the equilibrium in the system is shifted so that the effect of this factor is weakened. Therefore, the inertial link G_3 in work [1] included in the scheme of the second block in the form of negative feedback and takes into account the ability of the tribosystem to weaken the perturbing force, by rearranging the surface layers of materials from which the triboelements are made during secondary running-in. Outrageous forces include load, sliding speed, tribological characteristics of the lubricating medium, quality factor of the tribosystem.

Great value of the coefficient K_3 will contribute to the rapid change in the roughness of the friction surfaces, the restructuring of the structure of the surface layers, the appearance of oxidizing films on the friction surfaces (secondary structures). It can be assumed that the coefficient K_3 characterizes the structural adaptability of materials in the tribosystem, or processes of self-organization during operation, especially in transient modes. Conversely, small value K_3 will negatively affect the work of the tribosystem, the ability to self-organize will be low.

From the analysis of formula (7) it is possible to develop recommendations for increasing the value of the coefficient K_3. This requires increasing the value of the rheological properties of the structure of the bonded materials in the tribosystem $RS_{TS(\max)}$, and the values of the coefficients of thermal conductivity of movable materials a_{mov} and fixed a_{fix} triboelements and reduce the values of the deformation rate of the surface layers of the materials of the movable and fixed triboelements.

Time constant T_1, included in the differential equations of the dynamics of the tribosystem (1) characterizes the inertia of increasing the values of the quality factor of the tribosystem when changing external conditions (load, sliding speed, tribological properties of the lubricating medium).

The physical meaning of the time constant T_1 – this is the time required to change the roughness of the friction surfaces and rearrange the structure of the materials of the surface layers when changing external conditions, dimension - second.

Using the accumulated experience in running-in tribosystems, we write an expression for definition T_1:

$$T_1 = \frac{t_g}{3}, \text{ s}, \tag{11}$$

where t_g – running-in time of the tribosystem, dimension - second.

Time constant T_2, which is included in the left part of the differential equation (1), characterizes the time during which there is a stabilization of the temperature gradient by volume of triboelements, taking into account the thermal conductivity of materials when changing external conditions, dimension - second.

Using the methods of dimensional analysis, we write an expression to determine T_2:

$$T_2 = \frac{V_g}{a_g \cdot d_{acs} \cdot n_{acs}}, \text{ s}, \tag{12}$$

where V_g – the given volume of material of a tribosystem is defined by expression:

$$V_g = \frac{2 \cdot V_{mov} \cdot V_{fix}}{V_{mov} + V_{fix}}, \text{ m}^3 \tag{13}$$

where V_{mov} and V_{fix} – volumes of materials of movable and immovable triboelements, which are located under the working surfaces of friction, m^3.

The diameter of the actual contact spot d_{acs} and the number of contact spots on the friction surface n_{acs} depends on the load N, the modulus of elasticity of the contacting materials and the roughness of the friction surfaces. It is calculated according to the formulas given in the work [9].

The physical meaning of the time constant T_2 – this is the time of temperature equalization generated on the spots of actual contact. Decrease in size T_2 will help reduce the time of temperature equalization. As follows from expression (12) for this it is necessary:

- to reduce the volume of triboelements, for example, to execute them thin-walled or to apply a covering or plates on friction surfaces;
- increase the thermal conductivity of triboelement materials.

Time constant T_3, which is included in both the right and left part of the differential equation (1), characterizes the time during which the return of the tribosystem to a stable mode of operation after the cessation of the outrageous force, or the time to stabilize the parameters in the new mode of operation. Performed by rearranging the surface layers when changing external conditions, dimension - second. This is the time during which there is a change in the roughness of the surface layers, the processes of deformation and hardening of the surface layers.

Using the methods of dimensional analysis, we write an expression to determine T_3:

$$T_3 = \frac{V_{dg}}{\dot{\varepsilon}_g \cdot d_{acs}^3 \cdot n_{acs}}, \text{ s}, \tag{14}$$

where V_{dg} – the volume of deformed surface layers is given, m^3, is determined by the expression:

$$V_{dg} = \frac{2 \cdot V_{dmov} \cdot V_{dfix}}{V_{dmov} + V_{dfix}}, \text{ m}^3. \tag{15}$$

The volume of movable material V_{dmov} and fixed V_{dfix} triboelements, m^3, involved in the deformation during friction, is calculated by formulas (2) and (3). The depth of strain propagation is determined by the expressions given in the paper [9]:

$$h_{dmov} = 0{,}5 d_{acs}(1 - e^{-D_{mov}}), \tag{16}$$

where

$$D_{nov} = \frac{6{,}5 \cdot 10^8 \cdot \sigma_{acs}^2}{E_{mov} \cdot E_u}, \tag{17}$$

$$h_{dfix} = 0{,}5 d_{acs}(1 - e^{-D_{fix}}), \tag{18}$$

where

$$D_{fix} = \frac{6{,}5 \cdot 10^8 \cdot \sigma_{acs}^2}{E_{fix} \cdot E_u}, \tag{19}$$

where D_{mov} and D_{fix} – coefficients of deformation in movable and fixed triboelements, dimensionless quantities.

Reducing the time constant T_3 helps to reduce the time of self-organization of the tribosystem. As follows from expression (11), to reduce T_3 the following measures must be taken.

1. Reducing the depth of deformation in the surface layers. As follows from formulas (16) - (19) the depth of deformation in the materials of triboelements is affected:

- the magnitude of the voltage at the spots of actual contact σ_{acs}, which must be reduced;
- modulus of elasticity of triboelement materials E_{mov}, E_{fix}, which must be increased;
- tribological properties of the lubricating medium E_u, which need to be increased.

2. Increasing the rate of deformation in the surface layers of triboelements. Ways to increase $\acute{\varepsilon}_{mov}$ and $\acute{\varepsilon}_{fix}$ follow from expressions (9) and (10).

Analysis of these expressions allows us to conclude that the values σ_{acs}, E_{mov}, E_{fix} are in contradiction with the conclusions made earlier. So the magnitude T_3 will be optimal for the process of self-organization. One of the parameters that can control the value T_3, this is the sliding speed υ_{sl}. Increasing the sliding speed leads to an increase in the strain rate under pre-selected and constant conditions.

The expressions for determining the gain are obtained K_1-K_3, formulas (4) - (7), as well as time constants T_1-T_3, formulas (11) - (14), is the result of parametric identification of the mathematical model of functioning of tribosystems in the conditions of extreme lubrication.

Conclusions

Parametric identification of the tribosystem as an object of modeling the functioning of tribosystems in the conditions of boundary lubrication is performed. Using the analysis of the dimensions of significant factors, expressions are obtained to calculate the gain and time constants.

It is established that the coefficient K_1 takes into account the degree of influence of the load, sliding speed, tribological characteristics of the lubricating medium on the quality factor of the tribosystem. It is shown that the increase in the coefficient K_1 will have a positive effect on the processes inherent in tribosystems during operation. Coefficient K_2 – characterizes the magnitude of the change in volumetric wear rate and coefficient of friction when changing the magnitude of the load, sliding speed, quality factor of the tribosystem. Coefficient K_3 – characterizes the ability of the tribosystem to self-organize when changing the values of the input parameters by rearranging the surface layers of materials from which the triboelements are made during secondary running-in. Outrageous forces include load, sliding speed, tribological characteristics of the lubricating medium, quality factor of the tribosystem. It is shown that the value of the coefficient K_3 is large, will contribute to the rapid change in the roughness of the friction surfaces, the restructuring of the structure of the surface layers, the appearance of oxidizing films on the friction surfaces (secondary structures).

It is proved that the time constant T_1 – this is the time required to change the roughness of the friction surfaces and rearrange the structure of the materials of the surface layers when changing external conditions. Time constant T_2 characterizes the time during which there is a stabilization of the temperature gradient by volume of triboelements, taking into account the thermal conductivity of materials when changing external conditions. Time constant T_3 characterizes the time for which the tribosystem returns to a steady state of operation after the cessation of the outrageous force, or the time to stabilize the parameters in the new mode of operation. It is proved that the value T_3 will be optimal for the process of self-organization. One of the parameters that can control the value T_3, this is the sliding speed υ_{sl}. Increasing the sliding speed leads to an increase in the strain rate under pre-selected and constant conditions.

The expressions for determining the coefficients are obtained K_1 - K_3, as well as time constants T_1 - T_3, is the result of parametric identification of the mathematical model of functioning of tribosystems in the conditions of extreme lubrication. The value of these coefficients will be used in modeling the processes of friction and wear in tribosystems when changing design, technological and operational parameters, which will allow to choose rational designs for specific conditions of their operation.

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Войтов А.В. Параметрична ідентифікація математичної моделі функціонування трибосистем в умовах граничного мащення.

Виконано параметричну ідентифікацію трибосистеми, як об'єкта моделювання функціонування трибосистем в умовах граничного мащення. За допомогою аналізу розмірностей значимих факторів отримано вирази для розрахунку коефіцієнтів підсилення та постійних часу.

Встановлено, що коефіцієнт K_1 враховує ступінь впливу навантаження, швидкості ковзання, трибологічних характеристик змащувального середовища на величину добротність трибосистеми. Показано, що збільшення коефіцієнта K_1 буде позитивно впливати на процеси, які притаманні трибосистемам під час експлуатації. Коефіцієнт K_2 – характеризує величину зміни об'ємної швидкості зношування і коефіцієнта тертя при зміні величин навантаження, швидкості ковзання, добротності трибосистеми. Коефіцієнт K_3 – характеризує здатність трибосистеми до самоорганізації при зміні величин вхідних параметрів шляхом перебудови поверхневих шарів матеріалів з яких виготовлені трибоелементи під час вторинного припрацювання. До сил, що обурюють, відносяться навантаження, швидкість ковзання, трибологічні характеристики змащувального середовища, добротність трибосистеми. Показано, що велике значення коефіцієнта K_3 буде сприяти швидкої зміні величин шорсткості поверхонь тертя, перебудові структури поверхневих шарів, появі окиснювальних плівок на поверхнях тертя (вторинних структур).

Доведено, що постійної часу T_1 – це час, який необхідно для зміни шорсткості поверхонь тертя та перебудови структури матеріалів поверхневих шарів при зміні зовнішніх умов. Постійна часу T_2 характеризує час, за який відбувається стабілізації градієнта температур за об'ємами трибоелементів з урахуванням температуропровідності матеріалів при зміні зовнішніх умов. Постійна часу T_3 характеризує час, за який відбувається повернення трибосистеми до сталого режиму функціонування після припинення дії сили, що обурює, або час до стабілізації параметрів на новому режимі функціонування. Доведено, що величина T_3 матиме оптимальне значення для процесу самоорганізіції. Доведено, що одним з факторів, яким можна керувати величиною T_3, це швидкість ковзання $\upsilon_{ков}$. Збільшення швидкості ковзання призводить до збільшення швидкості деформації при заздалегідь обраних і незмінних умовах.

Отримані вирази для визначення коефіцієнтів K_1-K_3, а також постійних часу T_1-T_3, є результатом параметричної ідентифікації математичної моделі функціонування трибосистем в умовах граничного мащення. Значення цих коефіцієнтів буде використано при моделюванні процесів тертя та зношування в трибосистеми при зміні конструктивних, технологічних та експлуатаційних параметрів, що дозволить обирати раціональні конструкції для конкретних умов їх експлуатації.

Ключові слова: трибосистема; математична модель; диференційні рівняння; параметрична ідентифікація; коефіцієнт посилення; постійна часу; граничне змащення; добротність трибосистеми; швидкість роботи дисипації

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System analysis of friction and wear processes when using fullerene compositions in lubricants

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Abstract

The system-structural approach in researches of processes of friction and wear at application of fullerene compositions in lubricants is proved in the work. It is proposed to use a multilevel approach to study and model the processes of deformation of the surface layers of movable and fixed triboelements and the formation on energy-activated surfaces of wear-resistant structures containing fullerene molecules. The essence of the approach is to use multi-scale research methods to build mathematical models within a single research structure. Due to the fact that tribosystems differ in the integrity of the interconnected elements included in them, it is assumed that all processes occur at three hierarchical levels. At this level, they interact with each other and exchange energy and matter.

Input and output flows in studies of tribosystems are formulated. It is shown that the input streams include design parameters of the tribosystem, technological parameters, operating parameters. These parameters form the flow of matter, energy and information, which is the input effect on the tribosystem. The output flow from the tribosystem are the parameters: volumetric wear rate I, dimension m^3/hour; friction losses, which are estimated by the coefficient of friction f, dimensionless quantity. The output stream is the information flow of the tribosystem. When solving contact problems, this allows to take into account not only the level of stresses, but also the speed of deformation in the materials of the surface layers, as well as the depth of deformation, which in the models will take into account the volume of deformed material.

Depending on the tasks and requirements for their solution, the use of different methodological approaches for modeling is justified. It is shown that the application of mathematical models in the modeling of tribological processes depends on the correct choice of technical constraints that determine the range of optimal solutions.

Key words: fullerenes; fullerene solvent; fullerene compositions; tribosystem structure; dissipation speed; electrostatic field of the friction surface; deformation rate; volumetric wear rate; coefficient of friction.

Introduction

Processes of friction and wear in various designs of tribosystems belong to dynamic processes and develop according to the general laws of synergetics. A distinctive feature of such processes is the adaptation of the surface layer of tribosystems to the conditions of friction, which is called B.I. Kostecki structural adaptation of materials by friction, and then L.I. Bershadsky - adaptation, ability to learn and self-organization of tribosystems.

Self-organization is a fundamental phenomenon of nature, which is manifested in various areas of animate and inanimate nature. The essence of self-organization in tribosystems is that under the action of external perturbation the tribosystem adapts (learns, changes) so that its response to external perturbations maximally compensates for the cause of such perturbation, ie the ability of the tribosystem to return to stable conditions after cessation on the tribosystem of outrageous factors.

The use of fullerenes as antiwear, extreme pressure and antifriction additives to technical liquid lubricants gives an ambiguous answer about their effectiveness and limits of use. Such lubricants react to external influences on the tribosystem and are able to change the structure of surface layers, adapting to operating conditions. Controlling the process of formation of such structures will increase wear resistance and reduce friction losses of machines and mechanisms, which will help to save energy resources during operation.

One of the ways to solve this problem is the development of a systematic approach in studying the processes of friction and wear in the presence of fullerene compositions in lubricants and modeling of such processes in structures formed on the friction surface. The simulation results will allow to substantiate the composition and content of fullerene additives to lubricants for various purposes and groups of operation.

Literature review

In the last twenty years, a number of publications have appeared, where with the help of theoretical [1] and experimental research [2], as well as computer simulation [3] new knowledge about the processes of friction and wear of hard surfaces in the presence of an ultrathin film of liquid between them. In the given works it is established that in the course of functioning of a tribosystem the lubricating film becomes more and more thin, at first its physical properties change gradually, and then changes get sharply expressed character. Qualitative changes are expressed in the non-Newtonian shift mechanism and, according to the authors, in the replacement of ordinary melting - by glazing. However, the film continues to behave like a liquid.

In such films, phase transitions of the first kind to solid or liquid phases are possible, the existence of which has been proved in [4], whose properties cannot be described by such a term as viscosity. These films are characterized by a yield strength, which is a characteristic of fracture in solids and a large stress relaxation time.

In work [5] describes the dynamic properties of thin lubricating films by friction in the limit lubrication mode. Experiments performed on smooth friction surfaces in the presence of surfactants have shown that such phase transitions occur and are confirmed by other researchers [6]. Such processes cause stick-slip motion, which is characteristic of friction without lubricant and is characterized by periodic transitions between two or more dynamic states during the stationary process.

Since the intermittent motion is observed at a constant temperature of the friction surfaces, to explain it by the authors [6, 7] the concept of "shear melting" is offered. According to this concept, first the oil is solid (stick), then, when some critical stress value is exceeded, the oil abruptly turns into a liquid phase (slip) as a result of loss of strength. Upon further movement, under the action of the load, the oil again becomes solid (stick). According to the authors [8], such molecular rearrangement processes have a correlation in thin films at a short distance from the friction surfaces.

According to the work [9] the liquid state of thin films is characterized by an effective viscosity, which is many orders of magnitude higher than the viscosity for a bulk liquid and is non-Newtonian. This means that the effective viscosity decreases with increasing sliding speed, ie the thin film reduces the shear stress [10]. Under different conditions of sliding the authors of the work [11] it is established that the film changes the thickness and structure. In addition, liquids containing multicomponent additives undergo dynamic phase transitions, which is manifested in the appearance of intermittent modes of motion [12].

According to the experimental data presented in the work [13], oil on the friction surface is a very viscous fluid that behaves like an amorphous solid and is characterized by a yield strength. Therefore, based on rheological description of viscoelastic medium, which has a thermal conductivity in the [5] a system of kinetic equations is obtained, which agree and determine the behavior of shear stresses and strains, as well as the temperature in the thin film of oil on the friction surface [14]. The obtained rheological equations and the results of modeling using equations, allowed the authors to conclude that the value of the effective viscosity is very different from the value of the bulk viscosity and depends on the temperature. According to the authors [5] the specified feedback between the magnitude of stresses, temperature and deformation means that the transition of oil from solid to liquid state due to both heating and the effects of stresses created by solid friction surfaces. This is consistent with the consideration of the instability of the solid phase in the representation of shear dynamic melting in the absence of thermal fluctuations [15].

The results of the above analysis allow us to expand the existing understanding of the physics of processes occurring on the friction surfaces of tribosystems operating in the mode of extreme lubrication. This is especially true at the present stage of development of the science of tribology, when nanosized particles are used in lubricants and the classical law of Amonton-Coulon is not fulfilled. This is the opinion of the authors of the work [5]. The authors of this work argue that the melting of the ultrathin film of oil between the solid friction surfaces, is presented as a result of shear stresses and rapid heating of a small local volume. The critical temperature of the local volumes of the friction surface at which melting occurs increases with increasing characteristic value of shear viscosity and decreases with increasing modulus of shear of the oil according to the linear law. It is shown that the intermittent friction mode (stick-slip) is realized if the relaxation time of the temperature in the oil is much higher than the time value for shear stresses and strains.

The results of studies of boundary friction in the framework of a synergetic model based on the idea that the system is self-organizing are presented in work [16]. The paper describes the behavior of the limiting lubricant during the mutual movement of the friction surfaces, in particular, the studied hysteresis phenomena and fractal characteristics of time series of stresses. However, the author suggests that the properties of the oil layer are the same both inside the layer and near the contact surface. According to the results of the study of spatially inhomogeneous systems, the behavior of the lubricant in the process of friction is non-trivial. For example, at work [17] "vortex-like" movement of lubricant in the contact zone was detected. In addition, in [17] it is shown that during dry friction of rough surfaces, between them, as a result of mechanical action, a quasi-liquid layer is formed, as a result of which the coefficient of friction decreases with increasing shear rate. It is established that in the course of evolution the system strives for a homogeneous state in which shear stresses are realized in the whole region of contact, which have a constant value, which determines the relative speed of friction blocks.

In works [18, 19] attention is focused on the practical features of the use of lubricants with functional additives that provide a positive effect both in the manufacturing process and at the stage of operation of tribosystem parts. At the same time, the mechanisms of action of lubricating compositions on operational factors were not analyzed.

The analysis of the researches devoted to the thin-film object - the oil adsorbed on a friction surface allows to state that in the course of interaction of friction surfaces, the lubricating film has some phases: solid-like; rare and mixed. The phases, under the action of stress on the spots of actual contact and the rate of deformation of the material of the surface layers on the spots of contact, pass into each other. The classical term viscosity is not suitable for such phases, it is proposed to use the term - effective viscosity, the value of which is several orders of magnitude greater than the viscosity in the volume of lubricant. The magnitude of the roughness of the friction surface is a significant factor. The analysis also shows that the study of such thin-film objects is better done using the basic principles of the science of rheology, using the rheological equations of flow in such films. Based on the conclusions, we can put forward a working hypothesis that in the presence of nanoparticles in the lubricant, which are included in the form of aggregates (clusters, micelles), the effective viscosity in the volume of the thin-film object will act as a significant factor and determine wear resistance and friction losses. Determining the value of the effective viscosity and the dependences of its change on the magnitude of external influences, connected materials in the tribosystem, the design of the tribosystem and the tribological properties of the basic lubricant will control the processes of friction and wear.

Summing up the analysis of works on the formation of lubricating films on friction surfaces and the factors influencing this process, we can conclude that the aim of this study is to develop a system-structural approach in the study of friction and wear in the application of fullerene compositions in lubricants and theoretical studies of the formation of the lubricating film in the presence of such compositions. This task requires the development of a mathematical model of the interaction of electrically active heterogeneous fine systems at the interface friction surface - lubricating medium and modeling of such processes. The model should take into account the generation of friction surfaces of the connected materials of the electrostatic field and the influence of this force field on the electrically active units in the lubricating medium. As a result of such interaction, a lubricating film of a certain thickness and structure is formed on the friction surfaces.

Purpose

The purpose of this work is to develop a systematic approach to studying the processes of friction and wear in the presence of fullerene compositions in lubricants and to simulate such processes in structures formed on the friction surface. The simulation results will allow to substantiate the composition and content of fullerene additives to lubricants for various purposes and groups of operation.

Methods

The technical term tribosystem will mean a complex of at least four elements E and existing links between them R, which form a single set and operate within a more complex system of which it is part, ie $S = (E, R)$. Each tribosystem can be divided into subsystems, while maintaining the existing connections within the system, which allows you to consider the resulting subsystems separately. This division was first performed by G. Chikhos, where the subsystems are called friction planes. A characteristic feature of systems analysis is that when studying part of the population it is necessary to take into account the whole set of elements and connections.

Under the input streams we will understand: design parameters of the tribosystem; technological parameters; operating parameters.

The output flow from the tribosystem are the parameters: volumetric wear rate I, dimension m^3/hour; friction losses, which are estimated by the coefficient of friction f, dimensionless quantity. The output stream is the information flow of the tribosystem.

The task of this work is to study the processes of formation of surface structures of triboelements in the presence of fullerene compositions in the lubricant and the mechanisms of influence of such structures on the volumetric wear rate and friction coefficient. According to the formulated task is subject to change:

- concentration of fullerenes in the basic lubricant, dimension g/kg;

- concentration of fullerene compositions containing fullerene powder and vegetable oil as solvent of fullerene powder, dimension g/kg;

- tribological properties of the basic lubricant to which fullerenes or fullerene compositions are added, dimension J/m^3;
- coefficient of shape of the tribosystem, dimension m^{-1};
- structure of connected materials in the tribosystem, which is taken into account by a complex parameter - the internal friction of triboelement materials;
- load-speed range of operation or operation of the tribosystem, which is taken into account by the product of load and sliding speed, dimension J/s.

Flows of materials and energy are integral components of the processes of formation on the friction surface of wear-resistant structures, and the flow of materials reflects the object of influence, and the flow of energy - a means of influence.

In the framework of this work it is proposed to use a multilevel approach to study and model the processes of deformation of the surface layers of movable and fixed triboelements and the formation on energy-activated surfaces of wear-resistant structures containing fullerene molecules. The essence of the approach is to use multi-scale research methods and approaches to building mathematical models within a single research structure.

Due to the fact that the tribosystem differs in the integrity of the interconnected elements included in it, we assume that all processes occur at three hierarchical levels, fig. 1. At this level, they interact with each other and exchange energy and matter.

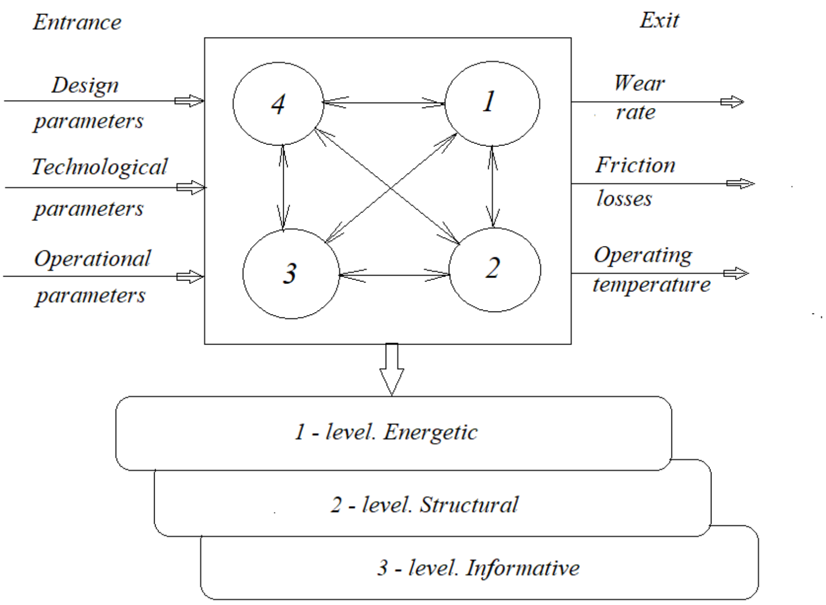

Fig. 1. Hierarchy of levels of the tribosystem:
1 – movable triboelement;
2 – fixed triboelement;
3 – lubricating or working medium;
4 – environment

Results

The first level in the hierarchy of the tribosystem is the energetic level. In the study of processes of this level, the input parameters are design, technological and operational factors, as shown in fig. 2.

The initial parameters are the speed of dissipation in the tribosystem – W_{TS}, dimension J/s. The rate of dissipation in the tribosystem is the part of the energy that goes to change the structure of the surface layers of materials of movable and fixed triboelements.

The energy (power) that is supplied to the tribosystem can be determined by expression:

$$W = N \cdot v_{sl}; \left[N \cdot \frac{m}{s} = \frac{J}{s} = W \right], \tag{1}$$

where N – is the load on the tribosystem, N;

v_{sl} – is the sliding speed, m/s.

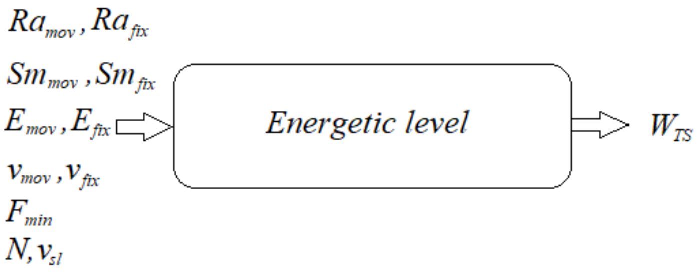

Fig. 2. Block diagram of the energy level of the tribosystem

The input parameters that affect the rate of dissipation in the tribosystem are:

1. Technological parameters - parameters of roughness of contacting friction surfaces:

- Ra_{mov}, Ra_{fix} – arithmetic mean deviation of points of profile of movable and fixed triboelements, m;

- Sm_{mov}, Sm_{fix} – average step of inequalities along the middle line of the profile of movable and fixed triboelements, m.

Parameters Ra and Sm are determined in accordance with GOST 2789-73.

2. Physico-mechanical properties of contact materials in the tribosystem:

- E_{mov}, E_{fix} – modulus of elasticity of materials of movable and fixed triboelements, Pa;

- υ_{mov}, υ_{fix} – Poisson's ratio of materials of movable and immovable triboelements.

3. Design parameters of the tribosystem:

- $F_{\min}$ – smaller area of friction of one of the triboelements, m^2.

- $\sigma_n = N/F_{\min}$ – rated voltage at contact of triboelements, Pa.

The rate of dissipation in the tribosystem, according to the work [20] is determined by the expression:

$$W_{TS} = W_{TS,mov} + W_{TS,fix}, \tag{2}$$

where $W_{TS,mov}$ and $W_{TS,fix}$ – is the speed of dissipation in movable and fixed triboelements, dimension J/s.

The speed of dissipation in movable and fixed triboelements, according to the work [20] is determined by the expression:

$$W_{TS,mov} = \sigma_{acs} \cdot \dot{\varepsilon}_{mov} \cdot V_{dmov} \cdot n \text{, J/s,} \tag{3}$$

$$W_{TS,fix} = \sigma_{acs} \cdot \dot{\varepsilon}_{fix} \cdot V_{dfix} \cdot n \text{, J/s,} \tag{4}$$

Voltage at the actual contact spots (ACS) – σ_{acs}, dimension Pa, and the number of contact spots n depends on the load on the tribosystem N, N, modulus of elasticity and roughness of contact materials of triboelements and is calculated by the formulas given in work [20].

The deformation rate of the material of the movable triboelement per unit ACS is calculated by expression [20]:

$$\dot{\varepsilon}_{mov} = 75(1+\upsilon_{mov})(0{,}86 - 1{,}05\upsilon_{mov})\frac{\sigma_{acs} \cdot \upsilon_{sl}}{E_{mov} \cdot d_{acs}} \text{, 1/s,} \tag{5}$$

for the material of the fixed triboelement:

$$\dot{\varepsilon}_{fix} = 75(1+\upsilon_{fix})(0{,}86 - 1{,}05\upsilon_{fix})\frac{\sigma_{acs} \cdot \upsilon_{sl}}{E_{fix} \cdot d_{acs}} \text{, 1/s.} \tag{6}$$

The diameter of the actual contact spot d_{acs}, m, calculated according to the formulas given in the work [20].

The volume of movable material V_{dmov} and fixed V_{dfix} triboelements, m^3, which participates in deformation in the process of friction, is calculated by the formulas given in the work [20]:

$$V_{dmov} = F_{\max} \cdot h_{dmov}, m^3, \tag{7}$$

$$V_{dfix} = F_{\min} \cdot h_{dfix}, m^3, \tag{8}$$

where $F_{\max}$ –is the large area of friction of one of the triboelements, m^2.

Depth of deformation of the surface layers of the movable material h_{dmov} and fixed h_{dfix} triboelements is calculated by the formulas given in the work [20]:

$$h_{dmov} = 0{,}5d_{acs}\left(1 - e^{-D_{mov}}\right), \text{ m}, \tag{9}$$

$$h_{dfix} = 0{,}5d_{acs}\left(1 - e^{-D_{fix}}\right), \text{ m}, \tag{10}$$

where D_{mov} and D_{fix} – is the coefficients that take into account the ability of the material to deform under the action of surfactants, for movable and fixed triboelements, respectively, dimensionless values. Calculated on the basis of work [20]:

$$D_{mov} = \frac{6{,}5 \cdot 10^8 \sigma_{acs}^2}{E_{mov} \cdot E_u}, \tag{11}$$

$$D_{fix} = \frac{6{,}5 \cdot 10^8 \sigma_{acs}^2}{E_{fix} \cdot E_u}, \tag{12}$$

where E_u – is the tribological properties of the lubricating medium, J/m^3, are determined on a four-ball friction machine, take into account the anti-wear and anti-emergency properties of lubricants, are calculated by the formula given in the work [21].

The processes that take place at the energy level are as follows. Under the action of load, sliding speed and temperature gradient, the formation of equilibrium roughness on the friction surfaces. The diameter of the actual contact spot changes d_{acs} in the direction of increase, which leads to a decrease in stress σ_{acs} on the spot of actual contact. It should be noted that during the running-in of the tribosystem, the diameter of the actual contact spot increases slightly, not more than twice. However, the number of contact spots n increases by an order of magnitude. After completion of the running-in process, the number of contact spots and their diameter are stabilized near equilibrium [22].

Due to the decrease in stresses on the spots of actual contact and the simultaneous increase in the diameter of the spots, the rate of deformation of the materials of the surface layers decreases, this follows from the formulas (5) and (6). At the same time, the depth of deformation in the surface layers decreases, this follows from the formulas (7) - (12).

As a result, after completion of running-in, the surface layers of the movable and fixed triboelements form a certain structure of the material, which corresponds to the input effect on the tribosystem. When you change the magnitude of the input action, all of the above processes will change, so the structure of the surface layers will change.

The criterion that is a measure of such changes is the rate of dissipation in the tribosystem – W_{TS}, dimension J/s. This criterion is a way out of the energy level of the tribosystem, fig. 2 and at the same time is the entrance to the second level - the structural level of the tribosystem.

The block diagram of the second - structural level of the tribosystem is presented in fig. 3.

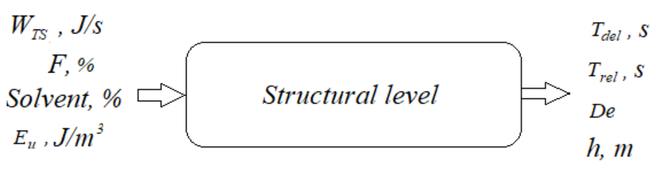

Fig. 3. Block diagram of the structural level of the tribosystem

Under the action of energy, the value of which is estimated by the speed of dissipation, the surface layers of movable and fixed triboelements act as a "generator of electrostatic force field". The force field of the friction surfaces will affect the formation of a lubricating film on the friction surfaces in the presence of solutions of fullerenes in the lubricant.

Adding electrically active heterogeneous fine systems of different concentrations to the basic lubricating medium will create an electrostatic field in the volume of lubricant (liquid). The increase in the force field in the volume of the liquid is associated with a high value of the dipole moment of the fullerene molecule, which is equal to $3{,}34 \cdot 10^{-30}$ C·m. Fullerene molecules, in the field of electrostatic forces of friction surfaces, will form clusters that actively interact with the "electrostatic field generator". The result of this interaction is the formation of wear-resistant structures on the friction surfaces.

Based on the analysis of work on the use of fullerenes, it is concluded that fullerenes are insoluble in technical oils of petroleum, semi-synthetic or synthetic origin. Such systems are characterized by sedimentation processes. In this paper, a working hypothesis is formulated that the use of "solvents" of fullerenes significantly increases the electrostatic field strength of the liquid. As solvents for fullerenes, you can use high-oleic vegetable oils, such as rapeseed, which are well soluble in all types of technical oils. During the solution of fullerene molecules in vegetable oil, micelles are actively formed, where the nucleus of the micelle is a fullerene molecule, or several fullerene molecules. The application of such a technological approach as the preliminary dissolution of fullerene molecules in vegetable oil, and then the addition of such a composition to base oils, will significantly increase the strength of the electrostatic field of the liquid. This is due to the fact that the dipole moment of the micelles based on fullerenes, which is equal to $p_m = 9{,}04 \cdot 10^{-26}$ C·m, an order of magnitude greater than the dipole moment of fullerene-based clusters, which is equal to $p_k = 3{,}31 \cdot 10^{-27}$ C·m. This will create more effective wear-resistant structures on the friction surfaces in comparison with technological approaches, where there is no pre-dissolution of fullerenes.

In the process of functioning of the tribosystem due to the influence of temperature, as well as load and sliding speed, the process of cluster and micelle formation, as well as their destruction, can occur simultaneously, therefore, the total electrostatic field of the lubricating medium E_{fl} defined as the sum:

$$E_{fl} = E_k + E_m, \text{ V/m},$$

where E_k and E_m – is the voltage of the electrostatic field of the liquid due to the formation of clusters and micelles, dimension V/m;

Working hypothesis on the formation of wear-resistant surface structures based on fullerene compositions (vegetable oil + fullerenes), has rational limits of effective use. It is necessary to confirm theoretically and experimentally that for tribosystems having a certain design and load-speed range of operation, there are optimal modes of operation, when the friction surface generates the maximum value of electrostatic field strength, which is the driving force for electrostatic field formation in the lubricating film volume.

The lubricating film formed on the friction surface can act as two structures, as an elastic solid, or as a viscous non-Newtonian fluid. In such structures, the process of stress relaxation at the spots of actual contact will take place in different ways, which requires the development of a mathematical model. The mathematical model should consist of a macro-rheological and micro-rheological model of stress relaxation on the actual contact spots in the presence of fullerene compositions in lubricants. The macroreological model can be represented in the form of second-order differential equations and their solutions, the microreological model in the form of expressions for determining the parameters included in the differential equations and their solutions. Solutions of differential equations will allow modeling the process of stress relaxation on actual contact spots in the tribosystem, which allows to determine the friction losses.

When planning research at the second - structural level, the following working hypothesis is formulated. The formation of a lubricating film on the friction surface of tribosystems containing fullerene compositions, in contrast to the known, must take into account the structural viscosity and structure of the formed film under the action of the electrostatic field of the friction surface. The working hypothesis of formation of such structures is offered, where films in the field of action of electrostatic forces acquire structure of gel, and out of action of electrostatic forces - structure of sol. According to the hypothesis, a framework of "crosslinked" molecules of fullerenes and oleic acid is formed on the friction surface, which absorbs stress. In the process of sliding, under the action of stresses, the framework can collapse, and fullerene molecules make rotational movements between the friction surfaces, which leads to a decrease in viscosity (the liquid acquires non-Newtonian properties). After coming out of contact, the structure of the frame is restored under the action of electrostatic forces of the surface. It is assumed that the structural viscosity of the lubricating film is influenced by the magnitude of the electrostatic field of the friction surface, the orientation of the flocks to the friction surface and the concentration of fullerenes in the lubricating film in the field of electrostatic forces.

The dependences of the change of the parameters of the rheological model of the stress relaxation process on the actual contact spots, which confirm the working hypothesis, can be performed theoretically, based on the developed rheological model. The dependences of the change in the stress relaxation time in the structure of the lubricating film on the friction surface, as well as the magnitude of the delay time in the stress relaxation and the Deborah number [23], will confirm the working hypothesis that the lubricating film acquires the properties of an elastic solid.

When planning research, it is suggested that such physical quantities as the relaxation time of stresses in the structure of the lubricating film and the Deborah number are a measure of the transition of the viscous properties of the lubricating film into elastic and vice versa - elastic into viscous.

Physical quantity - time of delay in distribution of stresses, characterizes inertia of structure of a lubricating film and possibility of residual deformations in this structure after stress removal. The large value of the delay time characterizes the presence of delays and the presence of residual deformations in the film structure after stress relief. Factors influencing the value of the structural viscosity of the lubricating film formed on the friction surface are to be established.

These processes of the second structural level are intended to formulate criteria for evaluating the elastic or viscous properties of wear-resistant structures on friction surfaces. Such criteria may be the relaxation time of the stresses at the spots of actual contact – T_{rel}, delay time in stress relaxation – T_{del}, Debory's number - De and the thickness of the lubricating film h. These are the initial parameters from the second structural level of the study, fig. 3. These initial values have a functional relationship with the values of the concentration of fullerenes, the concentration of the solvent - vegetable oil and the tribological properties of the basic lubricant, to which are added fullerene compositions. These are the input values of the second structural level of the study of this work.

The block diagram of the third - information level of the tribosystem is given on fig. 4.

The input factors of the third information level are: stress relaxation time on the spots of actual contact – T_{rel}, delay time in stress relaxation – T_{del}, Debory's number - De and the thickness of the lubricating film h. The initial parameters are the volumetric wear rate I, m^3/hour and coefficient of friction.

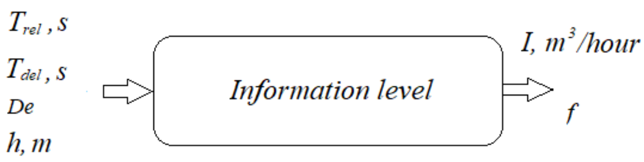

Fig. 4. Block diagram of the third information level of the tribosystem

The third level of information aims to calculate, using mathematical modeling, to determine the volumetric wear rate and friction coefficient. When performing the simulation of the initial parameters, the following assumption was made. Structures formed on friction surfaces, which have the properties of an elastic body or a viscous fluid, have a certain thickness h. The thickness of this structure depends on the magnitude of the voltage of the generated electrostatic field, which is affected by the magnitude of the rate of dissipation in the tribosystem – W_{TS}. The structures formed on the friction surfaces change the roughness of the friction surfaces. Such structures "align" the friction surface by reducing the magnitude Ra and increasing the magnitude Sm in movable and fixed triboelements. This will reduce the rate of dissipation in the tribosystem, which is determined by the formulas (2) - (12). Based on these values, a new value of the dissipation rate in the tribosystem is determined – $W_{TS(F)}$, which corresponds to the use of different concentrations of fullerene compositions in lubricants with different tribological properties, and calculates the value of the volumetric wear rate by expression [24]:

$$I = 6 \cdot 10^{-10} \exp\left(0{,}795 \cdot 10^{16} \cdot \frac{1}{E_u} \sqrt{\frac{\pi}{\left(\delta_{mov} \cdot \delta_{fix}\right)}} \cdot W_{TS(F)} \right), \text{ m}^3\text{/hour}, \tag{13}$$

where δ_{mov} and δ_{fix} – is the internal friction of the structure of materials of movable and fixed triboelements, is calculated by the expressions given in the work [25].

The coefficient of friction, which determines the friction losses in the tribosystem, is calculated by [24]:

$$f = \frac{W_{TS(F)}}{W} = \frac{W_{TS(F),mov} + W_{TS(F),fix}}{N \cdot \upsilon_{sl}}. \tag{14}$$

The third - the information level of processes in the tribosystem allows to theoretically obtain information about the effectiveness of fullerenes or fullerene compositions in lubricants. Based on the results of the research and mathematical modeling, the following practical questions can be answered:

- determine the design of tribosystems, where there is an optimal range of operation in terms of sliding speed and load, which will provide a minimum of friction losses, and the maximum percentage of their reduction compared to basic lubricants without fullerenes;

- determine the rational range of tribological properties of lubricants, the addition of which to fullerene compositions will give the maximum effect of reducing the volumetric wear rate and friction coefficient;

- to determine the influence of material compatibility in the tribosystem, which is determined by the amount of internal friction of the structure of materials of triboelements, which increases the effect of reducing the coefficient of friction from the use of fullerenes in basic lubricants.

The presented research structure demonstrates a multilevel approach within the scientific problem - the formation of surface wear-resistant structures on the friction surfaces of various structures of tribosystems in the presence of lubricants fullerene additives or fullerene compositions. This approach allows a more detailed and comprehensive study of the dynamics of processes occurring on the surface of the contact spots during friction. In particular, when solving contact problems, this allows to take into account not only the level of stresses, but also the speed of deformation in the materials of the surface layers, as well as the depth of deformation, which in the models will take into account the volume of deformed material.

In addition, the use of a multilevel approach allows you to develop models and model processes on the friction surfaces that occur, dividing them into levels within a single scientific problem, which is fundamentally important for the correct solution of dynamic problems of friction. It is shown that the application of mathematical models in the modeling of tribological processes depends on the correct choice of technical constraints that determine the range of optimal solutions. The search for optimal conditions for the use of fullerenes or fullerene compositions as additives to lubricants should be carried out under the condition of selected technical constraints arising from the operating conditions of tribosystems.

Conclusions

The system-structural approach in researches of processes of friction and wear at application of fullerene compositions in lubricants is proved. It is proposed to use a multilevel approach to study and model the processes of deformation of the surface layers of movable and immovable triboelements and the formation on energy-activated surfaces of wear-resistant structures containing fullerene molecules. The essence of the approach is to use multi-scale research methods to build mathematical models within a single research structure. Due to the fact that the tribosystem differs in the integrity of the interconnected elements that are part of it, it is assumed that all processes occur at three hierarchical levels. At this level, they interact with each other and exchange energy and matter.

Input and output flows in studies of tribosystems are formulated. It is shown that the input streams include design parameters of the tribosystem, technological parameters, operating parameters. These parameters form the flow of matter, energy and information, which is the input effect on the tribosystem. The output flow from the tribosystem are the parameters: volumetric wear rate I, dimension m^3/hour; friction losses, which are estimated by the coefficient of friction f, dimensionless quantity. The output stream is the information flow of the tribosystem. It is shown that this approach allows to study in more detail and comprehensively the dynamics of processes occurring on the surface of contact spots during friction. In particular, when solving contact problems, this allows to take into account not only the level of stresses, but also the speed of deformation in the materials of the surface layers, as well as the depth of deformation, which in the models will take into account the volume of deformed material.

Depending on the tasks and requirements for their solution, the use of different methodological approaches for modeling is justified. It is shown that the application of mathematical models in the modeling of tribological processes depends on the correct choice of technical constraints that determine the range of optimal solutions. The search for optimal conditions for the use of fullerenes or fullerene compositions as additives to lubricants should be carried out under the condition of selected technical constraints arising from the operating conditions of tribosystems.

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Кравцов А.Г. Системний аналіз процесів тертя та зношування при застосуванні фулеренових композицій в мастильних матеріалах**.**

В роботі обґрунтовано системно-структурний підхід в дослідженнях процесів тертя та зношування при застосуванні фулеренових композицій в мастильних матеріалах. Запропоновано використовувати багаторівневий підхід для дослідження і моделювання процесів деформації поверхневих шарів рухомого і нерухомого трибоелементів і формування на енергетично активованих поверхнях зносостійких структур, які містять молекули фулеренів. Суть підходу полягає в використанні різномасштабних методик дослідження до побудови математичних моделей в рамках єдиної структури досліджень. У зв'язку з тим, що трибосистеми відрізняються цілісністю взаємопов'язаних елементів, що входять до них, прийнято припущення, що всі процеси відбуваються на трьох ієрархічних рівнях. При цьому рівні взаємодіють між собою і обмінюються енергією і речовиною.

Сформульовано вхідні та вихідні потоки при дослідженнях трибосистем. Показано, що до вхідних потоків відносяться конструктивні параметри трибосистеми, технологічні параметри, експлуатаційні параметри. Перераховані параметри формують потік матерії, енергії та інформації, який є вхідним впливом на трибосистему. Вихідним потоком з трибосистеми є параметри: об'ємна швидкість зношування I, розмірність м^3/год; втрати на тертя, які оцінюються коефіцієнтом тертя f, безрозмірна величина. Вихідний потік є інформаційним потоком трибосистеми. При вирішенні контактних задач це дозволяє враховувати не тільки рівень напружень, а й швидкість поширення деформації в матеріалах поверхневих шарів, а також глибину поширення деформацій, що в моделях буде враховуватися об'ємом деформованого матеріалу.

В залежності від поставлених завдань і вимог до їх вирішення обґрунтовано застосування різних методичних підходів для моделювання. Показано, що застосування математичних моделей при моделюванні трибологічних процесів залежить від правильного вибору технічних обмежень, які визначають область існування оптимальних рішень.

Ключові слова: фулерени; розчинник фулеренів; фулеренові композиції; структура трибосистеми; швидкість роботи дисипації; електростатичне поле поверхні тертя; швидкість деформації; об'ємна швидкість зношування; коефіцієнт тертя

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Investigation of tribological characteristics of brake pairs elements of mobile machine

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Abstract

The subject of the experiments was the tribological properties of typical brake pads and disc characteristics. For the experiment was used Grey Cast Iron brake disc and semi metallic, low steel quantity and ceramic brake pads. The breaking process was imitated. The experiment was conducted at 0.75, 1.25 and 1.76 m/s sliding speed using 0.85 MPa contact pressure. The experiments lasted 10 minutes. The results of the experiments showed that best tribological characteristics have ceramic brake pads, despite the fact that brake disc temperature rapidly increase the with ceramic brake pads, but the friction coefficient (and braking torque) was the best. Semi metallic and low steel braking pads had very similar friction coefficient values, but wear and disc temperature values were more dissimilar.

Key words: brake system, pads, disc, material, coefficient of friction, wear.

Introduction

According to the data of the Department of Statistics of the Republic of Lithuania, the total number of vehicles is increasing by ~ 5 % annually, which affects traffic safety and environmental pollution [1]. More than 1.5 million vehicles are registered in Lithuania. The volumes of mobile construction and agricultural machinery are constantly increasing.

Vehicles pollute the environment not only with exhaust gases but also with wear products on the road surface, tires and brake system. Brake pads are one of the most wearing and polluting parts of a car [2, 3]. Brake system wear products can cause for as much as 90 % of vehicle pollution, and about 35 % of wear products are potentially released into the environment.

The main function of the braking system is to decelerate and / or stop the vehicle. When braking, the kinetic energy of the mobile machine is converted into heat. The intensity of heat release depends on the weight of the mobile machine, the speed of movement, the intensity of braking.

The brake system is one of the key safety features of a car [4]. During braking, there is friction, which results in active wear of the friction and disc surfaces of the brake pads [5]. When the brake discs and pads are heated, the coefficient of friction changes, which affects the stability of the braking system.

Literature review

The brake discs absorb most of the heat generated by the braking and their temperature rises from 20 to 700 °C in a few seconds. The material of the brake discs must be such as to withstand not only the normal mechanical load produced during braking but also the operating temperature. The following types of brake discs are distinguished according to their predominant composition [2].

1. Carbon fiber discs. These are discs with excellent thermal and mechanical properties, but due to their high cost they are not practical for everyday use (used in aviation, sports).

2. Aluminium alloy discs are extremely light, such as Aluminium-Copper alloy (Al-Cu), but their thermal conductivity is too low, so their use is limited.

3. Cast iron brake discs are most popular in mobile machines due to their good physical-mechanical properties, good price-quality ratio, such as Gray Cast Iron (GCI).

The main requirements for "friction-disc" brake friction pairs are: constant friction coefficient, low wear rate (possibly lower particulate emissions), quiet operation [6]. Brake element manufacturers do not provide material information (except for an indication of the material group to which the product belongs). The manufacturer shall specify only the vehicle model for which this item is intended.

Friction material is the most important element influencing the performance of the brake friction pair. Brake pads are divided into Organic / Low Steel Quantity / Non Asbestos Organic, Semi-Metallic and Ceramic according to the composition of the friction material [3]. There are many requirements for friction material. It must have the following characteristics [4]: stable coefficient of friction (little influence on speed and temperature); high operating temperature and resistance to abrasion, cracks; tribologically matched to metal parts; work quietly (without vibrations); adequate resistance to compressive and shear loads and to water, oil, salts or dirt; "Environmentally friendly"; inexpensive and technological.

The low metal quantity brake lining contains 5 - 35 % non-ferrous metals. The coefficient of friction is in the range of 0.38 - 0.50. They are ideal for high speed operation [7]. Organic brake pads are made of a mixture of common materials like rubber, carbon, glass / fiberglass and others, binded by resin.

Semi-metallic models consist of between 30 and 65 % metal.

Ceramic brake pads are free of metals and are made of metal oxides and carbides [8]. The coefficient of friction is in the range of 0.33 - 0.40 [7]. The blocks are made of ceramic fibers and fillers of a similar type, these blocks are more wear resistant, creates less noise and last longer. An important drawback is that it only works effectively when warmed up to operating temperature.

The actual contact loads in passenger car brake systems are up to 150 N / cm^2 [9].

The composition, microstructure and wear parameters of brake discs are analyzed in the work [10]. The microstructure and composition of gray cast iron are shown in Figure 1.

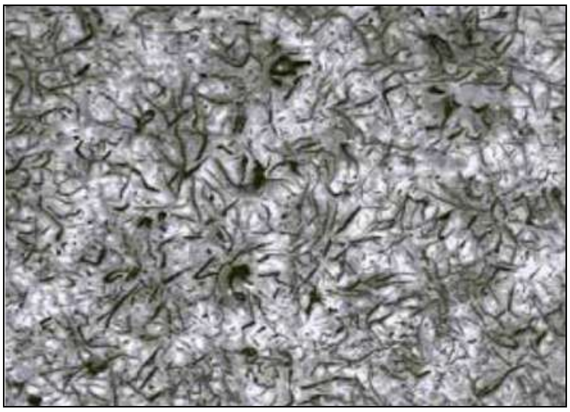

Fig. 1. Typical microstructure of gray cast iron disc brake (composition C 3.54%, Si 2.08% Mn 0.856% S 0.14% P 0.18%) [10]

Purpose

The aim of the study was to investigate the tribological characteristics of friction pairs (brake disc - brake pad) - the influence of different friction pad materials and load on brake disc heating, friction coefficient and wear.

Materials and Methods

Experimental research was performed in the Tribology Laboratory of the Department of Mechanical, Energy and Biotechnology Engineering of Vytautas Magnus University.

Three brake pads of different materials were selected for the study: Semi metallic (hereinafter No. 1), Low Steel Quantity (hereinafter No. 2) and Ceramic (hereinafter No. 3) and cast-iron brake discs. The latter is made of AUDI A8 original cast iron brake discs (Gray Cast Iron). The dimensions of the discs are ø65 × ø16 × 15 mm (Fig. 2). The weight of the discs is ~ 275 grams.

Laboratory tests of friction pairs were performed on a modernized tribological research machine СМЦ-2, in which two friction pads are symmetrically mounted symmetrically compress to the brake disc. This prevents the brake disc from bending and the axial load on the friction machine shaft.

The coefficient of friction μ is calculated by the expression:

$$\mu = M / (2 \cdot F_2 \cdot r_{average}). \quad (1)$$

here $r_{average}$ – distance from the centre of the brake lining to the axis of rotation of the brake disc, m;

F_2 – force of pressure of friction blocks to the disc, N;

M – is the registered braking torque of the test machine, Nm.

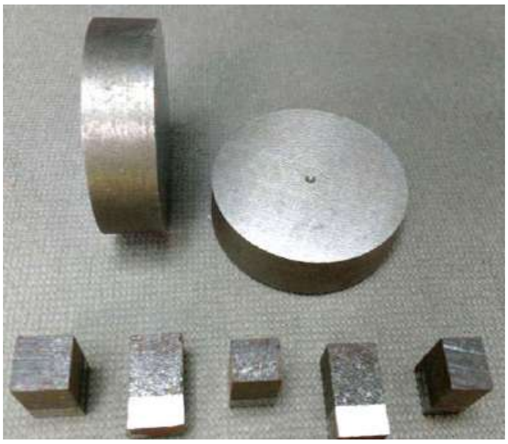

Fig. 2. Friction pads 14 × 14 × 20 mm (cut from brake pads) and gray cast iron blanks (ø70 × 15 mm discs) for the manufacture of brake discs

The difference in friction surface areas was up to 3% (block No. 1 area 196 mm^2, No. 2 - 189 mm^2, No. 3 - 189 mm^2). The weight of the blocks is ~ 12 grams. To determine the wear of the friction pairs, the discs and pads were weighed before and after the test with a KERN EG420 scale to the nearest 1 mg. The tests were repeated three times.

The tests were performed at sliding speeds of 0.75, 1.25 and 1.76 m/s. A contact load of 0.85 MPa brake lining was used in the study. The temperature of the brake disc was measured with a non-contact thermometer G900IR before and during the test (measured every 2 min intervals). The test duration of test was 10 min, chosen due to the relatively low mass of the disc (heat capacity).

Results

The wear results of the brake discs and pads are shown in the diagram (Figure 3). The higher the speed at which the brake pads of different materials operate, the greater and more uniform their wear (from 0.093 to 0.103 g at a sliding speed of 1.76 m/s). Ceramic blocks wear loss at 0.75 m/s is lower by 47% compared to semi metallic blocks, but wear loss at 1176 m/s and higher by 11 %. The main advantage of ceramic blocks is seen at a sliding speed of 1.25 m/s, at which these blocks wear loss is 0.045 g, and blocks of other materials varied in the range of 0.082–0.095 g.

At all working speeds, cast iron brake discs used with ceramic pads showed greater wear resistance (up to 3.8 times less) than discs used with pads of other tested materials (Fig. 3).

The temperature change of the brake discs during the test is shown in Figure 4.

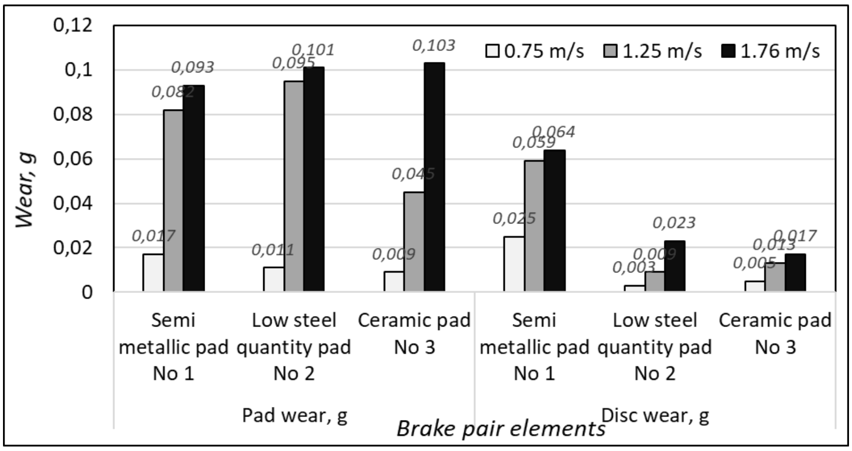

Fig. 3. Brake pad and disc wear during test (0.85 MPa contact pressure)

The slowest increasing of the temperature appeared in the friction pair of brake discs with semi-metallic (No. 1) and low steel quantity (No. 2) blocks. The fastest increasing of the temperature appeared where brake disc was in the pair with ceramic pads No. 3 (Fig. 4).

The temperature difference of the discs used with ceramic and semi-metal and low steel quantity pads becomes apparent after 4 min of operation. Meanwhile, the temperature difference of discs working with semi-metal and low-metal blocks becomes noticeable after 6 min of sliding. During 10 min of sliding, the temperature of the discs tested with different friction materials varied up to 40 ^{o}C (Fig. 4).

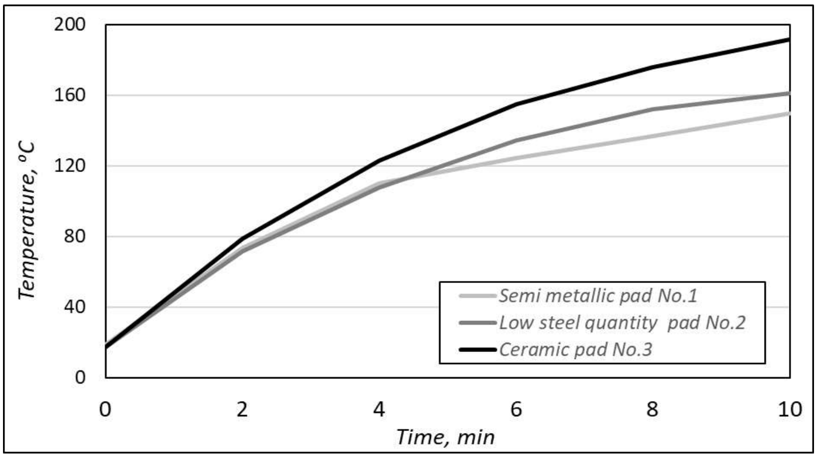

Fig. 4 Temperature of the brake discs during the testing of the pads of varied materials (sliding speed 1.75 m/s and contact pressure 0.85 MPa)

During the tests, the fixed braking torque of the friction machine was converted to the coefficient of friction according to formula (1). The coefficient of friction variation shown in Fig. 5.

The maximum values of the coefficient of friction (braking torque) are 0.345–0.435, recorded when the brake disc is working with ceramic brake pads No. 3 (Fig. 5). The results showed that semi metallic brake pads No. 1 worked unstable: the values of the coefficient of friction increased, and in the working range of 2.5–3.0 min, the coefficient of friction drastically reduce, occurred vibration. This phenomenon stabilized within one minute, but the work of the friction pair was not stable.

When the brake disc is working with low steel quantity pads No. 2, the value of the coefficient of friction increased to 4 min. The values of the friction coefficients of all investigated friction pairs started to decrease after 4–5 min of sliding time. The most likely cause is because of the increased temperature in the friction pair. The values of the coefficient of friction of low steel and semi metallic friction blocks are equal at the end of the test (0.31–0.32). Ceramic blocks have a higher coefficient of friction by 0.1 at the all testing time (Figure 5).

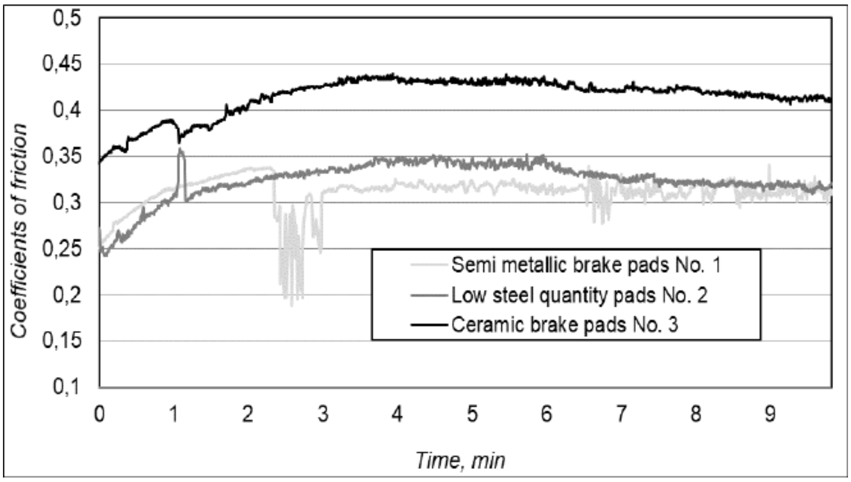

Fig. 5 Variation of the coefficient of friction during testing (1.75 m/s)

The average friction coefficients of the brake friction pairs are given in Table 1.

Table 1

Average values of coefficients of friction of brake friction pairs

Brake pad number	Average coefficient of friction at slip speed		
	0.75 m/s	1.26 m/s	1.76 m/s
Semi metallic pad (No. 1)	0.23	0.33	0.31
Low steel quantity pad (No. 2)	0.27	0.32	0.32
Ceramic pad (No. 3)	0.34	0.41	0.41

A significant increase in the coefficient of friction of all friction pairs is seen with an increase of sliding speed from 0.75 to 1.26 m/s (from 18.5 to 43 %). Increasing the sliding speed in the friction pairs (No.1 and No. 3) from 1.26 to 1.76 m/s, the average coefficients of friction did not change, while friction pairs with semi metallic pads (No. 2) has lower coefficient of friction by 6.1%.

Conclusions

As a result of our research, can be concluded:

1. Increasing the working speed of friction pairs from 0.75 to 1.25 m/s increases the wear of blocks by 5.0-8.6 times; the biggest advantage of the ceramic block is at a sliding speed of 1.25 m/s due to 2 times lower wear loss;
2. The semi metallic pads (No. 1) operated brake disc wears most intensively, while the disc worked with ceramic pads (No. 3) showed the lowest wear loss result;
3. The analysis of the dependences of the coefficient of friction and the temperature of the discs shows that with the increase of the temperature of working surface to 105–120 °C, the coefficient of friction increases, but when these values are exceeded, it starts to decrease;
4. The highest temperature of brake disc appear when working with ceramic brake pads, as well as the highest coefficient of friction is achieved when the disc worked with ceramic brake pads;
5. Ceramic brake pads provide about 0.1 higher coefficient of friction (0.41) compared to semi metallic and low steel quantity pads and gray cast iron brake disc (coefficients of friction in the range of 0.31-0.33).

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Influence of the parameters of hydrogen nitrogen nitrogen in a glow discharge on tribological and physico-chemical properties of steel 40X

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Abstract

One of the modern and effective methods of hardening metals is nitriding in a glow discharge in ammonia or in an anhydrous medium (nitrogen + argon) - BATR. This paper presents the results of experimental studies comparing the results of tribological and physicochemical properties of hardened surfaces obtained by nitriding with autonomous and interconnected BATR modes. The complex of traditionally fixed values of operating parameters (temperature, composition of the gas mixture, pressure and saturation time) without taking into account energy characteristics (voltage, current density and specific discharge power) significantly reduces the technological capabilities of BATR to achieve the necessary physicochemical properties of metal surfaces specified by conditions exploitation. Taking into account the energy characteristics of BATR, a significant reduction in the energy consumption of the nitriding process is achieved. The energy levels of the main subprocesses are significantly different: the formation of nitrides occurs at low energies, surface sputtering occurs at high voltage values, and nitrogen diffusion occurs at increased current density values. In cases where the energy of the flow is insufficient, either a glow discharge may not occur at all, or with a lack of voltage, the nitride ball on the surface is not sprayed and it acts as a barrier that prevents the diffusion process into the inner layers of the metal, which leads to low physicochemical and, correspondingly, tribological indicators of nitrided balls. The quantitative ratio between them and the required operational properties of the metal, respectively, can be achieved only through an independent combination of the energy and operating characteristics of BATR.

Key words: non-hydrogen nitriding in a glow discharge (BATR); wear resistance of nitrided balls; phase composition; voltage; current density; specific power of the discharge.

Introduction

One of the modern and effective methods of hardening metals is nitriding in a glow discharge in ammonia or in an anhydrous medium (nitrogen + argon) - BATR. Nitriding in a hydrogen-free environment excludes the possibility of explosion of the installation and hydrogen embrittlement of surfaces due to the diffusion of hydrogen formed during the decomposition of ammonia into the depth of the metal. In addition, the BATR process is absolutely environmentally friendly. Also, at present, most studies are devoted to nitriding processes with interdependent parameters, i.e., each combination of operating parameters (temperature, composition of the gas mixture and its pressure) automatically corresponds to a combination of energy (voltage, current density), as a result of which the latter are not considered as controlled. process factors [1].

Among all the operating characteristics of the process, the most significant parameter is the preset surface temperature, since to achieve and maintain it throughout the entire nitriding time, a certain specific combination of voltage and current density is required. Providing a given surface temperature due to factors alternative to a glow discharge allows changing the energy parameters of BATR in a wider range. In world practice, ensuring the independence of temperature from the energy characteristics of BATR is achieved by using chambers with "hot walls" [2]. Providing a given surface temperature due to factors alternative to a glow discharge allows not only to reduce the nitriding time, but also opens up fundamentally new possibilities for improving the controllability of the BATR process and obtaining the properties of surfaces of metallic materials specified by operating conditions, depending on the conditions of their operation. Despite the obvious new capabilities of BATR with independent energy parameters, it has not actually been investigated in practical and experimental terms. This paper presents the results of experimental studies comparing the results of tribological and physicochemical properties of hardened surfaces obtained by nitriding with autonomous and interconnected BATR modes.

Purpose of work – study of the influence of energy parameters on the tribological and physicochemical properties of the diffusion layers of steel 40X at BATR with interdependent and autonomous saturation parameters.

Materials and methods of testing

Nitriding was carried out on the UATR-1 unit designed and manufactured at the Podolsk Scientific Physics and Technology Center (PNFTTs) of the Khmelnitsky National University. The installation belongs to a diode-type model with direct current and was additionally equipped with heating elements placed in the gas-discharge chamber, which made it possible to arbitrarily change the voltage value U, and the value of the current density j was determined as the ratio of the current strength to the total area of the parts (samples) of the cage and suspension ($j = I/S$, A/m^2). The design features of the installation for carrying out BATR processes with interdependent and autonomous nitriding modes are described in detail in [3].

Nitriding modes

The influence on the structure, phase composition and, accordingly, on the physicochemical properties of the surface nitrided layers of temperature, composition of the gaseous medium, its pressure and saturation time has been comprehensively studied, for example, in [1 ... 4]. Therefore, saturation was carried out in a mixture of 80 % N_2 (nitrogen) and 20 % Ar (argon) at a temperature of T = 833 K for an hour. The voltage and current strength were chosen arbitrarily, based on the experience of preliminary studies. Technological modes of carrying out BATR processes are given in Table 1.

Table 1

Characteristics of BATR modes

Modes	Experiment serial number								
	1*/1**	2	3	4*/4**	5	6	7*/7**	8	9
Pressure p, torr (Pa)	0,4 (53,2)			0,8 (106,4)			1,2 (159,6)		
Voltage U, V	1100/680	820	515	840/610	515	300	700/540	515	300
Current density j, A / m2 ($j = I/S$)	11/15,3	7,2	3,2	13,2/16,4	7,2	2,8	15,8/17,2	12,8	7,2
Specific power of the glow discharge w, kW / m2	12,1/10,4	5,9	1,65	11,1/10,0	3,7	0,84	11,1/9,3	6,6	2,2

* – modes with interdependent parameters; ** – also, but with a modified suspension shape

The specific power of the glow discharge was determined by the formula:

$$W = UI / S = Uj ,$$

where S – is the total area of parts (samples) and suspension (cathode area).

Experiments 1*, 4*, 7* were carried out under interdependent nitriding regimes (the first series of experiments), i.e. each pressure of the mixture corresponded to the corresponding value of voltage and current density. The second series of experiments (1**, 4**, 7**) was carried out under the same conditions but with a changed shape of the suspension and, therefore, with a different cathode surface area. Experiments 2, 3, 5, 6, 8, and 9 correspond to the autonomous modes of the BATR. At the same time, modes 3, 5 and 8 were carried out at $U = 515B = \text{const}$, and modes 2, 5 and 9 at $j = 7,2\text{A/m}^2 = \text{const}$.

Metallographic studies were carried out on a MIM-10 microscope after etching microsections with a 3 % alcohol solution of nitric acid. The microhardness was determined with a PMT-3 microhardness tester at a loading of 0.98N with fixing the hardness values on the surface and at a distance of 25, 50, 100, 200, 500 μm.

For X-ray phase analysis, a DRON-3 diffractometer was used in the range of angles θ from 20º to 100º with a scanning step 0,1º and an exposure time of 10 s.

Tribological studies of the samples for wear resistance were carried out on a universal machine model 2168UMT according to the disk-finger friction scheme; friction without lubricant (dry); counterbody material - steel ШХ15 with hardness HRC61; pressure in the contact zone $p = 16$ MPa; controlled linear wear h, which was determined by the change in the linear size of the sample measured along the normal to the friction surface after passing the section of the friction path L in accordance with the accepted measurement step l (Table 2).

Table 2

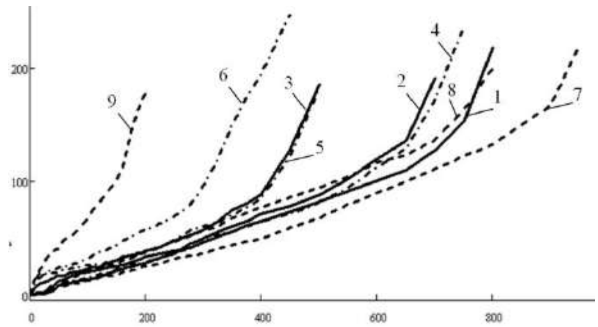
Frequency of measurement of the results of tribological studies

Friction path section, m	0 - 50	50 - 200	200 - 400	400 - 1000	more 1000
Measurement step, m	5	10	25	50	100

Research results and their discussion

The dependence of the wear resistance of the modified surfaces on the energy parameters of the discharge was confirmed by tribological studies. As a result of the experiments, it was found that under dry friction conditions for surfaces modified at higher energy values, the wear rate (Fig. 1, a) and the running-in period (Fig. 1, b) decrease, and the period of stable wear increases, and, with an increase in the content of alloying elements in steel, this pattern becomes more pronounced.

Of course, under other experimental conditions (for example, when studying the obtained samples for corrosion resistance), the dependence of the tribological properties of nitrided surfaces on the current density and voltage on the chamber electrodes may acquire a different character, but the very existence of such a dependence is beyond doubt, which refutes the provisions given in [5], according to which the energy characteristics of the discharge do not have a significant effect on the results of ATR.



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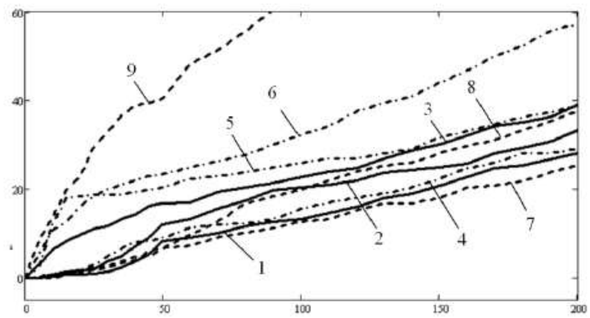


b

Fig. 1, a – wear curves of steel;
b – running-in area of wear curves of nitrided steel 40 Kh

The data presented allow us to draw a completely obvious conclusion about the effect of voltage and current density on the characteristics of the modified layer determine the wear resistance of nitrided steels in a glow discharge - but not just essential, but decisive. Moreover, in the field of the energy parameters of the regime, there is a certain limit below which the ATR process generally loses its meaning, since it leads to unacceptable results, and this is despite the fact that the values of the regime remain constant.

This means, in particular, that the set of traditionally fixed operating parameters (temperature, pressure, composition of the gas mixture and the duration of the process) does not give an unambiguous idea of the conditions for carrying out the ATR process, and therefore cannot serve as a basis for predicting its results.

This fact has been repeatedly confirmed when technological processes carried out at similar values of operating parameters, but in different installations (or even in one installation, but using a different suspension), led to completely different results.

This is due to the fact that the factors that determine the effectiveness of the nitriding process in a glow discharge are not only the parameters of the mode, but also such indicators as the cathode distance, the shape and size of the suspension and the test sample (or part), the presence of local exceptions on its surface, and many other factors.

The list of factors that are decisive for ATP, given by the American researcher David Pye in [6], including thirteen items.

Of course, direct consideration of all these indicators would lead to incredible complications in the management of the APR process. However, they can be taken into account indirectly, since the influence of all the listed factors reflect the energy parameters of the process.

It is known that the depth and phase composition of the nitride zone of the surface layer of steel determine its physicochemical properties. Thus, the nitride zone, which consists only of a phase, is characterized by a rather high plasticity, and the zone containing the phase has a lower plasticity, but a higher corrosion resistance [7, 8, 9].

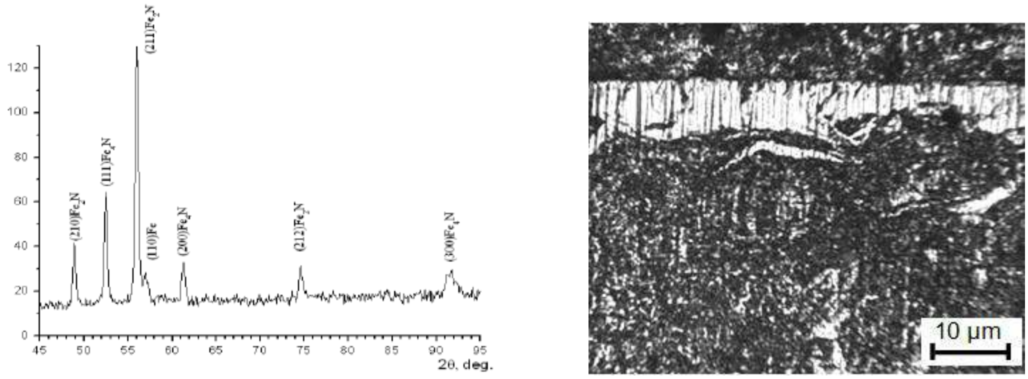
The layer without a nitride zone has the highest plasticity [9, 10]. In general, the thinner the nitride zone, the more plastic the nitrided layer, but the lower its resistance to abrasive and corrosion-mechanical wear.

Thus, according to [11], for parts that are operated in corrosive environments and for wear at low contact voltages, the BATR should be carried out at the maximum possible voltage and current density, which contributes to the formation of a phase and, accordingly, we have a high corrosion resistance, as well as good running-in of friction surfaces. In this case, in order to avoid the transition of a glow discharge to an electrospark one, the condition $w < w_{KR}$ should be observed, i.e. the specific power of a glow discharge should not exceed the critical specific power of the occurrence of an electric arc discharge.

A decrease in voltage and current density leads to an increase in the phase fraction, which contributes to an increase in the fatigue strength of parts operating under conditions of corrosion-mechanical wear (CMC) in corrosive environments [12].

The dependence of the wear resistance of the nitrided surfaces of steel 40Kh on the energy parameters of the discharge is confirmed by the results of testing samples with dry friction (Fig. 1, a, b) nitrided according to modes 6, 9, 3, and 5, which were nitrided at low values of U and j. in wear resistance than samples nitrided at higher values of U and j (7*, 1*, 8, 4* and 2, see Table 1). These samples (6, 9, 3, and 5) are characterized by small running-in periods at a constant wear rate and a high wear rate, determined by the tangent of the tangent to a point in the wear curve section.

The data of X-ray structural analyzes $\varepsilon, \gamma^{/}$ – (Fig. 2) and microdurometric measurements (Table 3) confirm and explain the results of tribological experiments (Fig. 1, a, b). So, at BATR according to mode 7* (Table 1), a nitride ball of the greatest thickness (Table 3), consisting of phases (Fig. 2, a), is formed on the surface. A decrease in the values of U and j (Table 1) for mode 8 contributes to a decrease in both the thickness of the nitride zone hN (Fig. 2, b) and the thickness of the diffusion layer h (Table 3), which were identified by measuring the microhardness along the depth of the layer. The phase composition of the nitride and diffusion layers remains practically unchanged.



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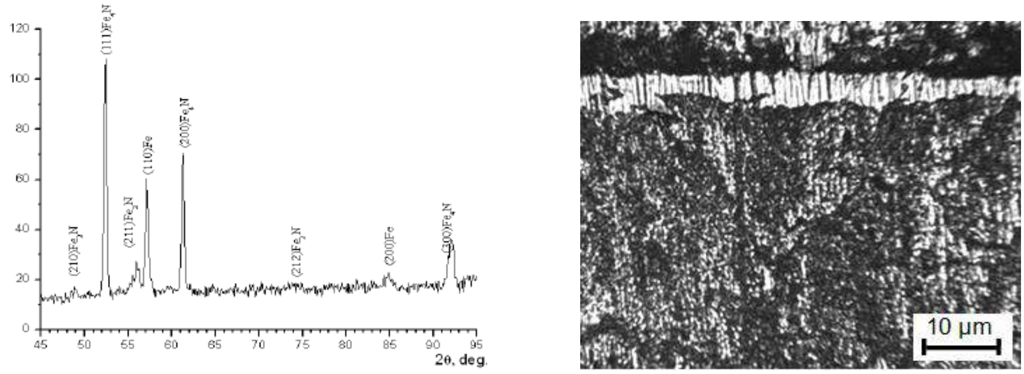

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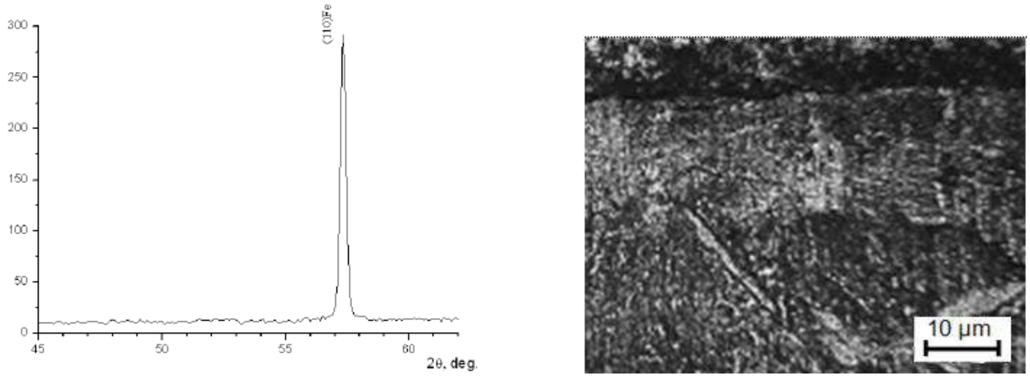


c

Fig. 2. Diffraction patterns and microsections of 40X steel samples:
a – nitriding according to mode 7;
b – according to mode 5;
c – according to mode 9

Table 3

Thickness of nitride hN, diffusion layers h and surface microhardness HV0.1

№ regime	1*	2	3	4*	5	6	7*	8	9
hN micron	5,88	2,42	0	3,64	3,2	0	8,76	6,6	0
h, micron	400	300	75	300	200	0	300	250	0
HV0,1	796	676	412	647	444	230	842	625	238

Nitriding according to mode 9 corresponds to the case when $w << w_{KP}$, i.e. the specific power of the discharge is much less than the required one and the glow discharge does not occur and, accordingly, $h = hN = 0$ (Fig. 2, c). We have similar cases when nitriding in modes 6, 3 and partly 5, which have low wear resistance.

Thus, it has been established that under conditions of dry friction for surfaces of steel 40Kh nitrided at high energy values, the wear rate and the running-in period decrease (Fig. 1), the latter refutes the statement of K. Keller [13] that the energy characteristics of BATR do not have a significant influence on the results of nitriding. On the contrary, the above BATR results allow one to draw completely obvious conclusions about the significant effect of voltage and current density on the physicochemical and tribological properties of nitrided layers. In addition, there is a certain limit of energy parameters below which (modes 9, 6, 3) carrying out BATR processes does not make sense at all, since it leads to negative results, and this despite the fact that the values of operating characteristics (temperature, composition of the gas mixture, its pressure, etc.) nitriding time) remain unchanged. Thus, the set of traditionally fixed operating parameters of the BATR cannot unambiguously characterize the conditions of the nitriding process and be the main one for predicting its results. This fact is confirmed by the fact that at similar values of the operating parameters, the change of the suspension led to completely different results of nitriding, which is explained by the change in the energy characteristics of the BATR process (Fig. 3).

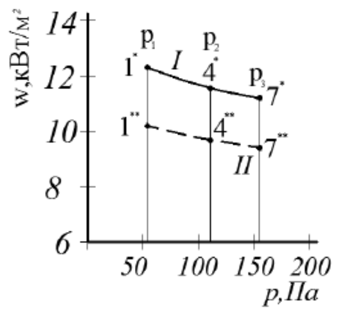


Fig. 3. Dependences of the specific power of the discharge w on the pressure of the gas mixture at interdependent BATR modes: I – modes 1*, 4* and 7*; II – also when changing the shape of the suspension (modes 1, 4** and 7**)**

In addition to comparing the physicochemical and tribological properties of nitrided balls of steel 40Kh at BATR with interdependent and autonomous energy modes of processes, the purpose of the work was also to check the effect of the specific power of the glow discharge w, which in [5] was called the plasma energy density (PEP) and where it is stated, that the dependence of the pressure of the gas mixture on the specific power of the discharge has an extreme character: i.e. the pressure of the gas mixture, which corresponds to the maximum w, ensures the production of a nitrided sphere of the greatest depth [5]. However, studies [3] and the results of our experiments (Fig. 3) do not confirm the existence of such an extreme dependence.

Changing the shape of the suspension led to a change in the values of w (curves I and II in Fig. 3), and in the second case, we have significantly lower energy costs for the nitriding process. However, this is not the main advantage of nitriding, but the main factor is the physicochemical properties of the hardened layers, which provide the specified performance characteristics of the hardened surfaces of parts. So, in works [6, 7, 11, 12] doubts are expressed about the legitimacy of the use of specific power as the only energy criterion due to the possibility of arbitrary combination of its values with different values of voltage and current density at a constant pressure of the gas mixture. The studies given by us have shown that it is more expedient to assess the changes in voltage and current density. The specific power of the discharge can only serve as an estimate of the transition from a "dark" to a glow or to an electric arc discharge.

The study of current-voltage characteristics of BATR with interdependent (without heating) parameters showed their significant difference when changing the shape and size of the suspension (Fig. 4).

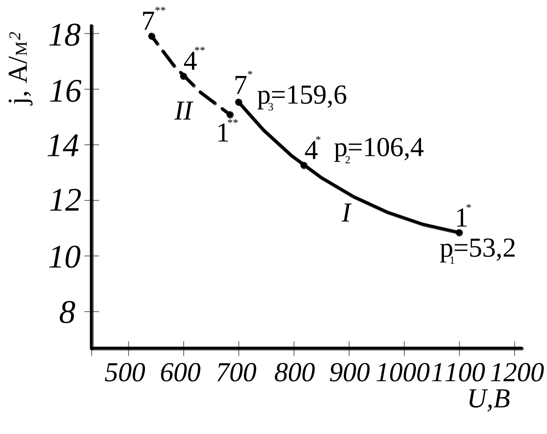


Fig. 4. Current-voltage characteristics of the BATR process at autonomous (interdependent) saturation modes (I and II are different forms of suspensions)

So the change of the suspension led to a shift of the current-voltage characteristic of the process to the left: a decrease in voltage values led to an increase in the current density with a simultaneous decrease in w (Fig. 3). At the same time, the operating parameters (pressure, temperature, mixture composition and nitriding time), as indicated above, remained unchanged. An increase in the mixture pressure also leads to a decrease in voltage with a simultaneous increase in the current density (Fig. 4). Since the increase in the value of the absolute value of the current is much less in comparison with the absolute value of the voltage (Table 1), the result is a decrease in the value of the specific power of the energy flow W (Fig. 3).

According to [3, 5], at a constant composition of the gas mixture in the pressure range of 100 - 120 Pa, a minimum of the active power of the discharge is observed. In our experiments, this dependence is not traced

(Fig. 3). Obviously, due to the fixation of not the active power of the energy flow acting on the metal surface and determined by the speed of electrons, but the electric power $(W = Uj)$, which is 35 % - 300 % [3], may be less active.

According to the energy theory, the following main subprocesses occur during BATR: the formation of nitrides, sputtering of the surface, and diffusion of nitrogen into the depth of the metal [3]. The energy conditions for the main subprocesses are significantly different. Thus, the formation of nitrides occurs at low energies and, on the contrary, the surface sputtering process is activated at high voltage values. The higher the current density, the more intensively the diffusion of nitrogen sputtered on the surface occurs deep into the metal with the formation of nitride and diffusion layers [3, 6, 11]. Each of the above processes is primarily implemented in those cases that are energetically most favorable under the given conditions [3]. In our studies, with a constant composition of the gas mixture and its pressure for each batch of samples of a series of experiments, the nitriding time, the main factors influencing the results of BATR are the current-voltage characteristics of the process. In fig. 5 illustrates the results of the dependence of the depths of diffusion h and nitride balls hN on the values of voltage U and current density j under interdependent nitriding modes.

With an increase in the mixture pressure (points 1*, 4* and 7*), the voltage U automatically decreases and the depth of the diffusion layer h decreases (Fig. 5, a) With a changed shape of the suspension (points 1**, 4 ** and 7**) on the contrary, with increasing pressure the values of h increase (Fig. 5, a, curve II). However, at the same time, at a pressure p = 106.4 Pa (points 4** and 4*) there is a minimum of the depth of diffusion h and nitride layers hN (Fig. 3, a; curve II and curve I in Fig. 5, b). On the contrary, with an increase in the mixture pressure (1* / 1**, 4* / 4**, 7* / 7**), the depth of the nitride zone hN increases and, accordingly, the stress decreases (Fig. 5, b). The sharp difference between curves I and II, as well as the presence of minima on curves I and II at points 4* and 4** (Fig. 5, a, b; curve II) indicates the dependence of the BATR processes on the shape, configuration of the suspension, and placement on it. parts, the presence of sharp edges, grooves, holes and the like on the parts. The list of factors that influence the results of BATR is given in [6] and includes 13 items. The presence of a minimum on the curve I (point 4* in Fig. 3, b), in addition to the listed factors, can be justified by a minimum of active (consumed) power at a pressure of p = 106.4 Pa [3, 6].

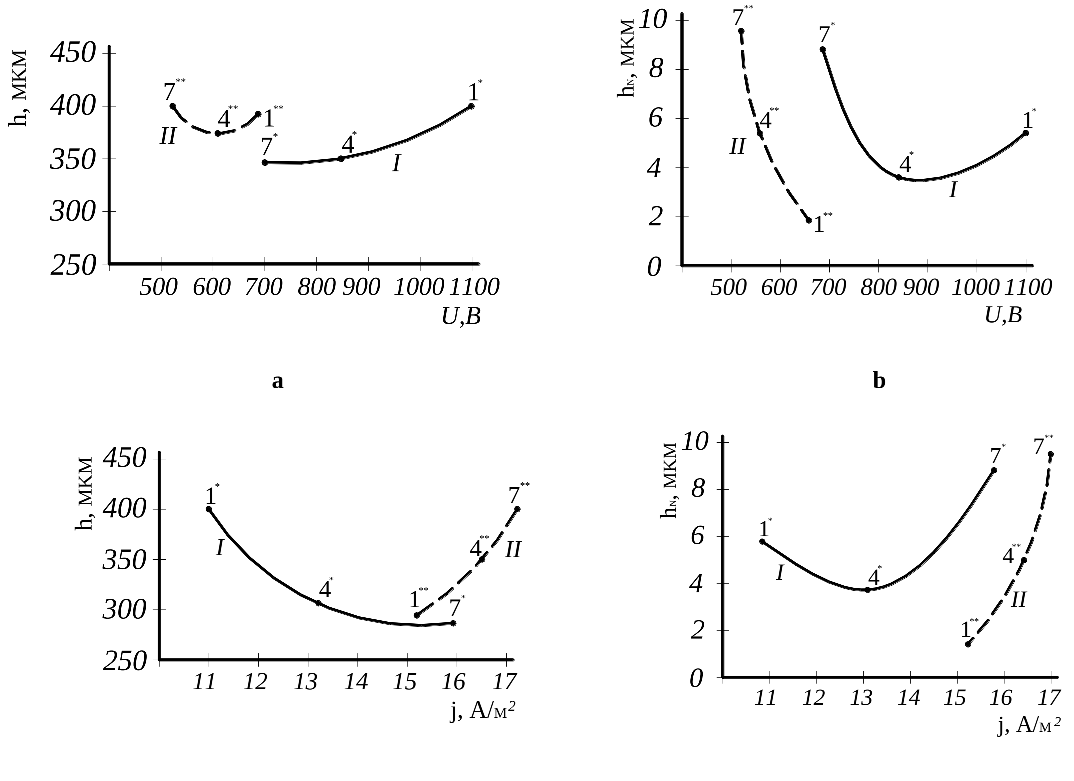


Fig. 5. Dependences of the depths of the diffusion sphere h (a) and the nitride zone (b) on the voltage value *U*; also (c, d) on the value of the current density *j* (I and II are different forms of suspensions) with the interdependent characteristics of the BATR

With an increase in the pressure of the gas mixture, the current density j increases and the value of h decreases (curve I in Fig. 5, c). At the same time, the depth of the nitride zone hN increases (curve I in Fig. 5, d). Here, too, at a pressure of p = 106.4 Pa, we have a minimum point (point 4*, curve I in Fig. 5, d).

With a change in the shape of the suspension, j increases with increasing pressure, and also increases, according to the dependences close to a straight line, h and hN (straight lines II in Fig. 5, c and d), which once again indicates the particular importance of taking into account the influence of additional factors on BATR results when designing a suspension, taking into account its shape and surface area, placing parts on the suspension, the presence of grooves, sharp edges on parts, etc.

In autonomous modes of BATR with increasing pressure at U = 515 V = const (points 3, 5 and 8 in Fig. 6), the specific discharge power increases, and at j = 7.2A / m2 = const (points 2, 5 and 9 in Fig. 6) decreases.

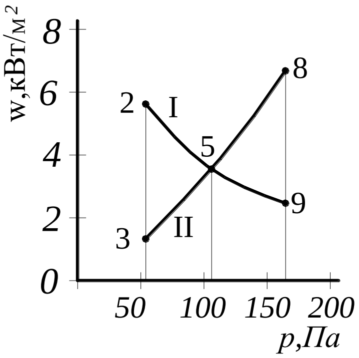


Fig. 6. Dependence of the specific power *w* on the pressure of the gas mixture p at BATR with autonomous saturation modes (curve I – nitriding at *U* = 515B = const and curve II – nitriding at *j* = 7.2A / m^2 = const)

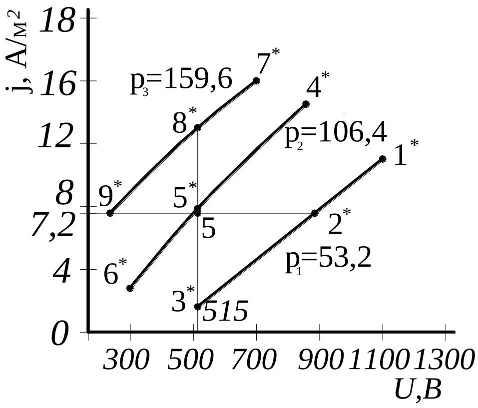
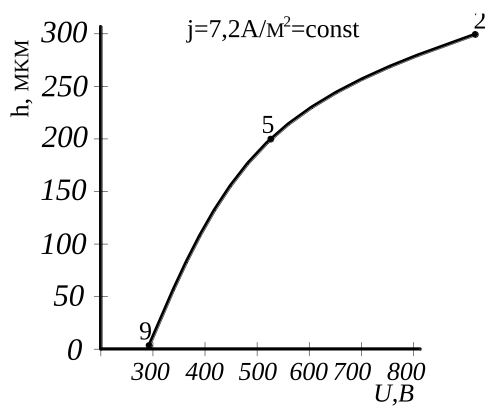


Fig. 7. Dependences of current-voltage characteristics of BATR on the pressure of the gas mixture

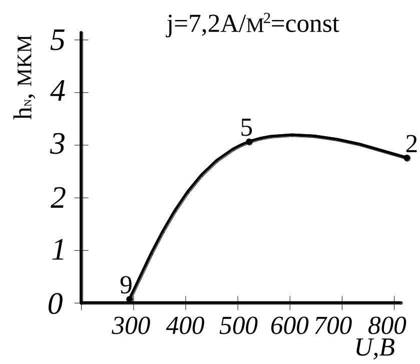
The results of X-ray structural analysis also indicate that the structure and phase composition of steel 40Kh depend on the energy characteristics of the discharge. The energy conditions for the main subprocesses (formation of nitrides, surface sputtering, and nitrogen diffusion), as mentioned above, differ significantly.

Consideration of the current-voltage characteristics shows (Fig. 7) that with increasing pressure p_1, p_2, p_3, the values of U and j increase. The indicated dependencies are straightforward. However, in all cases, the values of U and j for modes with interdependent parameters are higher than for stand-alone modes. In this case, with an increase in pressure, the voltage decreases, however, to ensure a glow discharge in the chamber, the value of the specific current density j rises automatically (points 1*, 4* and 7* in Fig. 7).

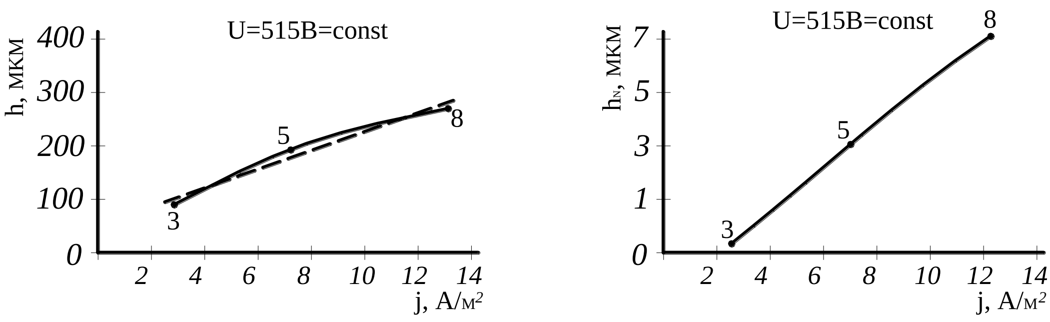
In the autonomous mode of the BATR, with decreasing pressure p, the values of U and j decrease, and at p = 159.6 Pa (point 9 in Fig. 8, a and b) $h = hN$ = 0. At a pressure p = 106.4 Pa U = 515 V, and j = 7.2 A/m^2 and, accordingly, we obtain the maximum value of hN (Fig. 8, b; point 5).



a



b



c **d**

Fig. 8 Dependences of the depth of the diffusion layer *h* (a) and the nitride layer *hN* (b) on the voltage at *j* = 7.2 A / m² = const; also c and d on the current density *j* at *U* = 515B = const

In the case of a fixed value U = 515 V, with an increase in pressure and current density j, the values of h and hN increase, practically along a straight-line relationship (Fig. 8, c and d). In this case, at p = 53.2 Pa and j = 3.2 A/m^2, there is no nitride layer at all, hN = 0 (point 3 in Fig. 8, d). The latter is explained by the low value of w = 1.65 kW/m^2 and, as a consequence, the absence of a glow discharge.

The ratio of the intensity of the main subprocesses determines the structure and phase composition of the nitrided layers. Depending on the flow combination of energy parameters, the formation of the nitrided layer can occur in different directions. The intensity of the formation of one or another phase can be different not only in magnitude, but also behind the sign, since the previously formed phase can disappear. Thus, the nitride phase hN, due to further diffusion of nitrogen into the depth of the metal, disappears and a zone of internal nitriding is formed - a nitrogenous solid solution Me [N] (mode 9, a and b), or the nitride layer consists of a small amount of the (Fe4N) phase (mode 3 on Fig. 2, c and d).

According to the results of X-ray structural analysis, at the maximum values of the energy parameters (all modes with interdependent parameters 1*/1**, 4*/4**, 7*/7**, mode 8 and partly 5 with autonomous saturation parameters), a nitrided layer is formed, which consists of (Fe2N), (Fe4N) and (Me [N]) - phases (Fig. 2, mode 7). A decrease in voltage and current density leads to an increase in the phase fraction in the nitride zone of the nitrided layer and, accordingly, to a decrease in the phase (Fig. 2, modes 7, 8). At minimum values of the energy characteristics, a nitride layer does not form on the surface and it consists only of a phase (Fig. 2, mode 9).

Thus, it has been established that under conditions of dry friction for the surfaces of 40Kh steel nitrided at high energy indicators, the wear rate and the running-in period decrease (Fig. 1, b), the latter refutes the statement of K. Keller [13] that the energy characteristics of BATR do not have significant influence on the results of nitriding. On the contrary, the above BATR results allow one to draw completely obvious conclusions about the significant effect of voltage and current density on the physicochemical and tribological properties of nitrided layers. In addition, there is a certain limit of energy parameters below which (modes 9, 6, 3) carrying out BATR processes does not make sense at all, since it leads to negative results, and this despite the fact that the values of operating characteristics (temperature, composition of the gas mixture, its pressure, etc.) nitriding time) remain unchanged. Thus, the set of traditionally fixed operating parameters of the BATR cannot unambiguously characterize the conditions of the nitriding process and be the main one for predicting its results. This fact is also confirmed by the fact that at similar values of the operating parameters, the change of the suspension led to completely different results of nitriding, which is explained by the change in the energy characteristics of the BATR process (Fig. 3 ... 8).

Conclusion

1. The complex of traditionally fixed values of operating parameters (temperature, composition of the gas mixture, pressure and saturation time) without taking into account energy characteristics (voltage, current density and specific discharge power) significantly reduces the technological capabilities of BATR to achieve the necessary physicochemical properties of metal surfaces specified by conditions exploitation. Taking into account the energy characteristics of BATR, a significant reduction in the energy consumption of the nitriding process is achieved.

2. The energy levels of the main subprocesses are significantly different: the formation of nitrides occurs at low energies, surface sputtering occurs at high voltage values, and nitrogen diffusion occurs at increased current density values. In cases where the energy of the flow is insufficient, either a glow discharge may not occur at all, or with a lack of voltage, the nitride ball on the surface is not sprayed and it acts as a barrier that prevents the diffusion process into the inner layers of the metal, which leads to low physicochemical and, correspondingly, tribological indicators of nitrided balls.

3. The priority in the formation of this or that phase $(\varepsilon, \gamma' u \alpha)$, the quantitative ratio between them and the required operational properties of the metal, respectively, can be achieved only through an independent combination of the energy and operating characteristics of BATR.

Designations

BATR hydrogen-free nitriding in a glow discharge; p – pressure; U is the voltage; j is the current density; w is the specific power of the glow discharge; S is the total area of parts (samples) and suspension (cathode area); h is the depth of the diffusion layer; hN is the depth of the nitride zone; HV 0.1 – surface microhardness; KMZ – corrosion-mechanical wear; phases of the nitrided layer: (Fe2N), (Fe4N) and (Me [N]).

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Стечишина Н.М., Стечишин М.С., Мартинюк А.В., Лукьянюк М.В., Люховець В.В., Білик Ю.М. Вплив параметрів водню азоту азоту у тліючому розряді на трибологічні та фізико-хімічні властивості сталі 40Х.

Одним із сучасних та ефективних методів зміцнення металів є азотування у тліючому розряді в аміаку або в безводному середовищі (азот + аргон) - БАТР. У цій статті представлені результати експериментальних досліджень, у яких порівнюються результати трибологічних та фізико -хімічних властивостей затверділих поверхонь, отримані азотуванням з автономними та взаємопов'язаними модами BATR. Комплекс традиційно фіксованих значень робочих параметрів (температура, склад газової суміші, тиск і час насичення) без урахування енергетичних характеристик (напруги, щільності струму та питомої потужності розряду) значно знижує технологічні можливості БАТР для досягнення необхідних фізико - хімічні властивості металевих поверхонь, визначені умовами експлуатації. Враховуючи енергетичні характеристики БАТР, досягається значне скорочення споживання енергії в процесі азотування. Рівні енергії основних підпроцесів істотно відрізняються: утворення нітридів відбувається при низьких енергіях, поверхневе розпилення відбувається при високих значеннях напруги, а дифузія азоту відбувається при збільшених значеннях щільності струму. У випадках, коли енергія потоку недостатня, або тліючий розряд може взагалі не виникати, або при нестачі напруги нітридна кулька на поверхні не розбризкується, і вона діє як бар’єр, що перешкоджає процесу дифузії в внутрішніх шарів металу, що призводить до низьких фізико -хімічних і, відповідно, трибологічних показників азотованих кульок. Кількісне співвідношення між ними та необхідними експлуатаційними властивостями металу відповідно може бути досягнуто лише шляхом незалежного поєднання енергетичних та робочих характеристик BATR.

Ключові слова: безводневе азотування у тліючому розряді (БАТР); зносостійкість азотованих кульок; фазовий склад; напруга; густина струму; питома потужність розряду.

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Development of methods for evaluation of lubrication properties of hydraulic aviation oils

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Abstract

A method for evaluation of the lubricating and rheological properties of hydraulic oils in tribological contacts has been developed, which consists in online studying samples of commercial batches of oils on a software and hardware complex with visual evaluation of the kinetics of changes in the main tribological indicators of friction contact. Using a roller analogy, the operation of gears in the conditions of rolling with 30% sliding is simulated. Samples of AMG-10 oil from two producers are analyzed. It is established that with increasing temperature of lubricant for Sample 2 ("Kvalitet-Avia" AMG-10), a long-term restoration of protective boundary films of oil is observed and the period of their formation increases by 2.5 times, causing the implementation of a semidry mode of lubrication at start-up. The total thickness of the lubricating layer is 1.27 times less as compared with Sample 1 ("Bora B" AMG-10 oil), regardless of the lubricant temperature. Also, the rheological properties of the oils have been determined. Sample 1 exhibits low shear stresses at the level of 9.4 MPa and high effective viscosity, 4249 and 5039 Pa·s, at a volumetric oil temperature of 20 and 100 ºC, respectively. For Sample 2, with increasing oil temperature to 100 ºC shear stress increases by 1.15 times and the effective viscosity in contact decreases by 1.53 times. Additives present in Sample 1 are characterized by more effective antiwear properties and thus increase the wear resistance of contact surfaces in the conditions of rolling with sliding thanks to strengthening of the surface metal layers during operation, while Sample 2 undergoes strengthening-softening processes which reduce the wear resistance of friction pairs.

Key words: aviation oils, lubricating layer, lubrication mode, effective viscosity, microhardness.

Introduction

Routes to increase the wear resistance of triboconjuction elements are based on modification of contact surfaces via selecting lubricant compositions and finding optimal operating modes for friction pairs. An important factor in ensuring a high efficiency of friction units is a high-quality choice of lubricants with high lubricating, antifriction and antiwear characteristics. Among various producers of commercial oil batches, it is important to choose a lubricant that meets not only required physical and chemical characteristics, but also has effective tribological properties. However, tribological indicators have not yet been standardized for a wide range of lubricants. Therefore, the development of methods for evaluating the quality of lubricants in tribological contact is an important area of research, the results of which can make it possible to provide valuable recommendations on oil workabiity in certain operation modes.

Literature review

The aviation hydraulic system is designed to control the mechanisms and systems responsible for flight safety. Its durability, operational survivability and reliability provide perfection of the design of units. Aviation hydraulic systems include forced hydraulic pumps. Ensuring their efficiency requires a high level of tribological properties of oils.

In particular, oils for aviation hydraulic systems must have an optimal level of viscosity, high viscosity-temperature properties in a wide temperature range and resistance to oxidation and foam formation. Oils must also have a sufficient level of tribological characteristics and be compatible with the structural and sealing materials of the components and units of the hydraulic system. A reduced viscosity of hydraulic oils causes the most intense manifestation of fatigue wear of the contacting parts of the hydraulic system. An increased viscosity significantly increases the mechanical losses of the drive, complicates the relative movement of pump parts and valves as well as makes it impossible for hydraulic systems to operate at low temperatures.

In [1], physicochemical and operational properties of optimized oil samples have been studied in an extended range of qualification test methods. The performance properties of new oils were compared with regular counterparts using a system of comparative evaluation of oil quality. All the results of computational and experimental studies of the properties of prototypes were included in the electronic database, on the basis of which valuable recommendations were developed for the introduction of new oils.

Many aircrafts use the hydraulic aviation oil AMG-10 in their hydraulic systems. Its lubricity is quite sufficient to prevent wear of hydraulic devices. To ensure stability over a long service life (2 - 3 years), unsaturated hydrocarbons are removed from the AMG-10 oil base and an antioxidant additive is added. The main disadvantage of AMG-10 oil is its low fire safety ("flash point in an open crucible" does not exceed 90 ^{0}C). In addition, during the operation of hydraulic systems with AMG-10 oil there observed a decrease in its viscosity due to the gradual destruction of the thickening additive. This leads to too rough operation of the mechanisms, oil overflow inside the hydraulic devices or to an external leak [2].

To improve the performance of oils for hydraulic systems, they are prepared from highly refined oil fractions with a viscosity index of not below 85 from low-sulfur and sulfur oils, which have undergone an acid-base or selective purification [3].

In [4], change in viscosity of AMG-10 oil in the temperature range 20 - 80 ºC at a constant flow rate gradient has been analyzed. This fluid was established to change its rheological properties: at a temperature below 50 ºC it is a Newtonian fluid, while at higher temperatures it becomes an Oswald de Ville fluid.

Analysis of publications on lubricating and rheological characteristics of oils for hydraulic systems has revealed that no comprehensive research in this direction has not been carried out yet. Therefore, study of the effective viscosity stability under mechanodynamic loads is of considerable interest.

Purpose

The purpose of the work is to develop a method for evaluating the lubricating and rheological properties of hydraulic oils in tribological contacts.

Objects of research and experimental conditions

Oils to be studied:

- sample 1 is oil "Bora B" AMG-10 according to TU U 19.2-38474081-010: 2016 with change 1 (produced by the LLC “Bora B”, Ukraine);
- sample 2 is oil AMG-10 according to GOST 6794-75 with changes 1 - 5 (produced by the LLC “NPP Kvalitet”, Russia).

Sample 1 was developed to organize work on avoiding oil import and overcome the critical dependence of the defense industry of Ukraine on import supplies of AMG-10 oil. The study of the samples was carried out on a software-hardware complex to evaluate the tribological characteristics of triboelements (Fig. 1), for which a special software had been developed for stepper motor control and online visual evaluation of the kinetics of changes in the main tribological parameters of tribocontact [5].

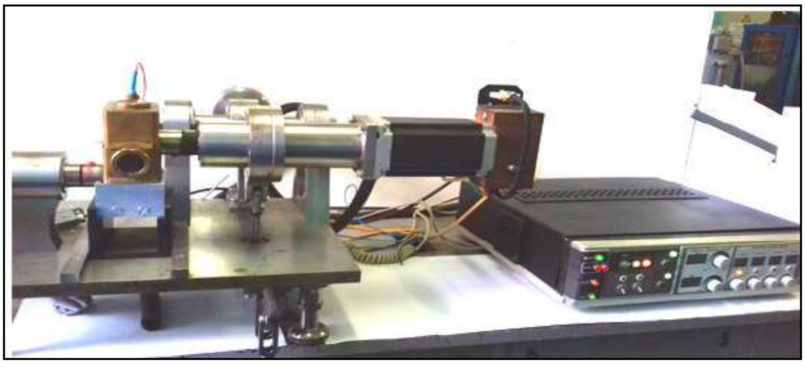

Fig. 1. Device for evaluation of tribological characteristics of triboelements

Work of gears in the conditions of rolling with sliding was modeled using the software-hardware complex by means of a roller analogy. Lubrication properties (hydrodynamic and non-hydrodynamic components of the lubricating film thickness) were determined by the method of voltage drop in the mode of normal glow discharge. Rheological characteristics of the lubricant (shear rate gradient, shear stress of lubricating layers, effective viscosity in contact) were evaluated by the kinetics of changes in the lubricating layer thickness, rotation speed of the leading and lagging surfaces and temperature of the lubricating layer.

Rollers (steel 30KhHSA, HRC 48 … 52, Ra 0.34 μm) were used as the material of contact surfaces. Lubrication of the contact surfaces was performed through immersing the lower roller in a bath of oil.

Testing was conducted in nonstationary conditions, which provide for the cyclicity of repetition in the start-up – stationary operation – braking – stop mode (Fig. 2). The total duration of the cycle was 80 s.

Fig. 2. Interface of subprogram for data processing during tribosystem operation in nonstationary friction conditions

Maximum rotation speed: 700 rpm for the leading surface and 500 rpm for the lagging surface. Sliding: 30%. Maximum contact load by Hertz: 200 MPa. Total number of cycles: 100. Temperature of oil: 20 ºC (cycles 1 - 45), rise to 100 ºC (cycles 46 - 50), 100 ºC (cycles 51 - 100).

Analysis of the main results

The investigated oil "Bora B" AMG-10 (Sample 1) is characterized by effective lubricating properties both during start-up and at the maximum rotation speed (Fig. 3). With increasing temperature in the tribological contact there is observed a decrease in the thickness of boundary adsorption layers due to changes in their nature: boundary layers of predominantly physical nature are replaced by boundary layers of chemical nature characterized by more effective antiwear properties. No failure of the lubricating layer during start-up and direct metal contact of the friction surfaces has been established. A semidry lubrication mode was only revealed for a short time, namely during the periods of running-in and initial temperature rise.

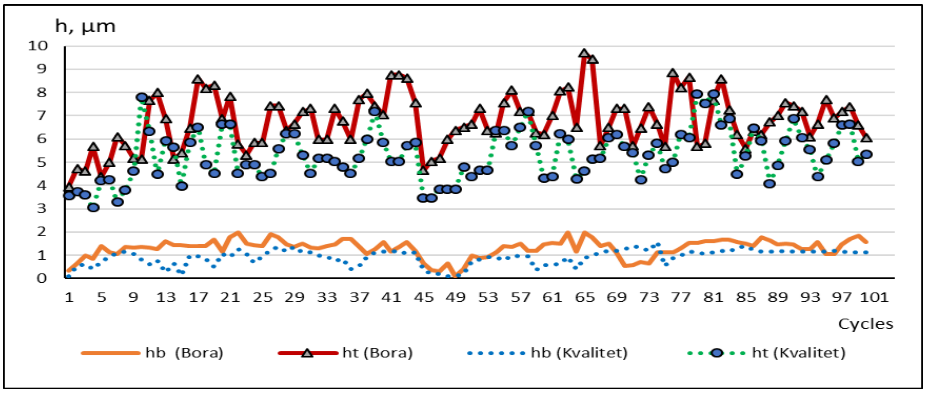

Fig. 3. The kinetics of change in the thickness of the boundary adsorption layers (h_b) and the total thickness of the lubricating layer (h_t) in the contact in the course of operation

At start-up, a mixed lubrication mode dominates regardless of the lubricant temperature, which indicates the effective starting properties of Sample 1 (Fig. 4).

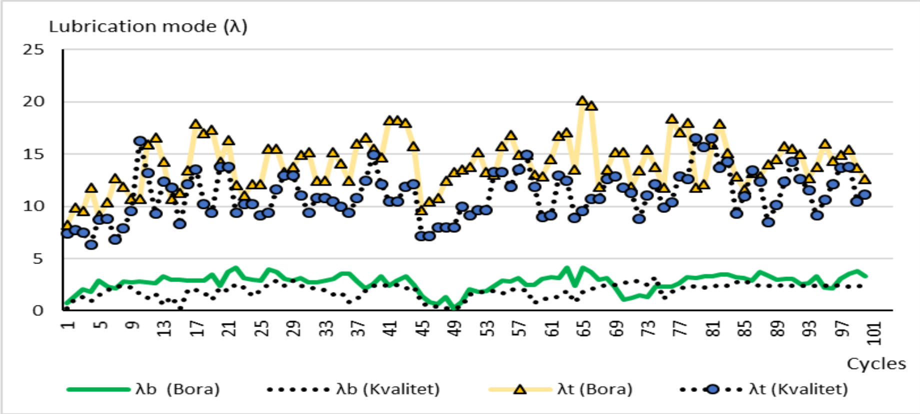

Fig. 4. The kinetics of change in lubrication modes in tribological contact (Classification of lubrication modes by λ: semidry (λ = 0 … 1); boundary (λ = 1 … 1.5); mixed (λ = 1.5 … 3); elastic-hydrodynamic (contact-hydrodynamic) (λ = 3 … 4); hydrodynamic (λ ≥ 4))

At maximum rotation speeds of the samples studied, the hydrodynamic mode of lubrication dominates, regardless of the oil temperature, which indicates effective separation of the contact surfaces due to formation of a lubricating layer.

Sample 2 (oil Kvalitet AMG-10) is characterized by effective lubricating properties at the maximum investigated rotation speeds, but during start-up its lubricating properties decrease (Fig. 3). At volumetric oil temperatures of 20 and 100 ^{0}C, the boundary adsorption layer thickness is 0.88 and 0.95 μm, respectively, which is, on average, 1.44 times less than that for Sample 1. This leads to deterioration of the lubrication mode in contact during start-up and to dominance of the boundary lubrication mode in 25 % of the operating cycles. With increasing lubricant temperature, a long recovery of the protective boundary oil films takes place, the time of their formation increases by 2.5 times as compared with Sample 1, thus causing the implementation of a semidry mode of lubrication at start-up.

At maximum rotation speeds of the samples, a hydrodynamic mode of lubrication dominates, regardless of the oil temperature, which indicates the effective separation of the contact surfaces due to the formation of a lubricating layer. The total lubricating layer thickness, which includes hydrodynamic and non-hydrodynamic components, is 1.27 times less compared with Sample 1, regardless of the lubricant temperature.

Let us analyze the kinetics of changes in the rheological characteristics of the oils studied.

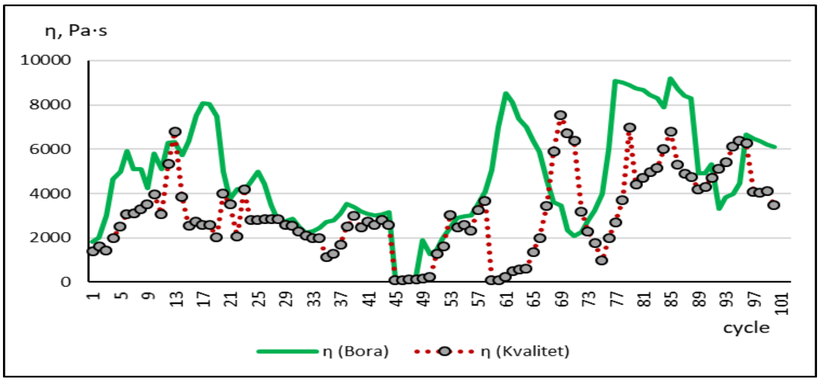

Fig. 5. The kinetics of change in the effective viscosity (η) of oils in the contact

The "Bora B" AMG-10 oil is characterized by effective rheological properties. Ensuring the hydrodynamic mode of lubrication at maximum speeds of the cycle, in the conditions of rolling with 30% sliding is due to the high bearing capacity of the lubricant and formation of hydrodynamic and non-hydrodynamic components of the lubricating layer thickness which are characterized by low shear stress, on average, 9.4 MPa regardless of oil temperature.

Despite the high shear rate gradients in the contact lubricating layer, from $5.63 \cdot 10^3$ to $5.73 \cdot 10^5$ s^{-1}, which occur at a maximum sliding speed of 0.71 m/s in the conditions of rolling with sliding, the lubricant is characterized by effective viscosity, on average, 4249 and 5039 Pa·s at a volumetric oil temperature of 20 and 100 ºC, respectively (Fig. 5). This indicates good resistance of the oil components to destruction under conditions of increasing shear rate gradient. The greatest reduction in effective viscosity to 105 … 250 Pa·s occurs in the conditions of initial increase in oil temperature (45 - 49 test cycles). This is due to change in the nature of the boundary adsorption layers characterized by effective adaptation in a wide range of temperatures.

Sample 2 (Kvalitet AMG-10 oil), similar to Sample 1, is characterized by effective rheological properties. The shear stress of the lubricating layers is, on average, 9.4 MPa at an oil temperature of 20 ^{0}C, which is close to that of Sample 1. When the oil temperature rises to 100 ^{0}C, this parameter increases to 10.82 MPa, which is slightly, by 1.15 times, higher than that for Sample 1.

As compared with Sample 1, the effective viscosity in contact is reduced, on average, by 1.53 times at an oil temperature of both 20 ^{0}C and 100 ^{0}C and is equal to 2764 Pa·s and 3309 Pa·s, respectively. With increasing temperature during 45 - 50 cycles of operation, a sharp decrease in this parameter to 78…240 Pa·s was established, which is due to adaptation of the lubricant boundary layers to the temperature change in the friction contact.

The range of change in the shear rate gradient of the lubricating layer (γ) in contact at a maximum sliding speed of 0.71 m/s in the conditions of rolling with sliding for Samples 1 and 2 is from $4.5 \cdot 10^3$ to $5.73 \cdot 10^5$ s^{-1}.

The formation of boundary films of lubricant with structural adaptability of the triboconjuction elements can lead to mechanical, physical and chemical changes in the surface layers of the metal, which can significantly affect the wear resistance of the contact surfaces. Analysis of changes in microhardness of the surface layers of steel 30KhHSA after 100 cycles revealed the dependence of this parameter on the type of test material. When using Sample 1 as a lubricant, strengthening of both leading and lagging surfaces was fixed. In particular, microhardness of the surface layers of the metal was increased by 512 and 517 MPa for the leading and lagging surfaces, respectively. In the case of using Sample 2, the leading surface of the metal was softened (decrease in microhardness upon testing was 696 MPa), while the lagging surface was strengthened (increase in microhardness was 444 MPa).

According to the data of the producers of Samples 1 and 2, they have the same base (mineral oil on the basis of deeply dearomatized low-solidificated fraction, which is obtained from hydrocracking products of a mixture of paraffinic oils and consists of naphthenic and isoparaffinic hydrocarbons). That is why it is active components of additives to Samples 1 and 2 that affect the kinetics of changes in microhardness of surface layers activated under friction. The additives present in Sample 1 are characterized by more effective antiwear properties and thus increase the wear resistance of contact surfaces in the conditions of rolling with sliding.

Conclusions

Sample 1 of oil "Bora B" AMG-10 (production: LLC "Bora B", TU U 19.2-38474081-010: 2016 with change 1) is characterized by more effective lubricating and rheological characteristics in nonstationary conditions of friction in the rolling with 30% sliding mode as compared to Sample 2 of AMG-10 oil (production: LLC NPP Kvalitet, GOST 6794-75 with changes1 - 5) according to the following criteria.

1. With Sample 1, there was not fixed any failure of the lubricating layer during start-up and direct metal contact of the friction surfaces. A semidry lubrication mode was only for a short time, during the periods of running-in and initial temperature rise. At start-up, regardless of the temperature of the lubricant, a mixed lubrication mode dominates, while at the maximum rotation speeds of the investigated samples a hydrodynamic lubrication mode dominates.

2. At a volumetric oil temperature of 20 and 100 ^{0}C, the thickness of boundary adsorption layers is 1.44 times that for Sample 2.

3. Sample 1 is characterized by low shear stresses, on average 9.4 MPa, regardless of oil temperature, and high effective viscosity, on average, 4249 and 5039 Pa·s at volumetric oil temperature 20 and 100 ^{0}C, respectively.

4. Additives present in “Bora B” AMG-10 oil (Sample 1) are characterized by more effective antiwear properties and increase wear resistance of contact surfaces in the conditions of rolling with sliding due to strengthening of surface layers of metal during operation, while in Sample 2 hardening-softening processes have been established, which cause a decrease in wear resistance of friction pairs.

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Ільїна О. А., Мікосянчик О. О., Мнацаканов Р.Г., Якобчук О.Є. Розробка методики оцінки якості змащувальних властивостей гідравлічних авіаційних олив.

Розроблено методику оцінки змащувальних та реологічних властивостей гідравлічних олив в триботехнічному контакті, яка полягає в дослідженні зразків товарних партій олив на програмно-апаратному комплексі з візуальною оцінкою кінетики зміни основних триботехнічних показників фрикційного контакту в режимі on-line. За допомогою роликової аналогії моделюється робота зубчастих передач в умовах кочення з проковзуванням 30%. Проаналізовано зразки оливи АМГ-10 двох виробників. Встановлено, що при зростанні температури мастильного матеріалу для зразка № 2 відбувається тривале відновлення захисних граничних плівок оливи, період їх формування збільшується в 2,5 рази, обумовлюючи реалізацію напівсухого режиму мащення при пуску. Загальна товщина мастильного шару в 1,27 разів менше, в порівнянні з оливою «Бора Б» АМГ-10 (зразок №1), незалежно від температури мастильного матеріалу. Визначено реологічні властивості олив та встановлені для оливи «Бора Б» АМГ-10 низькі напруження зсуву на рівні 9,4 МПа та висока ефективна в'язкість - 4249 та 5039 Па·с при об'ємній температурі оливи 20 та 100 ºC відповідно. Для зразка № 2 при зростанні температури оливи до 100 ºC напруження зсуву збільшується в 1,15 разів, ефективна в'язкість в контакті знижується в 1,53 рази. Присадки, наявні в оливі «Бора Б» АМГ-10, характеризуються більш ефективними протизношувальними властивостями та обумовлюють підвищення зносостійкості контактних поверхонь в умовах кочення з проковзуванням за рахунок зміцнення поверхневих шарів металу при напрацювання, для зразка № 2 встановлені процеси зміцнення – знеміцнення, що обумовлюють зниження зносостійкості пар тертя.

Ключові слова: авіаційні оливи, змащувальний шар, режим мащення, ефективна в'язкість, мікротвердість.

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Regression analysis of the influence of auger surface hardness on its wear during dehydration of solid waste in a garbage truck

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The article is dedicated to the study of the influence of the surface hardness of the auger on its wear during dehydration of solid waste in the garbage truck. Using the method of regression analysis, the logarithmic dependencies of auger wear depending on the hardness of its surface for different values of the friction path are determined. Graphical dependences of augur wear depending on the hardness of its surface for different values of the friction path are made up, and it confirms sufficient convergence of the obtained dependencies. Carried out additional regression analysis allowed to obtain the dependence of wear of the auger depending on the hardness of its surface and the friction path, which showed that during two weeks of operation and wear of the auger during dehydration of solid waste in the garbage truck increasing the surface hardness of the auger from 2310 MPa to 10050 MPa reducing the rate of growth of energy consumption of solid waste dehydration from 16.7 % to 1.5 %, and, consequently, to reduce the cost of the process of their dehydration in the garbage truck. The graphical dependence of the reduction of energy consumption of dehydration of solid household waste due to the increase in the hardness of the auger surface during its two-week wear is presented. It was established the expediency of further research to determine the rational material of the auger and ways to increase its wear resistance.

Key words: wear, surface hardness, auger press, garbage truck, dehydration, solid waste, regression analysis.

Introduction

In mechanical engineering, one of the main tasks is to increase the wear resistance and reliability of machine parts [1, 2]. A promising technology for primary processing of municipal solid waste (MSW), aimed at reducing both the cost of transportation of solid waste and the negative impact on the environment is their dehydration, accompanied by pre-compaction and partial grinding, during loading into the garbage truck. Dehydration of solid waste in the garbage truck is performed using a conical screw, the surface of which due to the existing friction wears out intensively. This is due to the fact that solid waste contains, in particular, components such as metal, glass, ceramics, stones, bones, polymeric materials, which can be attributed to abrasive materials because they have different shapes, sizes and hardness, and the presence of moisture 39 - 92 % by weight in MSW creates an aggressive corrosive environment. Therefore, the study of the influence of the hardness of the auger surface on its wear during dehydration of solid waste in the garbage truck is a topical task. Dehydration of solid waste in the garbage truck is performed using a conical auger, the surface of which due to the existing friction wears out intensively. This is due to the fact that solid waste contains, in particular, such components as metal, glass, ceramics, stones, bones, polymeric materials, which can be attributed to abrasive materials because they have different shapes, sizes and hardness, and the presence of moisture 39 - 92 % by weight in solid wastes creates an aggressive corrosive environment. Therefore, the study of the influence of the hardness of the auger surface on its wear during dehydration of solid waste in the garbage truck is an urgent task.

Literary review

The article [3] presents the results of experimental studies of wear resistance of different auger materials with different thermal and chemical-thermal treatment in a corrosive-abrasive environment on special friction machines that simulated the operating conditions of extruders in the processing of feed grain with saponite mineral impurities. The authors found that the wear resistance of materials in a corrosive-abrasive environment at elevated temperatures depends not only on the hardness of the friction surface, but also on its structure and phase composition and changes in the hardness gradient along the depth of the hardened layer. To ensure high wear resistance of extruders in the manufacture of animal compound feed with impurities of the mineral saponite, it is recommended to use for the manufacture of parts of the extrusion unit steel X12, hardened by nitro-hardening technology.

The author of [4] found that the intensity of abrasive wear of the screw surface of the auger, and wear resistance mainly depends on the hardness, surface roughness, the volume of abrasive particles involved in friction and the area of their contact with the surface

In the article [5], a mathematical model for calculating the wear rate of triboelements in a tribosystem operating in conditions of corrosion and abrasive wear was developed. The input factors were: active acidity, abrasiveness, roughness, load and sliding speed. Theoretically, the degree of influence of the above factors on the wear rate is established. It was found that abrasiveness is the most important factor, followed by the degree of decline - the level of active acidity and load.

A new design of the auger with a sectional elastic surface, which is designed to reduce the degree of damage to the grain material during its transportation is presented in the article [6]. The theoretical calculation of the interaction of the grain with the elastic section of the auger is carried out. A dynamic model has been developed to determine the influence of structural, kinematic and technological parameters of the elastic auger on the time and path of free movement of bulk material particles during their movement between sections, as well as to exclude the possibility of grain material interaction with the non-working surface of the auger working body.

In the article [7] it was determined that restoration of the auger requires surfacing or spraying a layer of a certain thickness on the end part of the coil of the auger, while the width of the restored layer is usually a few millimeters. An algorithm for selecting the optimal composite powder material for plasma spraying in order to increase the wear resistance of the working surfaces of machine parts, in particular the auger, is described. According to the authors, plasma spraying of composite powder materials will increase the durability of the auger by 2 - 3 times, which will reduce repair costs by tens of times.

In the article [8] the influence of geometrical parameters on productivity and design of the briquetting machine using the model of pressure based on the theory of piston flow is investigated. An analytical model that uses a pressure model was also developed based on Archard wear law to study the wear of augers of biomass briquetting machines. The developed model satisfactorily predicted the wear of the auger and showed that the greatest influence on it have the speed of rotation and the choice of material. The amount of wear increases exponentially to the end of the auger, where the pressure is the highest. Changing the design of the auger to select the optimal geometry and speed with the appropriate choice of material can increase the life of the auger and the productivity of the machine for briquetting biomass.

The work [9] is devoted to the analysis of the process of screw briquetting of plant materials into fuel and feed. Regularities of this process are the basis for determining the rational parameters of the working bodies. When designing briquette presses it is necessary to consider deformation of biomass taking into account change of physical and rheological properties at the moment of interaction with the auger mechanism

In the article [10] the wear of a twin-auger extruder of rigid PVC resins is investigated. The pressures around the cylinder when extruding two rigid PVC resins in a laboratory extruder with a diameter of 55 mm were measured and the forces acting on the auger core were determined. Numerical simulation of the flow was performed using the power parameters of the viscosity of the resins.

In the work [11] the process of pressing wood chips in auger machines was investigated. The processes occurring in different parts of the auger are established, formulas are defined that allow to calculate the loads acting on the auger coils, as well as to determine the power required for pressing. The specific energy consumption and the degree of heating of raw materials during pressing are determined.

The results of experimental studies of the process of solid dehydration based on the planning of the experiment by the Box-Wilson method are shown in the article [12]. Quadratic regression equations with the 1st order interaction effects were obtained using rotatable central composite planning for such objective functions as humidity and density of pre-compacted and dehydrated MSW, maximum drive motor power, energy consumption of solid waste dehydration. This allowed to determine the optimal parameters of equipment for dehydration by the criterion of minimizing the energy consumption of the process (auger speed, the ratio of the radial gap between the auger and the body, and the ratio of the auger core diameter to the outer diameter of the auger on the last coil) for both mixed and “wet”.

In the article [13] the improved mathematical model of work of the dehydration drive of MSW in the garbage truck is suggested that takes into account wear of the auger. It allowed to research numerically the dynamics of this drive during the start-up, and to define that with the increase of wear of the auger pressure of working liquid on the speed of the auger it is significantly reduced. The power regularities of change of the nominal values of pressures at the inlet of the hydraulic motor, angular speed and frequency of rotation of the auger from values of its wear are defined, the last of which describes detuning from optimum frequency of rotation of the auger in the course of its wear. It is established that the wear of the auger by 1000 μm leads to an increase in the energy consumption of solid dehydration by 11.6 %, and, consequently, to an increase in the cost of the process of their dehydration in the garbage truck and accelerate the wear process.

Purpose

Researching the influence of auger surface hardness on its wear during dehydration of solid waste in a garbage truck.

Methods

Determination of paired dependencies of auger wear from the hardness of its surface was performed by regression analysis [14]. Regressions were determined on the basis of literalizing transformations, which allow to reduce the nonlinear dependence to the linear one. The coefficients of regression equations were determined by the method of least squares using the developed computer program "RegAnaliz", which is protected by a copyright registration certificate, and is described in the article [15].

The following dependencies were used to determine the energy consumption of solid dehydration taking into account the auger wear [13]:

$$\begin{aligned}E &= 1504 - 15.92w_0 + 0.3214\rho_0 - 1.069n(u) - 2061(\Delta_{aug} + u)/(D_{\min} - 2u) - \\ &-1947(d_{\min} - 2u)/(D_{\min} - 2u) + 9.118\cdot 10^{-4} w_0\rho_0 + 0.002142 w_0 n(u) + \\ &+18.12w_0(\Delta_{aug} + u)/(D_{\min} - 2u) - 2.115w_0(d_{\min} - 2u)/(D_{\min} - 2u) + 4.392\cdot 10^{-4}\rho_0 n(u) - \\ &-2.005\rho_0(\Delta_{aug} + u)/(D_{\min} - 2u) + 0.3361\rho_0(d_{\min} - 2u)/(D_{\min} - 2u) + \\ &+0.09031w_0^2 - 7.923\cdot 10^{-4}\rho_0^2 + 0.008241n(u)^2 + 104172[(\Delta_{aug} + u)/(D_{\min} - 2u)]^2 + \\ &+1318[(d_{\min} - 2u)/(D_{\min} - 2u)]^2 \;[\text{kWh/tons}];\end{aligned} \tag{1}$$

$$n = 52.43 - 1.276\cdot 10^{-3} u^{1.5} \text{ [rpm]}, \tag{2}$$

where E – is the energy consumption of solid waste dehydration, kW·h/tons;
ρ_0 – initial density of solid waste, kg/m^3;
w_0 – initial relative humidity of solid waste, %;
n – the nominal speed of the auger, rpm;
u – auger wear, m;
Δ_{aug} – radial clearance between auger and housing, m;
$d_{\min}$ – outer diameter of the auger on the last coil, m;
$D_{\min}$ – is the diameter of the auger core on the last coil, m.

Results

The values of auger wear for different values of hardness of its surface and friction path are given in Table 1 [3].

As a result of regression analysis of the data in Table 1, is was determined the logarithmic dependencies of wear of the auger depending on the hardness of its surface for different values of the friction path:

$$u_{s=3000} = 334 - 34.07\ln H\,; \tag{3}$$

$$u_{s=6000} = 676.6 - 69.69\ln H\,; \tag{4}$$

$$u_{s=9000} = 999.8 - 103\ln H\,; \tag{5}$$

$$u_{s=12000} = 1295 - 132.7\ln H\,, \tag{6}$$

where u – wear, μm;
H – hardness of auger surface, MPa;

s – friction path, m.

Fig. 1 shows graphical dependences of auger wear depending on the hardness of its surface for different values of the friction path, made up using the dependences (3 - 6), which confirmed the sufficient convergence of the obtained dependencies compared with the data in the Table 1

Table 1

Screw wear values for different values of surface hardness and friction path [3]

№	Surface hardness, MPa	Wear, µm for friction path, m			
		3000	6000	9000	12000
1	2310	68	132	195	258
2	5180	53	103	153	203
3	6500	48	91	134	177
4	6510	43	80	116	152
5	6700	39	72	105	138
6	10050	25	43	63	88
7	5450	26	47	70	98
8	7860	24	43	64	90
9	8600	22	39	58	82
10	6950	28	52	78	108
11	8100	15	25	38	55

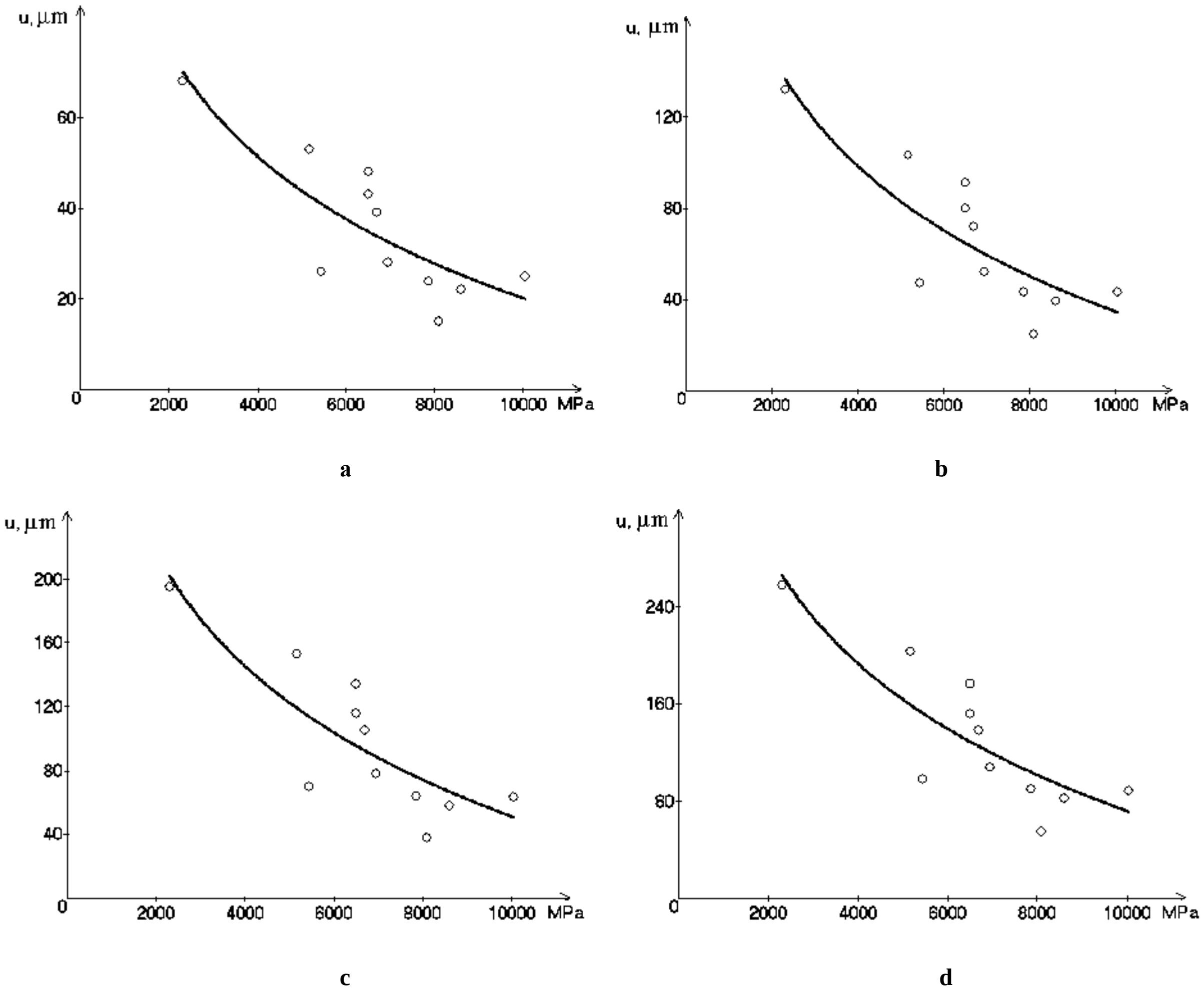


Fig.1. The wear of the auger depending on the hardness of its surface for different values of the friction path:
a – *s* = 3000 m;
b – *s* = 6000 m;
c – *s* = 9000 m;
d – *s* = 12000 m;
actual ○, theoretical —

Dependences (3 - 6) for different values of the friction path can be written in general as follows:

$$u = A(s) - B(s)\ln H, \quad (7)$$

where $A(s)$, $B(s)$ – regression coefficients that depend on the path of friction.

After the additional regression analysis, the regression coefficients which depend on the friction path can be described by power laws:

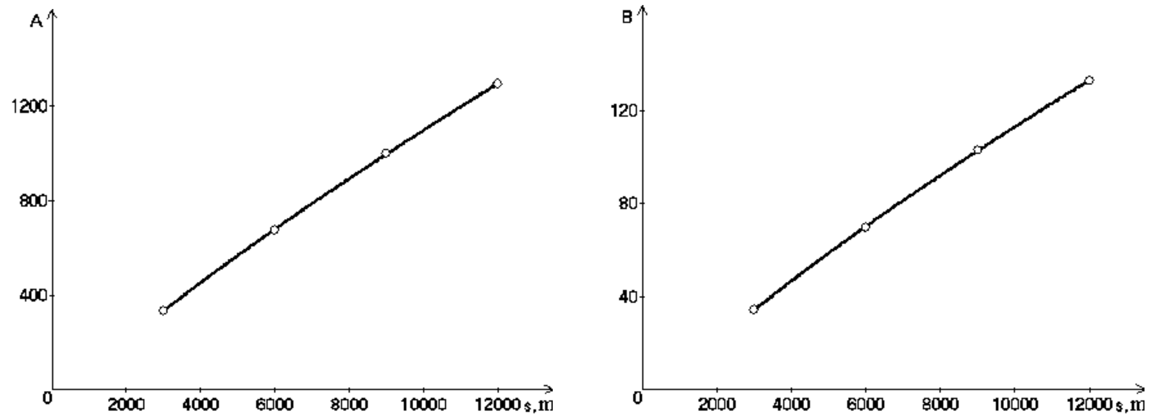
$$A(s) = 0.5796 s^{0.83} - 112.8; \quad (8)$$

$$B(s) = 0.08049 s^{0.8} - 14.71. \quad (9)$$

Table 2

The results of regression analysis of the dependence of the wear of the auger depending on the hardness of its surface for different values of the friction path

№	Type of regression	Correlation coefficient R for paired regressions					
		$u_{s=3000} = f(H)$	$u_{s=6000} = f(H)$	$u_{s=9000} = f(H)$	$u_{s=12000} = f(H)$	$A = f(s)$	$B = f(s)$
1	$y = a + bx$	0.81439	0.82376	0.82898	0.83452	0.99945	0.99918
2	$y = 1/(a + bx)$	0.66421	0.65974	0.67435	0.68887	0.92263	0.92049
3	$y = a + b/x$	0.78654	0.79114	0.79630	0.80646	0.93981	0.94199
4	$y = x/(a + bx)$	0.81920	0.80115	0.81299	0.82996	0.75583	0.59710
5	$y = ab^x$	0.75553	0.76285	0.77229	0.78180	0.97494	0.97347
6	$y = ae^{bx}$	0.75553	0.76285	0.77229	0.78180	0.97494	0.97347
7	$y = a \cdot 10^{bx}$	0.75553	0.76285	0.77229	0.78180	0.97494	0.97347
8	$y = 1/(a + be^{-x})$	0.66421	0.65974	0.67435	0.68887	0.92263	0.92049
9	$y = ax^b$	0.73813	0.74002	0.74896	0.76165	0.99969	0.99950
10	$y = a + b \cdot \lg x$	0.82369	0.83066	0.83603	0.84439	0.98598	0.98706
11	$y = a + b \cdot \ln x$	0.82370	0.83067	0.83604	0.84440	0.98598	0.98706
12	$y = a/(b + x)$	0.66421	0.65974	0.67435	0.68887	0.92263	0.92049
13	$y = ax/(b + x)$	0.56134	0.54627	0.55756	0.57539	0.99987	0.99976
14	$y = ae^{b/x}$	0.68071	0.67801	0.68615	0.70077	0.98768	0.98864
15	$y = a \cdot 10^{b/x}$	0.68071	0.67801	0.68615	0.70077	0.98768	0.98864
16	$y = a + bx^n$	0.75806	0.76923	0.77404	0.77662	0.99998	0.99996



a b

Fig. 2. Dependences of regression coefficients on the friction path a – *A* = *f*(*s*); b – *B* = *f*(*s*); actual○, theoretical —

The results of the regression analysis are shown in Table 2, where the cells with the maximum values of the correlation coefficient R for each of the paired regressions are marked in gray. Figure 2 shows the graphical dependences of the regression coefficients on the path of friction, constructed using the dependences (8,9), which confirm the sufficient convergence of the obtained dependencies.

After substituting the laws (8, 9) into the dependence (7), we obtain the law of wear of the auger depending on the hardness of its surface and the friction path:

$$u = 0.5796s^{0.83} - 112.8 - \left(0.08049s^{0.8} - 14.71\right)\ln H \, . \qquad (10)$$

Fig. 3 shows a graphical dependence of the wear of the auger in the plane of the parameters of influence: the hardness of its surface and the friction path.

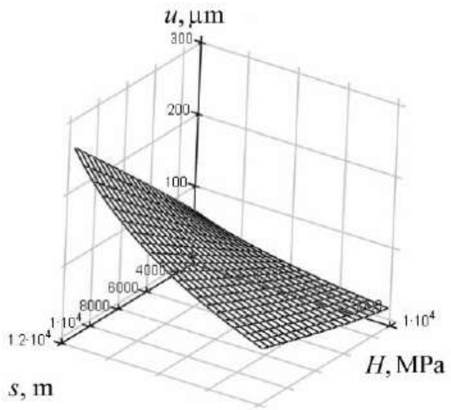


Fig. 3. The dependence of the wear of the auger *u* in the plane of the parameters of influence: the hardness of its surface *H* and the friction path *s*

Determine the path of friction for the surface of the auger dehydration of solid waste after its two-week operation:

$$s = \frac{\pi d_{av} n t_{cycle} K_{cycle} n_c}{26 \cdot 60} = \frac{3.14 \cdot 0.0775 \cdot 52.43 \cdot 855.6 \cdot 290 \cdot 28}{26 \cdot 60} = 56850 \text{ , m}$$

where d_{av} – the average diameter of the auger on the last coil, m;

t_{cycle} – the duration of the operating cycle of dehydration of one solid waste container, sec;

K_{cycle} – annual number of working cycles of the garbage truck taking into account dehydration of solid waste, pcs;

n_c – the number of solid waste containers loaded into the body of the garbage truck in one cycle, taking into account dehydration, pcs.

Figure 4 shows the graphical dependence of the influence of the hardness of the working surfaces of the auger of the mechanism of dehydration of solid waste on the energy consumption of the process (s = 56850 m), made up using dependencies (1, 2, 10).

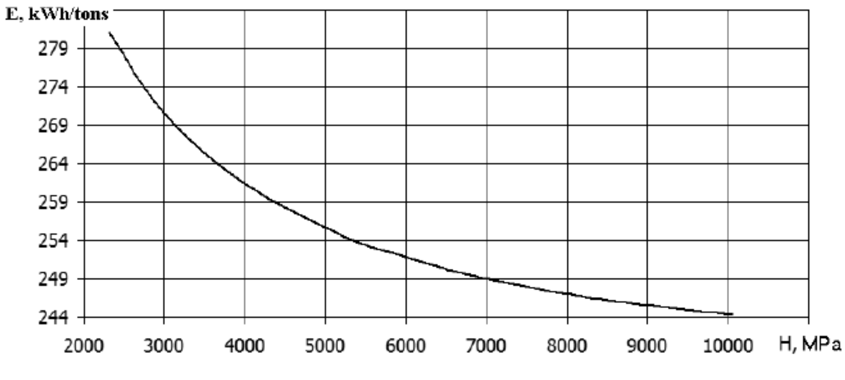


Fig. 4. The influence of increasing the hardness of the working surface of the auger to reduce the growth rate of energy consumption of the dehydration process after its two-week operation and wear (s = 56850 m)

Fig. 4 shows that after two weeks of operation and wear of the auger during dehydration of solid waste in the garbage truck the increase of the working surface hardness of the auger from 2310 MPa to 10050 MPa leads to a decrease in energy consumption of solid dehydration from 16.7 % to 1.5 %, and therefore it reduces the cost of the process of dehydration of solid waste in the garbage truck. Thus, the determination of the rational material of the friction surfaces of the auger and the ways to increase its wear resistance require further research.

Conclusions

The logarithmic dependence of auger wear depending on the hardness of its surface for different values of the friction path are determined. Carrying out additional regression analysis allowed to obtain the dependence of wear of the auger depending on the hardness of its surface and the friction path, which showed that during two weeks of operation and wear of the auger during dehydration of solid waste in the garbage truck increasing the surface hardness of the auger from 2310 MPa to 10050 MPa reducing the rate of growth of energy consumption of solid waste dehydration from 16.7 % to 1.5 %, and, consequently, it allowed to reduce the cost of the process of their dehydration in the garbage truck. Therefore, determining the rational material of the auger and ways to increase its wear resistance require further research.

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Березюк О.В., Савуляк В.І., Харжевський В.О. Регресійний аналіз впливу твердості поверхні шнека на його знос під час зневоднення твердих побутових відходів у сміттєвозі.

Стаття присвячена дослідженню впливу твердості поверхні шнека на його знос під час зневоднення твердих побутових відходів у сміттєвозі. За допомогою використання методу регресійного аналізу визначено логарифмічні закономірності зносу шнека залежно від твердості його поверхні для різних значень шляху тертя. Побудовано графічні залежності зносу шнека залежно від твердості його поверхні для різних значень шляху тертя, що підтверджують достатню збіжність отриманих закономірностей. Проведення додаткового регресійного аналізу дозволило отримати закономірність зносу шнека залежно від твердості його поверхні та шляху тертя, за допомогою якої встановлено, що при двотижневій експлуатації та зношуванні шнека під час зневоднення твердих побутових відходів у сміттєвозі збільшення твердості поверхні шнека з 2310 МПа до 10050 МПа призводить до зниження темпів зростання енергоємності зневоднення твердих побутових відходів з 16,7% до 1,5%, а, отже, і до здешевлення процесу їхнього зневоднення у сміттєвозі. Представлена графічна залежність зниження енергоємності зневоднення твердих побутових відходів внаслідок збільшення твердості поверхні шнека при його двотижневому зношуванні. Виявлено доцільність проведення подальших досліджень з визначення раціонального матеріалу шнека та шляхів підвищення його зносостійкості.

Ключові слова: знос, твердість поверхні, шнековий прес, сміттєвоз, зневоднення, тверді побутові відходи, регресійний аналіз.

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Influence of filter elements on the operation of tribomechanical systems

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Annotation

Oil filter is a part of a gasoline or diesel engine lubrication system designed to clean the engine oil.

Depending on where it is installed, the oil filtration system, they are divided into three types:

- through-flow filter, which passes through all the oil that the pump feeds into the engine. A pressure regulating by-pass valve is installed upstream of the filter to protect the gaskets with oil seals. If the filter element is too dirty, the valve directs oil flow past the filter, preventing oil starvation of the bearings. Keeps engine from failing due to lack of lubrication;

- a partial-flow filter is mounted parallel to the main oil line and cleans only a portion of the oil that enters the engine. Gradually the whole volume of oil passes through the filter element, giving a fairly high cleaning efficiency. However, this method does not provide absolute protection of parts from chips and other abrasives;

- the combination filter combines a full-flow and a partial-flow cleaning principle. It consists of two filter elements, one mounted parallel to the oil line and the other cut into it. This ensures maximum cleaning efficiency and long filter life. The filter elements are divided into two types according to their efficiency in removing fine impurities: coarse filters, which remove coarse impurities, and fine filters, which remove fine impurities. According to the design of the housing and the possibility of replacing the filter element, filters are divided into multiple (collapsible) and disposable (non-collapsible). Modern engines may use filters in the form of a cartridge, which is inserted into a special compartment.

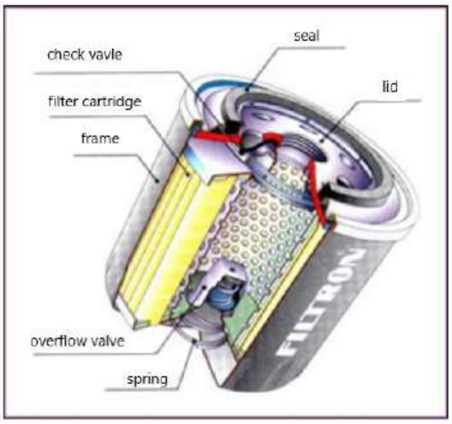
During operation, the oil is first routed to the filter and then through the oil channels to the interacting parts in the engine. This principle is used on all standard passenger cars. A settling filter (gravity filter) is a tank with a filter element and a settling tank in which impurities are deposited by gravity. The centrifugal filter operates similarly to the gravity filter, only dirt settles in it under the action of centrifugal force resulting from the rotation of the body.

Key words: filter, oil, thickness of lubricating layer, friction coefficient.

Introduction

The popular design of the oil filter includes a metal housing in which a corrugated porous paper filter element is installed and a spring-loaded bypass (overflow) valve.

A backflow (overflow) valve is installed at the inlet of the housing to prevent oil leakage after the engine is stopped. This allows oil to be stored in the oil passages and ensures nominal pressure immediately after the engine is started. The tightness of the housing connection to the oil channel is ensured by a rubber gasket. The oil, which is supplied by the oil pump, passes through a paper filter and is sent to the engine channel. If the filter is dirty or the oil is very viscous and does not flow well through the filter element, excess pressure opens the bypass valve which directly starts the oil in the engine.



Design of the oil filter

Analysis of recent research and publications

The influence of the degree of oil contamination by mechanical impurities on tribotechnical properties of lubricating materials is investigated by many scientists: Wenzel E.S. [1], Garkunov D.N., Dmitrichenko M.F., Mnatsakanov R.G., Mikosyanchik O.O., Bilyakovich O.M., Zhuravel D.P. [2, 3], and others. The paper [4] raises the question and analyzes the experimental data on the possibility of using synthetic porous-fiber polypropylene as a filtering material instead of the existing ones. The quality of oil filtration determines the service life of the engine. In [5] the possibility of using crystalline urea for the process of purification of waste mineral engine oils from acidic aging products was considered. Preference should be given to well-known manufacturers: Bosch; Mahle; Mann; Fram. You should also pay attention to the filters that are recommended by the car manufacturer. Visually determine the degree of contamination of the oil filter is impossible. Therefore, it is necessary to follow the manufacturer's recommendations for its replacement. Replace the filter when changing the engine oil. Depending on the oil brand and peculiarities of the car design, replacement is performed after 8-15 thousand kilometers of mileage. What happens, if the oil filter is removed. If the filter element of the lubrication system is removed, abrasive particles, especially metal filings, will get on the sliding bearing shells. This will accelerate the wear of the metal, which in turn will also enter the lubrication system. As a result, an avalanche-like process will begin, which will end with an overhaul or replacement of the engine. If the filter is not changed in time, it will get clogged and the crude oil will start circulating through the bypass valve. This will also lead to engine failure [6].

Statement of the problem

To establish the effect of contamination of engine oil during operation it is necessary to analyze the formation of the thickness of the lubricating film in tribosystems.

Basic research and presentation of scientific results

Laboratory methods of identifying wear particles. Solid particles of suspension in the filtration process can not only be trapped on the surface of the filter partition, but also penetrate into its pores. Penetration of solid particles into the pores of the partition is undesirable because it leads to a sharp increase in its resistance, which is more difficult to reduce by subsequent washing than by placing solid particles on the surface of the partition. Therefore, when separating such suspensions, it is advisable to prevent the penetration of solids into the pores of the bulkhead and retain them on its surface.

Spectral analysis of oil samples taken from the engine lubrication system by monitoring the concentration of metals and other impurities in the oil in grams per ton is widely used in engine operation. In most cases, iron and copper concentrations are monitored. Oil samples are taken 15 ... 40 minutes after the engine is stopped while the wear particles are in suspension. Frequency of sampling is set depending on the predicted intensity of wear and maintenance intensity, as a rule, at least after 200 hours of operation. If the content of wear products in the oil increases, sampling is performed more frequently.

Normal curve of the law of metal content change is shown in Fig. 1.

The 1st area - running-in period of the engine components. The graph first shows growth and then decrease of metal content at the end of running-in period up to the value characterizing normal metal content. There is an individual value of metal concentration for each engine within permissible limits, depending on the engine operation modes (the time of operation in different modes).

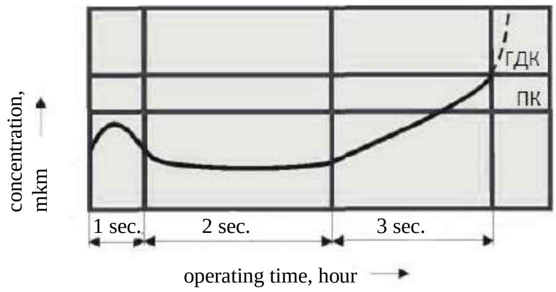


Fig.1. The normal law of change in the content of metals in the oil in the course of engine operation. PC - increased concentration, at achievement of which the engine is allowed to operate under "special control". MPC - maximum permissible concentration, reaching which the engine is to be withdrawn from operation

The 2nd area - a period of normal wear. There may be a slow growth of metal concentration as the engine runtime increases.

The 3rd area - the period of intensive wear. This period precedes the destruction of the engine unit (or is associated with partial destruction). Intensity of concentration increase is higher, than on site 2.

The normal law of change of metal content in oil is realized only if the engine is in the process of wiping surfaces with the release into the oil of wear particles not more than 15 ... 30 μm. It is a size that most modern instruments are capable of accounting for.

One of the most common and dangerous types of wear is pitting. It is fatigue pitting of contact surfaces [7]. The beginning of the pitting process is the plastic deformation of the surface and the formation of fatigue microcracks on the friction surfaces. Up to the appearance of the first cavern of fatigue pitting the only kind of wear particles formed in oil are spherical particles of 3... 5 microns, the weight contribution of which to the total mass of particles formed during normal wear is a few percent, coinciding with the error of the spectrometer measurements.

Further wear releases large particles, which are not taken into account in the spectral analysis until significant damage occurs with the release of a large number of both large and small particles. Changes of metal content in oil during fatigue damage of parts are shown in Fig. 2.

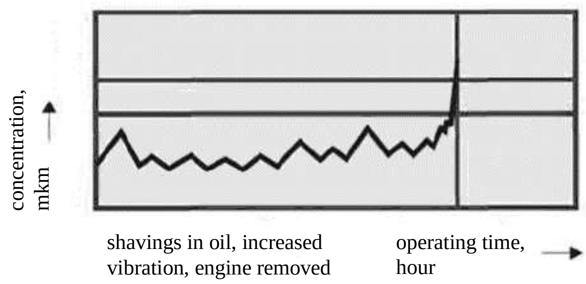


Fig. 2. Change of metal content in oil during fatigue damage of rubbing surfaces

There are technologies of spectral analysis, in which all particles in the sample are taken into account. This is achieved by dissolving wear products in an acid solution and then analyzing them on a special spectrometer.

The pherographic analysis is used to detect fatigue damage processes. As a rule, part of an oil sample is used for spectral analysis. The ferogram is examined under a microscope where the shape, size and number of particles are determined.

Laboratory examination of oils using a mesh filter element

The formation of an oil film between two parts which are in contact and move relative to each other depends on the speed of reciprocal movement. In such cases, we speak of a hydrodynamic friction regime, when the oil film is drawn into the gap between the interacting parts, separating the parts. The film is drawn into the gap more effectively (the film becomes thicker) with increasing velocity. But an increase in velocity results in an increase in the amount of heat generated by friction. The temperature of the oil rises and it becomes more fluid. This leads to a reduction in film thickness as a result of oil rarefaction. The friction coefficient depends on the roughness and accuracy of the geometry of the contacting surfaces and the presence of foreign particles in the oil (surface irregularities, foreign particles, disrupt the integrity of the film, which leads to the appearance of zones operating in semi-dry friction mode). The friction force value is proportional to the load, and it is at a constant friction coefficient. Sometimes the integrity of the oil film can be broken, and the coefficient of friction begins to grow. Then, even if the load is constant, the torque increases and conditions are created for the sliding bearings to rotate. The increased load reduces the thickness of the oil film, increasing the risk of its destruction [8]. In this case, more heat is released and leads to an increase in local temperatures in the friction zone. Oil liquefaction occurs, which leads to a further decrease in the thickness of the oil film and increases the probability of occurrence of sticking in the friction zone.

These factors have a particularly strong effect during the initial period of machine operation, during the running-in period of the parts. During this period of operation, micro-irregularities are triggered, destroying the oil film. At this moment the rubbing pairs are the most sensitive to overloading.

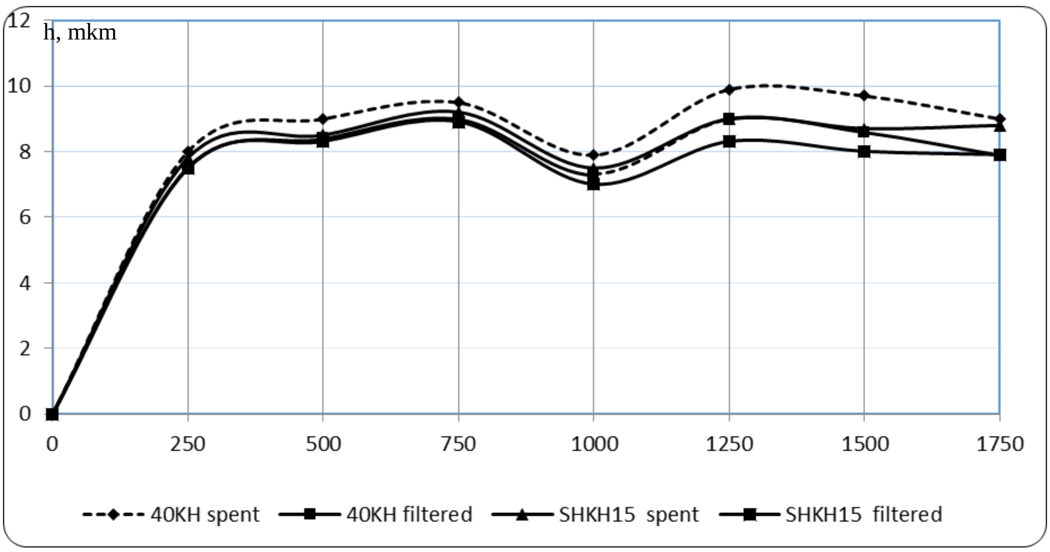
When performing this experimental study, the aim was to establish the mechanism of formation of the lubricating layer thickness in the contact, to determine the dynamics of wear of tribocoupling elements depending on the material of the contact surfaces, antifriction and rheological characteristics of the engine oil. At research was used the used (operating time 15 thousand km in the gasoline engine) motor synthetic oil Mobil-1 0w-40 in two states:

1) directly drained from the internal combustion engine;
2) filtered with Champion C102 filter.

The studies were carried out on the SMTs-2 unit, in the start-up (4.5 sec) - stop mode (3 sec). Number of cycles in the experiment was N = 1750, which corresponds to 4 hours of continuous operation of the installation, following one another, without interruption. In the friction pair was reproduced rolling mode at a relative slip of 18 %. As samples were used cylindrical rollers (d = 50 mm) made of Steel 40KH (HRC = 43) and Steel SHKH15 (HRC = 58). The maximum Hertz contact load was 570 MPa, and the volumetric temperature of the oil during the experiment was 70 °C.

The specified Mobil-1 0w-40 oil provides reliable protection for a wide range of both gasoline and diesel engine models, including turbocharged engines, intercooled charge air systems, as well as advanced direct-injection diesel (DI type) engines operating in any severity conditions.

Regularities of formation of the lubricating film thickness at start-up, depending on the use of different steel grades and used and filtered oils are shown in Fig. 3.



N cycles

Fig. 3. Formation of the lubricating film thickness in contact with used samples of Mobil-1 0w-40 engine oil depending on the hardness of the surface material

Regardless of the hardness of the contact surfaces, it was found that during N < 100 cycles, according to the calculated parameter λ, the boundary regime of lubrication dominates, during 100 < N <150 cycles the elastohydrodynamic regime in the contact, the subsequent operating time up to N = 1750 cycles is characterized by the realization of the hydrodynamic regime in the contact. Only during the initial break-in period, which corresponds to N < 200 cycles, a high carrying capacity of the lubricating layer formed on the contact surfaces of steel 40KH is observed; during operating time 200 < N <1750 cycles a general trend is observed for the growth of the lubricating layer thickness during break-in - for the two studied steel grades this parameter averages 8.2 - 10 μm.

Since the experiments were conducted under nonstationary operating conditions, there is a high probability of the lubricating layer breaking off at stopping, which leads to direct contact between the surfaces. Under these lubrication conditions, the formation of boundary adsorption films on the elements of the tribocoupling is of great importance.

To establish patterns in the kinetics of formation of adsorption boundary films, we measured the thickness of the lubricating film at the stop (Fig. 4).

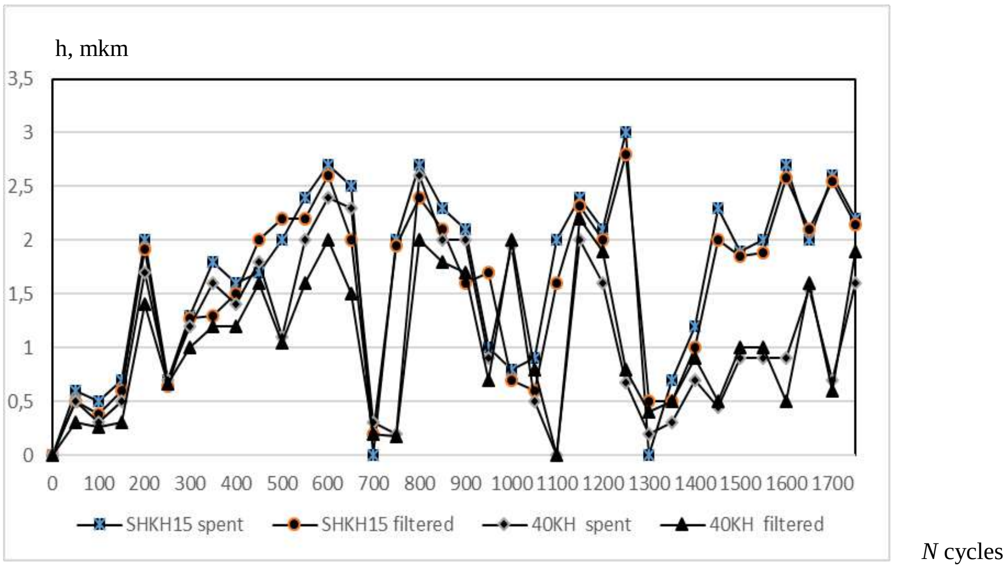


Fig. 4. Kinetics of changes in the thickness of the boundary adsorption films

For the Steels 40X and the SHKH15 after the running-in phase, which corresponds to N < 450 cycles, it was found that there is no sagging of the lubricating film thickness. With further operating time up to N = 1750 cycles, the breakdown of the lubricating film for these steel grades was only 3 %. We assume that this is associated with the ability of chromium to reach the contact surface [1], which increases the adsorption force of the interaction of the surface films of the metal in the polar-active components of the lubricant, manifested in an increase in the thickness of the non-hydrodynamic component of the lubricating film to the end of operating time, which is 1.7 μm and 2.5 μm for the Steel 40X and the SHKH15 , respectively. At the same time, a clear pattern in the formation of the boundary adsorption films is observed depending on the condition of the oils, namely: in the filtered oil sample a lower value of the studied index by 6 % compared to the used oil is observed. This can be explained by the presence of less content of mechanical impurities and fuel combustion products.

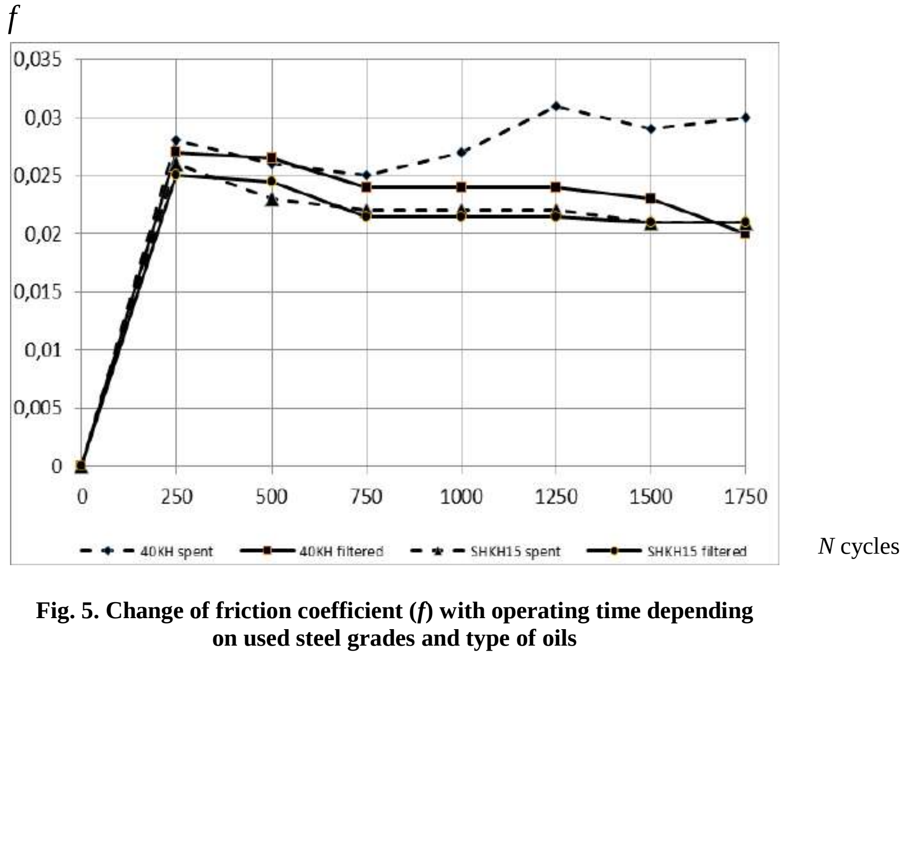


Fig. 5. Change of friction coefficient (*f*) with operating time depending on used steel grades and type of oils

When studying the friction coefficient (Fig. 5) for rollers made of the Steel 40KH in the initial period of running-in up to N < 200 cycles value of this parameter is less - 0,028, and for filtered oil 0,026 that is explained by the greatest value of a hydrodynamic component of greasing film thickness at the start-up, further at 250 < N < 950 cycles value of friction factor is established almost the same for the Steel SHKH15 and varies within 0,028-0,026; At N > 1000 the highest value of the studied parameter when using the Steel 40KH for the used oil - 0,031 - 0,029, which is associated with a smaller thickness of the non-hydrodynamic component of the lubricating film. Also the decrease of antifriction properties at the final stage of workover is determined for the samples of the Steel 40KH with use of filtered oil. This decrease is caused by the highest values of the oil film displacement stress in the contact due to the increase of the oil effective viscosity, namely, for the Steel 40X: η_{ef} = 10,19 – 10^2 Pa · s, for the SHKH15 the ratio is respectively (1 : 0,745) (fig. 5). At the same time, for the filtered oil, the improvement of antifriction properties with both steels under study is found, explained by the highest values of the non-hydrodynamic component of the lubricating layer thickness.

So, the kinetics of friction coefficient change in the contact is influenced by the rheological properties of the lubricant. In our opinion, the mechanism of this process is as follows. At initial formation of boundary adsorption films there is structure of lubricant components on the friction-activated contact surface, which leads to increase of shear stresses of oil film and correlation growth of friction coefficient. As the boundary films adapt to dynamic load conditions, the displacement stresses of adapted adsorption films decrease, which also causes increase of antifriction properties. So, at N > 1000 cycles of operating time boundary films with thickness of 2,5 microns, stable to mechanical and thermal destruction, which are characterized by the most effective antifriction properties, in comparison with another type of lubricant, are formed on the surfaces from the Steel SHKH15 using filtered oil.

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Дмитриченко М.Ф., Савчук А.М., Туриця Ю.О., Міланенко О.А. Вплив фільтруючих елементів на роботу трибомеханічних систем.

Масляний фільтр – це деталь системи мащення бензинового або дизельного двигуна, призначена для очищення моторної оливи.

В залежності від місця монтажу, системи фільтрації оливи вони діляться на три типи:

- повнопоточний фільтр, що пропускає крізь себе всю оливу, яку насос подає в двигун. Для захисту прокладок з сальниками перед фільтром встановлюється перепускний клапан, що регулює тиск. При надмірному забрудненні фільтруючого елемента клапан направляє потік оливи повз фільтра, запобігаючи масляне голодування підшипників. Це не дає двигуну вийти з ладу через відсутність мащення.
- частковопоточний фільтр монтується паралельно головному оливопроводу і очищає лише частину оливи, що надходить у двигун. Поступово весь обсяг оливи проходить через фільтруючий елемент, що дає досить високу ефективність очищення. Однак такий спосіб не забезпечує абсолютного захисту деталей від стружки та інших абразивів.
- комбінований фільтр поєднує повнопоточний і частковопоточний принцип очищення. Він включає два фільтрувальних елементи, перший з яких встановлюється паралельно масляній магістралі, а другий врізається в неї. Це забезпечує максимальну ефективність очищення і довгий термін служби фільтрів. По ефективності видалення дрібних домішок фільтруючі елементи діляться на два види: фільтри грубого очищення, призначені для видалення великих включень; фільтри тонкого очищення, що видаляють мілкофракційні включення. По конструкції корпусу і можливості заміни фільтруючого елемента фільтри діляться на багаторазові (розбірні) і одноразові (нерозбірні). У сучасних двигунах можуть застосовуватися фільтри у вигляді картриджа, який вставляється в спеціальний відсік.

Під час роботи олива спочатку подається до фільтра, а потім по масляним каналам до взаємодіючих деталей в двигуні. Цей принцип застосовується на всіх серійних легкових автомобілях. Фільтр-відстійник (гравітаційний) представляє собою ємність з фільтруючим елементом і відстійником, в якому домішки осідають під дією гравітації. Відцентровий фільтр, що працює аналогічно гравітаційному, тільки бруд осідає в ньому під дією відцентрової сили, що виникає в результаті обертання корпусу. Процес утворення масляної плівки між двома контактуючими й рухомими відносно одна одної деталями залежить від швидкості взаємного переміщення.

При виконанні даного експериментального дослідження метою являлося встановлення механізму формування товщини мастильного шару в контакті, визначення динаміки зношування елементів трибоспряження залежно від матеріалу контактних поверхонь, антифрикційних та реологічних характеристик моторної оливи.

Ключові слова: фільтр, олива, товщина мастильного шару, коефіцієнт тертя.

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Research of Wear Resistance of a Covering of Shafts of the Turbocharger of the Diesel Engine Restored by Means of a Gas-Dynamic Spraying

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Abstract

The analysis of tribological researches on the most perspective way of restoration of a primary resource of engines by means of a gas-dynamic spraying is resulted in article. It was found that to reduce the coefficient of friction and increase the wear resistance of the coating is theoretically justified the use of copper-zinc powders brand C-01-11, applied by gas-dynamic spraying. It is established that the physical and mechanical properties of the coatings (roughness, microhardness, friction coefficient) on the restored turbocharger meet the requirements of the manufacturer. The coefficient of friction in the connection, the rotor shaft (reduced powder with copper and zinc), with a plain bearing (made of tin-lead bronze Bros - 10 - 10) is 20 % less than in the connection where the rotor shaft is made of steel 40. The total wear in the bearing assembly with the restored gas-dynamic sprayed rotor shaft is 20 % lower than in the assembly where the rotor shaft is restored by the basic technology. The technology of restoration of a surface of a shaft of a rotor of the turbocompressor, under the bearing of sliding (gas-dynamic spraying) which increases its resource by 23 % in comparison with basic technology of repair of a shaft of a rotor is developed. This allows you to increase its operating time with the established regulatory and technical documentation for overhaul of the engine. A stand for testing diesel turbochargers with recovery technology has been developed, which allows to determine the parameters and characteristics of diesel engine turbochargers in different periods of operation, running-in and adjustment. Tests on the stand showed that turbochargers with restored rotor shafts according to the proposed technology after 2000 operating hours increase all performance by 13 % more than turbochargers repaired by the basic technology. Operational tests have shown that turbochargers repaired using the proposed technology have an operating time of 989 moto-hours more than turbochargers repaired with existing technology.

Key words: gas-dynamic spraying, wear resistance, friction steam, turbocharger, recovery technology, wear intensity.

Introduction

At the present stage of economic development of Ukraine there are questions of design, production and efficient operation of existing machinery and equipment. Improving the efficiency of the existing fleet of machines can be achieved by increasing the efficiency of equipment, reducing the cost of its operation and reducing downtime for technical reasons [1].

The fleet of tractor and agricultural machinery, as well as mobile diesel power plants used in the agro-industrial complex, is now characterized by accelerating its moral and material wear. This increases the intensity of machine failures and the duration of downtime associated with the restoration of their efficiency, increasing the cost of unscheduled maintenance and repair (TE and P).

The main power unit of almost any machine used in agricultural trials is a diesel. The analysis showed that the least reliable unit of the diesel engine is a turbocharger (TCR). It accounts for more than 15 % of engine failures. In turn, more than 80 % of TCR failures are due to the bearing assembly of the rotor shaft [2, 3]. The difficulty of repairing turbochargers of tractor and agricultural machinery is that the areas where the equipment is operated are remote from the repair and maintenance bases. To reduce machine downtime for technical reasons, it is necessary to increase the turbocharger to the value set by the regulatory and technical documentation for TE and P engine.

Analysis of ways to increase the efficiency of friction joints shows that the most promising for the repair of the bearing assembly of the turbocharger is to restore the seating surface of the rotor shaft of the turbocharger with materials that reduce the load on the unit.

Literature review

Repair of agricultural machinery by 70 - 80 % is carried out using spare parts, while downtime of machines due to their lack or low quality leads to large losses of agricultural products. The cost of spare parts is constantly growing and therefore the restoration of worn parts with the provision of their resource at the level of new - one of the most effective ways to save money. According to GOSNITI 85% of details, at their defectation, have wear no more than 0,3 mm, ie their working capacity is restored at drawing of coverings of insignificant thickness. However, the life of refurbished parts, compared to new parts, in many cases remains low [4].

Scientists and specialists from such organizations as TsNIIME, SPKTB Soyuzlesremmash (GNTSLPK), MGUL, VLTA and others carried out a large amount of work to organize and improve the technical level of operation of forest machines and repair work in the industry. Among them V.V. Balikhin, V.V. Bykov, I.V. Voskoboinikov, V.N. Vinokurov, V.A. Goberman, N.S. Eremeev, V.M. A. V. Kotikov Pitukhin, A.V. Serov and others.

To increase the net power of internal combustion engines, the most effective currently is the installation of a turbo compressor. The turbocharger increases engine power by 30 % without increasing its displacement.

The turbocharger is also the least reliable part of a diesel engine. More than 60 % of failures are due to the turbocharger. Table 1 shows the share of failures using the example of the repeatability factor of the main elements of a turbocharger. About 81 % of turbocharger failures are caused by it.

Table 1

The results of laboratory tests for wear resistance of friction pairs

Stationary specimen	Moving specimen	$I_к$	I_p	I_Σ
Tin-lead bronze BrOS-10-10	Steel 40	$7.14 \cdot 10^{-11}$	$4.23 \cdot 10^{-11}$	$1.37 \cdot 10^{-10}$
Tin-lead bronze BrOS-10-10	Nickel-plated coating	$1.27 \cdot 10^{-11}$	$1.57 \cdot 10^{-11}$	$2.84 \cdot 10^{-11}$
Tin-lead bronze BrOS-10-10	Copper-zinc coating	$0.74 \cdot 10^{-11}$	$1.38 \cdot 10^{-11}$	$2.12 \cdot 10^{-11}$

At the same time; there are examples when the resource of parts restored by progressive methods is several times higher than the resource of new parts. When choosing a method of restoring parts, it is necessary to ensure high quality coatings, low cost of the process, minimum material consumption, labor and energy costs. At the same time it is necessary to concentrate the attention on such ways which increase reliability not only details, but also all assembly unit as a whole [5].

The main trend in the development of modern tractors and combines; engines - increase of aggregate capacities at practical preservation of their weight and dimensions due to application of turbocharging. High technical and economic indicators of gas turbine supercharging, as a way to increase power by 15 … 30 %, led to its widespread use in tractor and combine engines. At present, all combine and multi-tractor engines (SMD-60/61, SMD-62/63, SMD-64/65, SMD-66/67, SMD-31/32, SMD-17/18, etc.) are provided with gas turbine supercharging. For supercharging of these engines turbocompressors of the standard sizes are used: TKR - 11 (with external diameter of a wheel of the compressor Dk = 110 mm), TKR-9 (with Dk = 90 mm), TKP - 8,5 (with Dk = 85 mm), TKP - 7 (with Dk = 70 mm).

Experience in operation of turbochargers type TKP - 11, installed on engines SMD-60/61, SMD-62/63, SMD-64/65; SMD-66/67, SMD-31/32, SMD-17/18, YAME-238NB, YaMZ-240NB shows that this unit is the least durable unit. Thus, according to GOSNIT, the number of turbocharger failures is 2 … 13 % of the total number of engine failures. The life of the repaired turbocharger is only 62 % of the life of the new one [6].

Currently, there are two main areas in the repair of worn parts of turbochargers. The most common methods - installation of repair parts, the method of repair dimensions, plastic deformation. Rarely used on the surface of the layer: metal, compensating for the amount of wear (galvanic, surfacing methods).

All existing methods, along with the advantages, have certain disadvantages. When repairing turbochargers, it is necessary to restore parts made of different materials (steel, bronze, aluminum alloy) and different configurations; (planes, cylindrical outer and inner surfaces). In this regard, to restore worn parts, you need a large range of equipment used [7, 8].

To improve the quality of repair and increase the service life of overhauled turbochargers, it is necessary to develop a recovery technology that ensures durability and trouble-free operation of the unit for the period before overhaul of the diesel engine on which this turbocharger is installed [9].

One of the promising ways to restore the primary life of the engines is gas-dynamic spraying, but the practical application for the restoration of smooth cylindrical surfaces on the example of the rotor shaft of the turbocharger, this method did not work. Therefore, the task is to theoretically substantiate the increase in the total wear resistance of the node "rotor shaft - plain bearing" and develop a method of restoring the surface of the rotor shaft, which will increase the turbocharger to overhaul the engine.

Purpose

The purpose of the research is to increase the reliability of turbochargers for diesel tractors and agricultural machinery by increasing the service life of the rotor shaft.

Research methodology

To determine the coefficient of friction and assess the intensity of wear, a set of laboratory tests was performed, including checking the properties of the surface (hardness, roughness) and testing for the intensity of wear on the friction machine.

A stand for overhaul tests of overhauled turbochargers was developed for bench tests.

For research, turbochargers were removed from diesels operated on farms. Next, the turbochargers underwent a recovery procedure using the proposed technology. The repaired turbochargers were tested on a stand of our own design, according to our proposed new method. To pass operational tests, turbochargers were installed on tractors operated on farms [10].

The wear resistance of the coating is the most important criterion for assessing the service life of a combination exposed to friction. Its value is greatly influenced by the physical and mechanical properties of the coatings, the state of roughness and microhardness of the surface layer of the joint, as well as the coefficient of friction and the load force acting on the joint.

When measuring the roughness of the coating, it was found that after testing in the reduced shaft of the rotor coated with powder C-01-11, the roughness parameter for Ra was 0.235 μm, which is significantly less than in other test samples. Reducing the value of surface roughness significantly reduces the intensity of wear. During the bench tests, the turbocharger was disassembled at intervals of 100 moto-hours. The value of the roughness parameter did not change with increasing operating time.

Research results

During tribotechnological laboratory tests on the AI-5018 friction machine, measurements of various parameters at simulation of various types of work of the engine were made. In fig. 1 shows the dependence of the change in the coefficient of friction in the test materials under the influence of simulating the dynamic load.

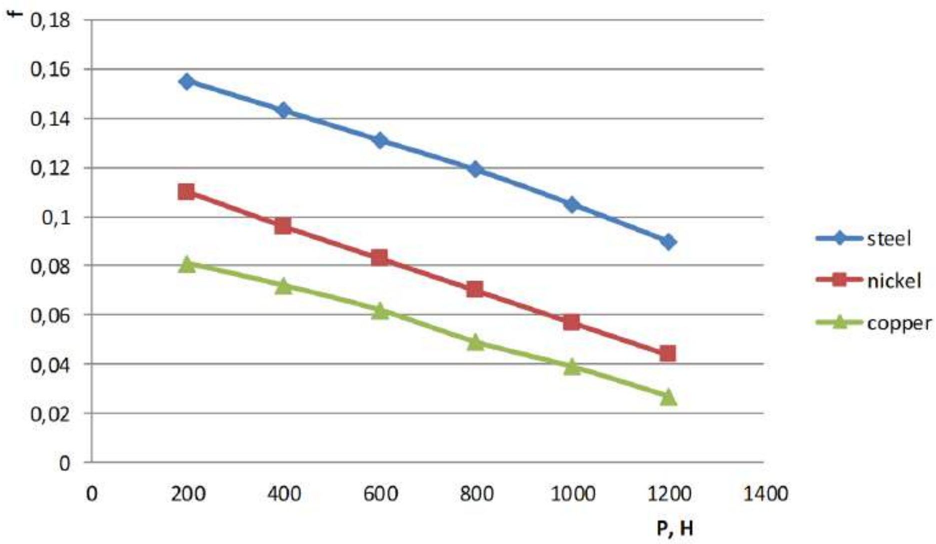


Fig. 1. Dependence of change of coefficient of friction on force, acting on the connection

From the presented dependence it follows that the rotor shaft of the turbocharger coated with copper-zinc powder brand C-01-11 has a lower coefficient of friction than a standard shaft made of steel or a shaft having a nickel coating N-00-14. This is due to the fact that initially the basic coefficient of friction in copper is lower than in nickel and especially steel.

After a comparative assessment of the mutual wear intensity of the rotation pairs, the dependences of the change in the wear intensity were revealed [11]. In fig. 2 shows the total wear intensity of the samples after testing for wear resistance on the friction machine AI - 5018. In table 1 shows the wear intensity of the rotor roller, simulating the shaft, and the pad, which simulates a plain bearing when tested on a friction machine.

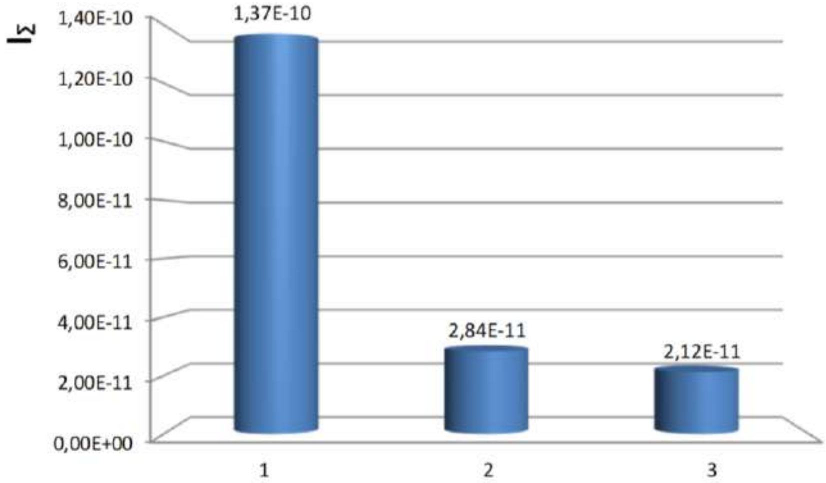


Fig. 2. The total wear intensity of the experimental samples:
1 – steel with bronze;
2 – nickel with bronze;
3 – copper with bronze

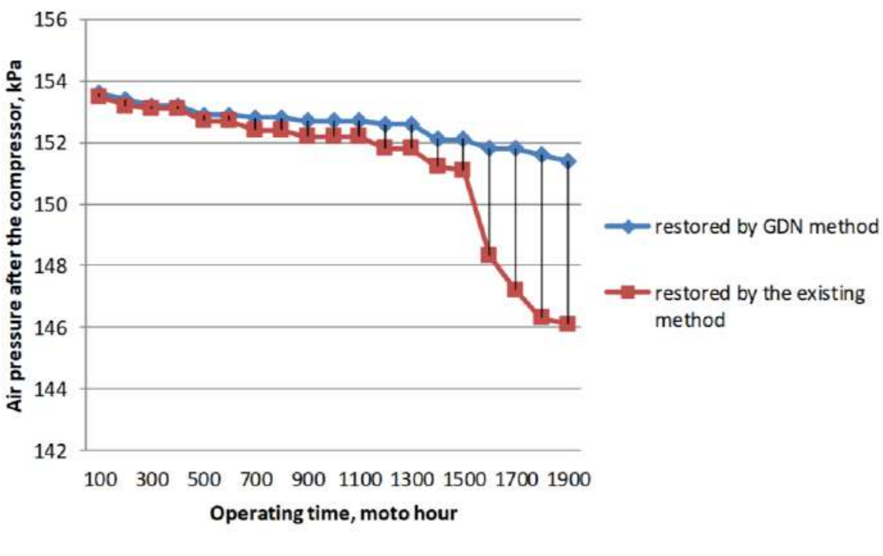
The diagram shows that the wear rate of the bearing assembly, restored by gas-dynamic spraying with copper-zinc powder is seven times less than the standard friction pair of steel with bronze.

Wear intensity in friction pairs "copper-zinc - BrOS-10-10" is 4 times less than in pairs "steel - BrOS-10-10".

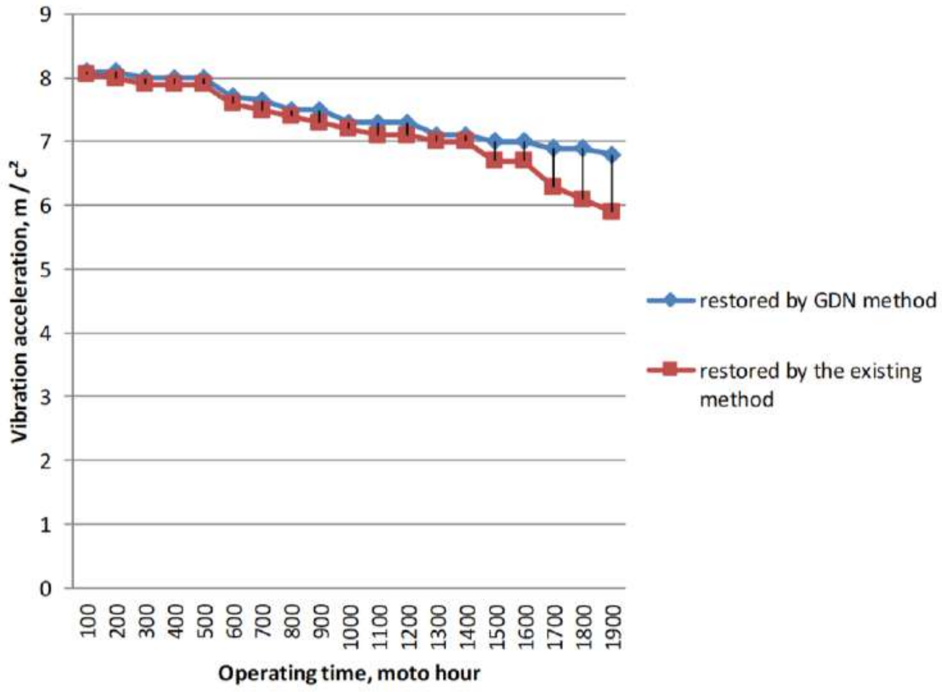
Comparative tests of turbochargers repaired by the proposed method of gas-dynamic spraying and turbochargers repaired by existing technology, on the stand for acceptance tests, showed that turbochargers repaired by the proposed method provide the main performance [12].

In fig. 3 presents the results of processing changes in the main operating parameters of the turbocharger, which affect the performance of the engine.

From the graphs it turns out that the turbocharger, repaired by the proposed technology has after running in 2000 moto-hours. the main operating parameter, the air pressure in the combustion chamber is 13.6% higher than that of turbochargers repaired by previously existing technology, and vibration acceleration increases by 12.4%. Development of the turbocharger of these characteristics allows to operate equipment without carrying out unplanned repairs [13].



a



b

Fig. 3. The main operating parameters of the turbocharger:
a – air pressure in the combustion chamber;
b – vibration acceleration

After statistical data processing, the scatter of the main performance of the turbocharger was.

During operation, the equipment worked in normal modes, without which or complaints about the instability of the turbochargers. Operating time was 900 … 2000 moto-hours. At the enterprises where the tested turbochargers were tested, they received positive recommendations for their work [14]. After testing, the turbochargers were disassembled and measured the wear of the shaft surface under the plain bearing. Mutual wear on the new turbochargers in the bearing assembly was 35% of the limit, and in the bearing assembly, restored by gas-dynamic spraying, was 15 % of the limit.

Conclusions

The analysis of literature and information materials showed that during the operation of the internal combustion engine the turbocharger is the most loaded. The most common failure is the wear of the bearing assembly of the turbocharger, their share in the total number of failures is 81%. Restoration of the bearing unit with increased wear resistance is the most urgent task. Existing methods of recovery are either expensive or do not provide the required durability. One of the most promising ways to restore the primary life of engines is gas-dynamic spraying. To reduce the coefficient of friction and increase the wear resistance of the coating, the use of copper-zinc powders brand C-01-11, applied by gas-dynamic spraying, is theoretically justified.

As a result of the conducted researches the necessary recommendations on the technique of tests of turbochargers were made. The development complements and develops the capabilities of the defect system and calibrates in terms of improving the efficiency and ease of operation of bench equipment during repair and maintenance of turbines.

Thus, the technical feasibility of using the technology of recovery by gas-dynamic spraying and the choice of copper-zinc powder brand C-01-11 for coating the rotor shaft of the turbocharger.

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Марченко Д.Д., Матвєєва К.С. Дослідження зносостійкості покриття валів турбокомпресора дизельного двигуна відновленими за допомогою газодинамічного напилення.

В статті приведено аналіз трибологічних досліджень з найбільш перспективного способу відновлення первинного ресурсу двигунів за допомогою газодинамічного напилення. Виявлено, що для зниження коефіцієнта тертя і підвищення зносостійкості покриття теоретично обґрунтовано застосування мідно-цинкових порошків марки С-01-11, нанесених методом газодинамічного напилення. Встановлено, що фізико-механічні властивості покриттів (шорсткість, мікротвердість, коефіцієнт тертя) на відновленому турбокомпресорі відповідають вимогам заводу виготівника. Коефіцієнт тертя в з'єднанні, вал ротора (відновлений порошком мідь з цинком), з підшипником ковзання (з олов'янисто-свинцевої бронзи Брос - 10 - 10) на 20 % менше, ніж у з'єднання, де вал ротора виготовлений із сталі 40. Сумарний знос в підшипниковому вузлі з відновленим газодинамічним напиленим валом ротора на 20 % нижче, ніж у вузлі, де вал ротора відновлений за базовою технологією. Розроблена технологія відновлення поверхні валу ротора турбокомпресора, під підшипник ковзання (газодинамічним напиленням), яка збільшує його ресурс на 23 % в порівнянні з базовою технологією ремонту валу ротора. Це дозволяє збільшити його напрацювання встановленою нормативно-технічною документацією на капітальний ремонт двигуна. Розроблено стенд для випробування турбокомпресорів дизелів при технології відновлення, який дозволяє визначити параметри і характеристики турбокомпресорів дизельних двигунів в різні періоди експлуатації, обкатки і регулювання. Випробування на стенді показали, що турбокомпресори з відновленими валами роторів за запропонованою технологією після 2000 мото-год напрацювання підвищують усі робочі характеристики на 13 % більше, ніж турбокомпресори, відремонтовані за базовою технологією. Експлуатаційні випробування показали, що турбокомпресори, відремонтовані за запропонованою технологією мають напрацювання на 989 мото-год більше, ніж турбокомпресори, відремонтовані за існуючою технологією.

Ключові слова: газодинамічне напилення, зносостійкість, пара тертя, турбокомпресор, технологія відновлення, інтенсивність зношування.

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Mathematical modeling of energy loss reduction in gear boxes of oil and gas technological transport

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The work is aimed at solving the problem of reducing energy losses in transmission units of hoisting machines for repairing wells. Method of rapid heating and maintaining the optimal temperature in the transmission units of lifting units by using the heat of the exhaust gases was proposed. Analysis of features of a design of transmissions of lifting installations for repair of wells is carried out. Studies of viscosity-temperature characteristics of modern transmission oils and temperature regime in transmission units have been performed. A mathematical model of energy release in transmission units during the operation of lifting units is proposed. Installed energy consumption for friction in the gears of the transmission units. Friction energy losses in bearings of transmission mechanisms of lifting units are determined. A method for reducing energy losses in transmission units of hoisting installations for well repair is proposed. Experimental studies of the implementation of the proposed method of reducing energy losses in transmission units. Dependence of power losses in the gearbox of lifting units depending on the temperature and grade of transmission oil is established. The dependence of power losses in the gearbox of the lifting unit of the UPA 60 / 80 A model depending on the temperature and grade of transmission oil is obtained. The results of calculations of fuel consumption in the gearbox of the lifting unit of the UPA 60 / 80A model with different power drives and at different temperatures of transmission oil are given.

Key words: oil and gas technological transport; hoisting installation for repair, gearbox; transmission unit; exhaust gases; heat utilization; power.

Introduction

The oil and gas complex of Ukraine and the world together with other structures includes numerous production units of oil and gas technological transport. Well-lifting installations for repairing wells have been widely used in the gas and oil industries. Mobile installations for current and overhaul of wells are widely used for repair of wells and downhill equipment and perform lifting and lowering of compressor and drill pipes, pump rods, pumps, ropes and current-carrying cable for electric pumps, gas lifts, etc.

The purpose of technological transport of oil and gas industry is to ensure the continuous operation of the main production by performing technological operations and transport work in the specified volumes, at the specified time and with minimal energy costs and costs. Therefore, the problem of reducing energy consumption for lifting plants, in particular reducing energy consumption in their transmissions, is an urgent task.

Analysis of modern foreign and domestic research and publications

Problem of improving energy efficiency is one of the main priorities of our country. And recently, under the influence of external factors, there are radical changes in approaches to the formation of state energy policy [1]: there is a transition from outdated models of the domestic energy sector, dominated by fossil fuels, to new models that maximize the use of alternative non-fossil fuels and the dominance of one type of energy production is minimized. At the same time, maximum priority is given to the use of energy from renewable sources and increase energy efficiency [2].

In order to achieve the goals set by the state energy policy, it is necessary to minimize the energy consumption of all facilities and equipment operated in the oil and gas industry of Ukraine, and in particular this fully applies to oil and gas technology transport as a whole and such an important its component as a lifting installation for repairing wells [3]. These units have energy-intensive, compared to the electric drive, diesel engines with transmissions, which requires, inter alia, the search for new ways to improve the energy performance of power drives of hoists.

For heavy-duty well repair units, standard ultra-heavy-duty automobile chassis or ultra-heavy-duty wheeled or tracked transport bases assembled from standard components - axles, wheels, transmissions, gearboxes, engines, but combined with a special frame - are used. Such wheelbases, depending on the required load capacity, have four or six axles, of which two or four are leading. The world has mastered the production of a large number of units for the repair of wells, with a capacity of 16 tons to 350 tons on wheels and tracks [4]. The most common in terms of frequency of application is the unit of current and overhaul of wells for work with open mouths. Lowering and lifting operations under pressure are performed much less often, so the units of current and overhaul of wells for this purpose are much smaller. In addition, due to the complexity of pressure operations, sometimes wells are blocked and repaired as usual. When using the same units that allow you to perform lowering and lifting operations under pressure, a complex process of silencing can be avoided [5].

Depending on the operations performed by mobile installations, their complete set can change. So, only for raising and lowering of pipes and rods the lifting installation is composed of the minimum number of knots. To use the hoisting rig for drilling, power is selected for the rotor, drilling pump and flushing system, which allows the use of the hoist for overhaul of wells. For lowering and raising electric pumps, the lift is completed with a drum for winding the cable, and for the suspension of the rods, the tower is equipped with a special gripper [6]. Hydraulic couplings and sometimes torque converters installed between the engine and the transmission are used to improve the performance and control efficiency of mobile well repair installations, as well as for more efficient pairing of drive motors in foreign-made hoists, especially large-capacity ones. In this case the transmission is called hydromechanical [7].

Hydraulic couplings or torque converters must be used for paired drive motors in foreign-made hoists of high capacity when operating on a single transmission. In this case, the total power transmitted to the transmission increases due to the self-regulation of each engine speed, which is impossible with their rigid mechanical connection. The torque converter allows you to smoothly change the speed while changing the torque. But with increasing transformation factor, the efficiency of this hydraulic machine decreases and decreases up to 0.60-0.65, which is a serious disadvantage of this method of regulation. To provide an even greater range of regulation, the drive with a torque converter of mobile well repair units can also be supplemented with a manual transmission [8].

Using of hoists for repairing wells in low temperatures entails difficulties in thermal preparation not only of the engine, pumping equipment, compressor equipment, hoisting system, but also transmission units, due to the high initial viscosity of the transmission oil in the cold period of a year. Many scientists have devoted their work to the problems of efficient operation of the transmission. Both mechanical transmissions based on automobile chassis [9] and mechanical transmissions based on tractor units [10], as well as automatic transmissions [11] were considered.

In a number of works [12, etc.], the use of forced oil supply to friction surfaces has been proposed as a way to reduce power losses in transmission units. Forced oil supply reduces losses depending on the number and depth of immersion of gears in oil, however, as shown by these studies, the greatest efficiency of oil application for rotating parts under pressure is observed from the moment of shift to stabilization of oil temperature, in addition, this method requires preheating olives. Studies by a number of authors [13, etc.] show that the temperature regime of car transmission units is one of the main factors influencing both power losses and the intensity of wear of transmission mechanisms. It is established that ensuring the optimal thermal regime of the transmission units will reduce additional fuel consumption by up to 10% and wear intensity up to 8 times. Studies [14] show that in a month with a low ambient temperature, there is an increase in failures of the ZF gearbox due to insufficient oil supply to the friction points.

For example, the number of failures of the front bearing of the secondary shaft increases by 33 %, the rear support bearing - by 20 %, and the number of failures of other parts of the gearbox increases by 12 %. Thus, the number of failures of the gearbox of this brand in general case increases by 65 %.

Based on the above data we can conclude that modern transmission oils with different viscosity-temperature characteristics do not provide torque transmission without power loss during heating of the transmission, and the optimal temperature of the transmission units is 303-313 K.

Coverage of previously unresolved parts of the overall problem

In general, the main energy losses during the operation of power drives for lifting wells are:

- internal losses in the transmission, determined by the efficiency of the mechanical or hydraulic transmission;
- internal losses in the hoist system, determined by the efficiency of the hoist system;
- internal losses in the drive engine, determined by the efficiency of the diesel internal combustion engine;
- losses due to suboptimal drilling process;

- losses due to suboptimal operation of drilling pumps;
- hydraulic losses in pipelines and fittings;
- internal losses in the winch, determined by the efficiency of the winch;
- internal losses in the rotor, determined by the efficiency of the rotor;
- losses due to the use of suboptimal fuels for the drive engine;
- losses associated with the lack of recovery of excess heat.

Transmission of the well repair unit consists of a set of couplings, shafts, chain gears, a winch and a hoist connected to the hook block. The number of speeds and their ratio are determined depending on the technology of lowering and lifting operations. In a car or tractor transport base, the standard gearbox of the transport base itself is often used as a torque converter from the engine. This scheme of lifting installation, built on mechanical transmissions, is currently the most common in Ukraine. However, comprehensive studies on reducing energy consumption in the transmission of hoists for repairing wells by using the heat of the exhaust gases to ensure rapid heating of the transmission units and maintain the optimal thermal regime has not yet been conducted.

Formulation of aims of the article

The costs of operation of technological transport account for a significant share in the cost of oil and gas products, so reducing energy consumption and cost of technological work during the operation of power drives for lifting wells - an urgent problem for oil and gas industry.

Therefore, the aim of this article is theoretical and experimental studies of ways to improve the energy efficiency of transmission units of hoisting installations for the repair of wells by using the heat of exhaust gases.

Mathematical model of energy release in transmission units during operation of lifting units

Analysis of kinematic diagrams of transmission units of hoists for repairing wells shows that they mainly consist of cylindrical and bevel gears, planetary gears, different types of bearings, seals and other elements in which there is internal mechanical and hydraulic friction and friction. rotating gears with oil bath.

Mechanical gearboxes in most cases gear less often chain with step speed control are mainly used as transmissions and power converters for hoists for repairing wells made in Ukraine, the former CIS and foreign production of small capacity.

A mathematical model has been developed to determine the energy $Q_{mp.a.}$ generated in transmission units during the operation of lifting units.

Calculation of the energy generated in the transmission units of lifting units is based on the determination of the efficiency of internal friction sources and taking into account the power transmitted through the transmission units. Based on the efficiency of the units and mechanisms that make up the unit, you can determine the amount of energy generated in the unit, according to the formula:

$$Q_{mp.a.} = \sum_i N_i \cdot t_i \cdot \left(1 - \eta_{b.mp.i}\right), \text{ J}, \tag{1}$$

where N_i – power transmitted by the i-th node of the transmission unit, Wt;

t_i – duration of operation of the i-th unit of the transmission unit, s;

$\eta_{b.mp.i}$ – efficiency of the i-th unit of the transmission unit.

Thus, the calculation of the internal energy release in the transmission is based on the functional relationship between the efficiency of the nodes of the transmission units and the power transmitted through the transmission nodes. Taking into account the fact that any transmission unit consists of elementary components and mechanisms, each of which has its own efficiency, the assessment of energy is based on the specific design features of the considered units.

The design of transmission units of hoisting installations for well repair is a set of cylindrical, bevel gears and chain gears of internal and external gearing. For example, the transmission of UPA-60/80 hoists on the KrAZ-63221 chassis (mechanical, two-band, eight-speed) consists of cylindrical spur gears (1st gear and reverse) and cylindrical helical gears (2 - 8 gears). Primary, secondary and intermediate shafts are installed in the crankcase sockets on ball and roller bearings.

Taking into account the design features, the total efficiency $\eta_{в.mp.кn}$ of the considered gearboxes is determined by the following expressions:

$$\eta_{в.mp.кn} = \eta_{zn}^{x} \cdot \eta_{gm}^{y} \cdot \eta_{mn}^{z}, \tag{2}$$

where η_{zn} – efficiency taking into account energy losses due to friction in cylindrical gears;

η_{gm} – efficiency taking into account energy losses due to hydraulic friction in the unit;

η_{mn} – efficiency taking into account energy losses due to friction in bearings;

x, y, z – the number of gears, gears in contact with the oil in the transmission unit, and bearings, respectively.

The front, middle and rear axles of UPA-60/80 hoists are a combination of one pair of bevel gears with helical teeth and one pair of cylindrical helical gears. The leading bevel gear of the front, rear and middle axles is mounted on two roller tapered roller bearings. The intermediate shaft of the axles is mounted on three tapered roller bearings.

Taking into account the design features, the total efficiency of the considered main gears in the bridges is determined by the following expressions:

$$\eta_{b.mp.gn} = \eta_{zn}^{x} \cdot \eta_{gm}^{y} \cdot \eta_{mn}^{z} \cdot \eta_{\kappa n}^{\chi}, \tag{3}$$

where $\eta_{\kappa n}$ – efficiency, which takes into account the losses in the gearing of bevel gears;

χ – the number of bevel gears in gear.

Friction energy losses in gear units of transmission unit

The value of efficiency of transmission units is determined by the viscosity of the oil used for lubrication, the gear ratios, the coefficients of friction of the gear teeth, the number of gear teeth and other factors. Typical values of the efficiency of gears at the optimum temperature are presented in table 1. It should be noted that at the optimum temperature the efficiency of gears reaches rather high values and provides effective work of transmission units with the minimum losses of energy. However, operating transmission mechanisms in conditions of negative temperatures, their efficiency becomes significantly lower. Thus, at typical winter temperatures of Ukraine (-15 … -10) °C at the beginning of the transmission efficiency is in the range of 0.55 … 0.60 [9]. In the process of moving the car as the self-heating units of the transmission, their efficiency increases after 30-40 minutes. after the beginning of the movement reaches values of 0.88 … 0.93 [9].

Table 1

Typical values of efficiency of gears at the optimum temperature

Transmission status	Average efficiency depending on the type of transmission	
	cylindrical, η_{zn}	conical, $\eta_{\kappa n}$
New	0,975	0,96
After running-in	0,98	0,97
Maximum earnings	0,99	0,98

The determining factor influencing the efficiency of transmission mechanisms is the viscosity of the transmission oil. Due to the fact that the dynamic viscosity of the transmission oil at negative temperatures varies widely, the value of the efficiency of the gears will also be variable. For gears whose dimensions are known, the efficiency can be determined by the formula [10]:

$$\eta_{зn} = 1 - 2{,}3 \cdot \mu_{mz} \cdot \gamma_{zм} \cdot \left(\frac{1}{z_1} \pm \frac{1}{z_2} \right), \tag{4}$$

where μ_{mz} – coefficient of friction in gearing;

$\gamma_{zм}$ – coefficient taken into account the displacement of the gear;

z_1, z_2 – the number of teeth of the master and slave gears accordingly.

The plus sign in expression (4) is valid for gears with external gearing, and the minus sign is valid for gears with internal gearing.

The main difficulty in using formula (4) is the analytical determination of the coefficient of friction in the gearing due to the influence of several factors simultaneously. Thus, in [10, 11] the following regularities of change of the coefficient of friction are established μ_{mz}:

- the coefficient of friction depends little on the material of the gears and the magnitude of the contact stress in the gearing;

- the coefficient of friction decreases with increasing sliding speed and rolling speed;
- the coefficient of friction increases with increasing surface temperature of the friction pairs;
- the coefficient of friction decreases with increasing viscosity of the oil.

Thus, currently the determination of the coefficient of friction in the gearing of cylindrical and bevel gears is carried out only on the basis of experimental dependences. A large amount of experimental studies with transmission oils [9, 11] allowed to derive an empirical formula for determining the coefficient of friction in gears:

$$\mu_{mz} = \mu_0 - 0{,}026\,\lg \cdot E_0, \tag{5}$$

where μ_{mz} – the coefficient of friction of the teeth at a viscosity of 1 ° E. For transmission oil is accepted μ_{mz}= 0,11 [8];

E_0 – conditional viscosity of transmission oil in degrees Engler, E.

The translation of the non-systemic unit of viscosity of oils into the unit of kinematic viscosity ν is performed by the empirical formula [7]:

$$\nu = 0{,}073E_0 - \frac{0{,}063}{E_0}, \text{ St.} \tag{6}$$

Given the fact that in the transmission units it is necessary to operate with the dynamic viscosity of the transmission oil, it is necessary to make the transition to the appropriate units Pa · s. The solution of this problem is carried out through the relationship of dynamic and kinematic viscosity. The ratio of dynamic η_{ol} and kinematic ν_{ol} viscosity of oil is characterized by expression:

$$\eta_{ol} = \nu_{ол} \cdot \rho_{ol} \cdot 10^{-4}, \text{ Pa · s,} \tag{7}$$

where ρ_{ol} – density of transmission oil, kg/m^3.

Taking into account the formula (6), expression (7) will take the following form:

$$\eta_{ol} = \left(0{,}073 \cdot E_0 - \frac{0{,}063}{E_0}\right) \cdot \rho_{ol} \cdot 10^{-4}, \text{ Pa · s.} \tag{8}$$

As a result of simple mathematical transformations (8) we obtain: $\left(0{,}073 \cdot E_0^{\,2} - 0{,}063\right) \cdot \rho_{ol} \cdot 10^{-4} - \eta_{ол} \cdot E_0 = 0$ and solve the quadratic equation.

$$E_0 = \frac{\eta_{ол} + \sqrt{\eta_{ol}^2 + 0{,}018\rho_{ол}^2 \cdot 10^{-8}}}{0{,}15\rho_{ol} \cdot 10^{-4}}\ . \tag{9}$$

Substituting (9) into formula (5) we obtain the expression for calculating the coefficient of friction in the gear:

$$\mu_{mz} = \mu_0 - 0{,}026\lg\left(\frac{\eta_{ол} + \sqrt{\eta_{ol}^2 + 0{,}018\rho_{ol}^2 \cdot 10^{-8}}}{0{,}15\rho_{ol} \cdot 10^{-4}}\right). \tag{10}$$

Then the expression for determining the efficiency of the cylindrical transmission mechanism as a function of the dynamic viscosity of the oil will be as follows:

$$\eta_{зп} = 1 - 2{,}3 \cdot \mu_0 - 0{,}026\lg\left(\frac{\eta_{ol} + \sqrt{\eta_{ol}^2 + 0{,}018\rho_{ol}^2 \cdot 10^{-8}}}{0{,}15\rho_{ol} \cdot 10^{-4}}\right) \cdot \gamma_{zм} \cdot \left(\frac{1}{z_1} \pm \frac{1}{z_2}\right). \tag{11}$$

The efficiency of bevel gears is determined by the formula [14]:

$$\eta_{кп} = 1 - \frac{\pi \cdot \eta_{mz} \cdot \zeta}{2} \cdot \left(\frac{1}{z_{np1}} \pm \frac{1}{z_{np2}} \right), \tag{12}$$

where ζ – the coefficient of the duration of the gear teeth. For bevel gears it is recommended to accept ε = 5,5 [11];

z_{np1} and z_{np2} – the number of consolidated cogs of the leading and driven gear.

The consolidated number of gear cogs is determined from the expression:

$$z_{np.i} = \frac{z_i}{\cos \varphi_i},$$

where φ_i – the angle between the generator and the axis of the initial cone of the *i*-th gear.

Then expression (12) will take the following form:

$$\eta_{кп} = 1 - \frac{\pi \cdot \eta_{mз} \cdot \zeta}{2} \cdot \left(\frac{\cos \varphi_1}{z_1} \pm \frac{\cos \phi_2}{z_2} \right). \tag{13}$$

Taking into account the dependence of the coefficient of friction in the gearing on the dynamic viscosity of the transmission oil we obtain the equation for determining the efficiency of the bevel transmission of the transmission mechanism:

$$\eta_{кп} = 1 - \left(\mu_0 - 0{,}026 \lg \left(\frac{\eta_{ол} + \sqrt{\eta_{ol}^2 + 0{,}018 \rho_{ол}^2 \cdot 10^{-8} l}}{0{,}15 \rho_{ol} \cdot 10^{-4}} \right) \right) \frac{\pi \cdot \zeta}{2} \cdot \left(\frac{1}{z_{np1}} \pm \frac{1}{z_{np2}} \right). \tag{14}$$

Friction energy losses in bearings of transmission mechanisms

To determine the efficiency of rolling bearings, it is first necessary to analyze the design of the transmission unit. For example, we will analyze the design of the gearbox of hoisting installations for the repair of wells UPA-60/80. The analysis shows that the design uses ball and roller bearings with cylindrical and tapered rollers. The efficiency of such bearings is due to rolling friction losses. The main factors influencing the efficiency of rolling bearings include the nature of the load, the viscosity of the oil and speed.

Analytical calculation of the efficiency of rolling bearings η_{ns} can be performed according to the formula:

$$\eta_{ns} = 1 - \frac{N_{bn}}{N_{bb}}, \tag{15}$$

where N_{bb} – power supplied to the drive shaft of the transmission unit on which the rolling bearing is installed, Wt;

N_{bn} – power loss in the bearing, Wt.

The power loss in the bearing is determined from a known expression:

$$N_{bn} = \frac{2\pi \cdot n_b \cdot M_{mn}}{60}, \tag{16}$$

where M_{mn} – moment of friction in the bearing, N × m;

n_b — the speed of rotation of the shaft on which the bearing is mounted, s^{-1}.

The moment of friction in rolling bearings is determined from the expression [14]:

$$M_{mn} = d_b \cdot P_n \cdot \mu_{np}, \text{ N×m}, \tag{17}$$

where d_b – diameter of a neck of a shaft under the bearing, m;

P_n – current load on the bearing, N;

μ_{np} – consolidated coefficient of friction.

The values of the consolidated coefficients of friction μ_{np} for different types of bearings that can be in the transmission units of lifting units for the repair of wells UPA-60/80 are given in table. 2.

Table 2

The values of consolidated coefficients of friction of different types of bearings

Bearing	Type	Consolidated coefficient of friction μ_{np}
Ball	Radial	0,0015…0,0025
	Radially stopping	0,002…0,004
	Stopping	0,003…0,005
Rolling	Cylindrical	0,004…0,010
	Conical	0,005…0,015

Taking into account the formulas (15 - 17), the expression for calculating the efficiency of the rolling bearing will take the following form:

$$\eta_{ns} = 1 - \frac{\pi \cdot n_b \cdot d_b \cdot P_n \cdot \mu_{np}}{30 N_{bb}} . \tag{18}$$

Then, for example, for the gearbox of UPA-60/80 hoists on the KrAZ-63221 with ball radial, cylindrical roller, and roller tapered roller bearings, which takes into account the energy loss due to friction in the bearings depending on the speed of the transmission shafts and power which is transmitted to consumers (winch, rotor, etc.) will be determined by the formula:

$$\eta_{mn} = \left(1 - \frac{0{,}001\pi \cdot n_b \cdot d_b \cdot P_n}{30 N_{bb}}\right)^a \cdot \left(1 - \frac{0{,}007\,\pi \cdot n_b \cdot d_b \cdot P_n}{30 N_{bb}}\right)^b \cdot \left(1 - \frac{0{,}01\pi \cdot n_b \cdot d_b \cdot P_n}{30 N_{bb}}\right)^c . \tag{19}$$

Experimental studies of implementation of the proposed method of increasing fuel energy

Experimental verification of the adequacy of the developed mathematical model was carried out in the laboratory of heat engines of the Department of Road Transport of Ivano-Frankivsk National Technical University of Oil and Gas based on the power drive of the diesel engine D21A1 (Fig. 1), which includes measuring equipment and gearbox stand. Given that the mechanical transmission has physical, geometric and thermal similarity, the results of the research can be extended to the gearboxes of cars and hoists.

Fig. 1. Appearance of experimental stand based on the diesel engine D21A1:
1 – converted to diesel engine D21A1;
2 – transmission;
3 – compressor K-5M;
4 – air heating device

Measuring complex based on a personal computer, an eight-channel motor tester, and chromel-copel thermocouples was used to record the temperature of exhaust gases and transmission oil. One of the sensors was installed at the outlet of the exhaust manifold (to the recuperator), the other - in the transmission units, immersed in oil. The sensitive elements of the temperature sensors (chromel-droplets) were located in the centre of the cross section of the exhaust pipe. The readings of the exhaust gas and transmission oil temperature sensors were recorded continuously.

Two temperature sensors were installed in the gearbox housing to study the temperature of the transmission oil. Sensor № 1 was installed in the lower layer of the gears of the first gear and reverse, sensor № 2 - in the upper layer of oil near the gears of the 5th gear. The drive of the primary shaft of the transmission was carried out from the diesel engine D21A1. The crankshaft of the diesel engine rotated by means of the electric motor of a direct current. To do this, the standard starter of the diesel engine D21A1 was replaced by a special DC gear motor. The motor was powered by a 14 V and 500 A power supply.

Measurement of power losses in the gearbox was performed by measuring the voltage and current on the drive motor. The study of power losses with a manual transmission included the determination of total losses (hydraulic and mechanical) to overcome the forces of resistance to rotation depending on the temperature of the transmission oil. Losses in the gearbox of the car are similar to the cost of power consumed by the motor of the installation, taking into account the power of mechanical losses lost in the drive motor. Two brands of the most widely used transmission oils were used for research: mineral TAP-15V SAE 80W-90 API GL-3 and semi-synthetic TM-5-18 SAE 75W90 API GL-5. The first oil is used in the oil and gas industry of Ukraine for transmissions of hoists manufactured in the former CIS countries, the second oil - for transmissions of hoists manufactured in the USA, Canada and European countries.

As a result of the study it was found that the power required to scroll the gearbox at an ambient temperature of 263 K at the time of starting the engine for mineral oil TAP-15V was 902 W for semi-synthetic TM-5-18 - 625 W (Fig. 2). Further scrolling of the transmission at 273 K led to a reduction in power consumption, reaching 720 W for oil TAP-15V and 540 W for TM-5-18. At an oil temperature of 303 K power losses when using oils of different grades were almost equal and amounted to 448 W and 425 W for mineral and semi-synthetic oils respectively.

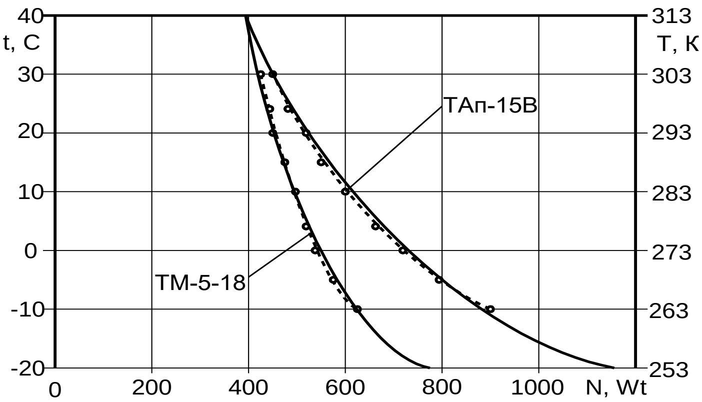

Fig. 2. Dependence of power losses in the gearbox depending on the temperature and grade of transmission oil

To verify the adequacy of the obtained analytical model in Fig. 3 were placed the theoretical dependences of the change in power loss in the gearbox from the temperature of the transmission oil. The combination of theoretical and experimental dependences showed that the maximum difference in the range of changes in power losses from temperature does not exceed 6 %. This indicates a satisfactory adequacy of the obtained mathematical model.

On the basis of the developed and confirmed mathematical model for the lifting installation for repair of UPA 60 / 80A wells on the KrAZ-63221-04 chassis calculations for power losses in a gearbox were carried out. The UPA 60 / 80A lifting unit on the KrAZ-63221-04 chassis can be equipped with a 176 kW (240 hp) engine or a 220 kW (300 hp) engine. As a result of the study it was found (Fig. 3) that the power required to scroll the gearbox at an ambient temperature of 253 K at the time of starting the engine for mineral oil TAP-15V is 14.20 kW, for semi-synthetic TM-5-18 - 9.50 kW. Further scrolling of the transmission at 273 K leads to a reduction in power consumption, reaching 9.30 kW for oil TAP-15V and 6.95 kW for TM-5-18. At an oil temperature of 313 K, power losses when using oils of different grades are almost equal and are for mineral and semi-synthetic oil respectively 4.85 and 4.80 kW.

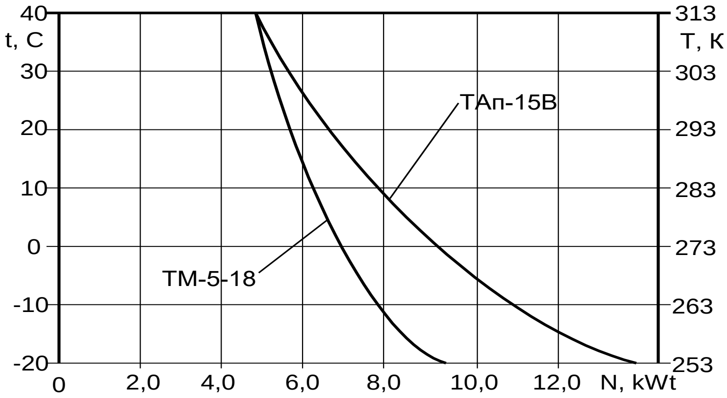

Fig. 3. Dependence of power losses in the gearbox of the lifting unit model UPA 60 / 80A depending on the temperature and grade of transmission oil

We will calculate the fuel consumption in the gearbox of the lifting unit UPA 60 / 80A with different power drives and at different temperatures of the transmission oil (Table 3).

Table 3

Results of calculations of fuel consumption in the gearbox of the lifting unit model UPA 60 / 80A with different power drives and at different temperatures of transmission oil

Temperature, K	Power consumption, kW		Engine / values of the minimum effective specific fuel consumption, g/(kWh)		Fuel consumption per gearbox drive, semi-synthetic oil, kg/h	Fuel consumption per gearbox drive, mineral oil, g/h
	Semi-synthetic oil	Mineral oil	YaMZ-238BE2	YaMZ-238VM		
313	4,80	4,85	195	214	0,93 - 1,02	0,94 - 1,04
273	6,95	9,30			1,36 - 1,49	1,81 - 1,99
253	9,50	14,20			1,85 - 2,03	2,77 - 3,03

On the main technological modes connected with drilling and repair of wells it is possible to accept average specific fuel consumption of 220 g/(kWh). Values of the minimum effective specific fuel consumption of atmospheric engines YaMZ-238VM with a capacity of 176 kW at the speed of the crankshaft of the engine 1300 min^{-1} amounted to 214 g/(kWh); nominal effective specific fuel consumption of atmospheric engines YaMZ-238VM at the speed of the crankshaft of the engine 2100 min^{-1} – 259 g/(kWh). The values of the minimum effective specific fuel consumption of supercharged engines YaMZ-238BE2 with a capacity of 220 kW at the speed of the crankshaft of the engine 1400 min^{-1} were equal to 195 g/(kWh); nominal effective specific fuel consumption of supercharged engines YaMZ-238BE2 at the speed of the engine crankshaft 2100 min^{-1} – 238 g/(kWh).

Conclusions

Studies have shown that reducing energy consumption in the transmission of hoists for repairing wells by using the heat of the exhaust gases to ensure rapid heating of the transmission units and maintain the optimal thermal regime is quite profitable.

As a result of calculations it was established (tab. 4) that the overuse of fuel necessary for scrolling of a transmission of the lifting installation of the UPA 60 / 80A model with various power drives at ambient temperature of 253 K at the moment of start for TAp-15B mineral oil makes 1, 83-1,99 kg for semi-synthetic TM-5-18 - 0,92-1,01 kg in comparison with temperature of 313 K. Further scrolling of a transmission at 273 K leads to decrease in fuel consumption having reached 0,87-0, 95 kg for oil TAp-15B and 0.43-0.47 kg for TM-5-18 compared to a temperature of 313 K.

Therefore, the obtained data showed that the high efficiency of heat transfer to the transmission units and the reduction of energy consumption in the transmission can be achieved due to the differences in the temperatures of the exhaust gases and the transmission oil. A significant stimulus for the further development of such systems is that they determine the possibility of cumulative improvement of the characteristics of the vehicle on a set of indicators. Their implementation on vehicles allows to utilize waste thermal energy and reduce fuel consumption by oil and gas technological transport.

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Микитій І.М. Математичне моделювання зниження втрат енергії в коробках передач технологічного транспорту нафти та газу.

Робота спрямована на вирішення проблеми зменшення втрат енергії в блоках передачі підйомних машин для ремонту свердловин. Запропоновано метод швидкого нагрівання та підтримки оптимальної температури в трансмісійних блоках підйомних агрегатів за допомогою тепла відпрацьованих газів. Проведено аналіз особливостей конструкції передач підйомних установок для ремонту свердловин. Проведено дослідження в'язко-температурних характеристик сучасних трансмісійних масел та температурного режиму в агрегатах трансмісії. Запропоновано математичну модель виділення енергії в блоках передачі під час роботи підйомних установок. Встановлені витрати енергії на тертя в шестернях трансмісійних агрегатів. Визначено втрати енергії тертя в підшипниках механізмів передачі підйомних агрегатів. Запропоновано метод зменшення втрат енергії в блоках передачі підйомних установок для ремонту свердловин. Експериментальні дослідження реалізації запропонованого методу зменшення втрат енергії в блоках передачі. Встановлено залежність втрат потужності в коробці передач підйомних агрегатів в залежності від температури та марки трансмісійного масла. Отримано залежність втрат потужності в коробці передач підйомного агрегату моделі UPA 60 / 80A в залежності від температури та марки трансмісійного масла. Наведено результати розрахунків витрати палива в коробці передач підйомного агрегату моделі UPA 60 / 80A з різними приводами потужності та при різних температурах трансмісійного масла.

Ключові слова: технологічний транспорт нафти і газу; підйомна установка для ремонту, коробка передач; блок передачі; вихлопні гази; використання тепла; потужність.

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Influence of lubrication on the friction and wear of car rolling bearings

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Abstract

The goal of the work is to develop computational and experimental approaches to determine the wear resistance of friction units with internal contact of cylinders with slip. The scientific novelty consists in taking into account the slip for calculating the friction path and the wear of the cylinders with internal contact and the proposed method for identifying the parameters of the wear law based on the test results. Practical value is the proposed methods to account load, slip and lubrication conditions on the resource for the design of friction units. The dependences for determining the friction path for internal rolling of cylinders have been considered. The design of an experimental setup for studying friction and wear of cylinders with slip has been proposed. Experimental studies have been carried out: paths of friction; wear of surfaces both with a key and without a key; the effect of lubricants on wear has been studied. The form of the wear model is proposed to determine the effectiveness of methods for increasing wear resistance. The method for determining the parameters of the wear law has been implemented based on the test results. The results show the efficiency of copper powder as an additive to a lubricant. It has been established that the wear of cylinders with a key is greater than the wear of cylinders without a key due to different friction paths. A practical example of determining the wear of a car hub shaft using the wear patterns is presented.

Keywords: Friction Pair, Bearing, Wear Parameters, Lubrication, Additive, Suspension Hub, Laboratory Test

Introduction

Most of the contact conditions for machine parts are internal or external cylinder contact. The internal contact of the cylindrical parts of the shaft and the hole can be either with a gap or with an interference fit. In this paper, machine units are considered in which a solid cylinder contact with a hollow cylinder in a different fit and the surfaces of the cylinders roll over each other with slipping. Such machine units include: the connection of the rolling bearing rings with the shaft; key connections; wave transmission; contact in needle bearings. The types and mechanisms of wear of cylindrical couplings operating in rolling conditions with slip are common. The friction path may be small for small gaps and large enough for large gaps. In the case of a small friction path surface damage is observed in the form of fretting corrosion. In the case of a long and continuous friction path, normal wear is observed. The definition of contact pressures is based on the contact mechanics of contact surfaces. At the same time, not enough attention is paid to methods for determining the friction path or the magnitude of slip in contact. The goal of the work is to develop computational and experimental approaches to determine the wear resistance of friction units with internal contact of cylinders with slip. In this study, it is proposed to take into account the slip values when calculating the actual friction path and the total wear of the cylinders with internal contact and the proposed method for identifying the parameters of the wear law based on the results of laboratory tests. The proposed approaches make it possible to determine the effect of load, slip and lubrication conditions on the durability of the friction unit by the wear criterion.

Literature review

Much attention is paid to the study the problems of rolling friction in cylindrical tribosystems. Thus, in [1] the effect of rolling resistance force, sample rotation speed and loading on wear is investigated. A mathematical model has been obtained that makes it possible to predict the wear of a friction pair under rolling friction conditions. The papers [2-3] study the role of the slip coefficient in the development of cracks and fatigue life of rail materials. The results show that the slip coefficient has an important influence on the wear of materials. The article [4] presents models for calculating the speed of the rollers of a cylindrical roller bearing. It was found that the slip of rollers depends on the bearing design, load, shaft speed, lubricant properties and temperature. In [5] on the basis of dynamic analysis of rolling bearings, differential equations of a cylindrical roller bearing are established. The influence of relative slip, sliding friction coefficients and residual stresses on the distribution of shear stresses are analyzed in paper [6]. In article [7] a bearing with turbulent lubrication is considered. The simulation results are in good agreement with the experimental data. In [8] it is noted that the powder lubricants are effective, despite the increased friction indicators compared to liquid oils. This article [9] provides an equation for calculating the frictional moment of a dry lubricated tapered roller bearing, which takes into account the misalignment of the rollers. A torque model to optimize bearing design has been developed. The papers [10-13] consider the modeling of friction and wear during rolling with slip. Thus Numerical simulation of single surface irregularities passing through lubricated rolling contacts during sliding is carried out in [10]. Phenomena have been found to explain how the roughness moves through the contact and lubricates it by sliding. In [11-12] new models of rolling bearing durability are proposed. Load capacity equations have been written for both point and line contacts. The value of the empirical constant of the ball bearing has been determined based on the regression analysis of the experimental data. In the article [13], a method for determining the equivalent stiffness for bearing devices with self-elimination of the gap has been implemented. In addition, a linear model has been proposed, taking into account the effect of contact on the vibration mode and rotor frequency. Simulations and experiments have been performed to test the effectiveness of the stiffness identification method. In works [14-15] models of bearing wear are proposed and methods of identifying the parameters of these models are described.

Thus, the determination of the amount of slip during rolling of cylinders is one of the important problems of contact mechanics and tribology in general. Without knowing the amount of slip, it is impossible to simulate the wear of rolling bearings. Known studies have not provided a calculation method for assessing the amount of slip during cylinder contact. Taking this into account, the only correct thing is to determine the slip value only experimentally.

Methodology for investigating slip with internal cylinder contact

The problem of describing the rolling slip process is one of the most difficult in contact mechanics. Note that almost all variants of rolling mechanisms relate to external contact of cylinders or balls. Let us consider the case of internal rolling of cylinders that are not connected to each other in the tangential direction (Fig. 1*a*).

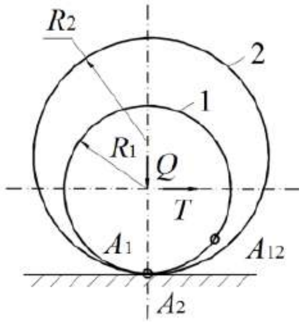


Fig. 1. Cylinder contact during rolling

Under the action of the forces Q and T, the outer cylinder rolls along a plane. At this time, the inner cylinder with its outer surface of radius R_1 rolls along the inner surface of radius R_2 of the outer cylinder. Let the outer cylinder, when rolling, make a full revolution $\varphi=2\pi$ and point A_2 returned to its original position, having passed the path $S_2=2\pi R_2$. During this time, the inner cylinder, when rolling along the inner surface of the outer cylinder, will travel the path $S_1=2\pi R$ and will not return to the starting position A_1. In this case, the point A_1 will not reach the initial position by the amount of the path length ΔS:

$$\Delta s = 2\pi R_2 - 2\pi R_1 = 2\pi\Delta\,, \tag{1}$$

Next, we consider the conjugation with the gap Δ of the shaft of radius R_1 and hollow cylinder of radius R_2, connected by internal gearing with the number of teeth z (Fig. 1*b*). When the cylinders are rolling internally, the teeth prevent the cylinders from sliding as in free rolling. The friction path is distributed between the teeth. When turning one tooth, the slip will be by the value:

$$s_1 = 2\pi\Delta / z. \tag{2}$$

With internal rolling of the shaft over a hollow cylinder, connected by a key. This connection is similar to the sliding of the teeth at $z = 1$: $s_k = 2\pi\Delta$. Note that formulas (1-2) determine the total amount of slip in the considered rolling section. In fact, there is continuous sliding throughout the entire section. Thus, it has been found that the main role for the type and value of slippage of the coupled cylinders is played by the gap between the shaft and the hollow cylinders.

In some cases, cylinders or shafts in machines rotate, and their surfaces roll over each other. As mentioned above, there are two possible cases: the inner and outer cylinders are connected by a key or pin type device; the cylinders are not connected in the circumferential direction and roll freely over each other. It is known that both in one and in the other cases, the interaction of the cylinders is worn out. Let us consider the method of testing for wear and identification of the laws of wear with internal contact of the cylinders. Experimental studies were carried out on a setup, the design of which is shown in Fig. 2.

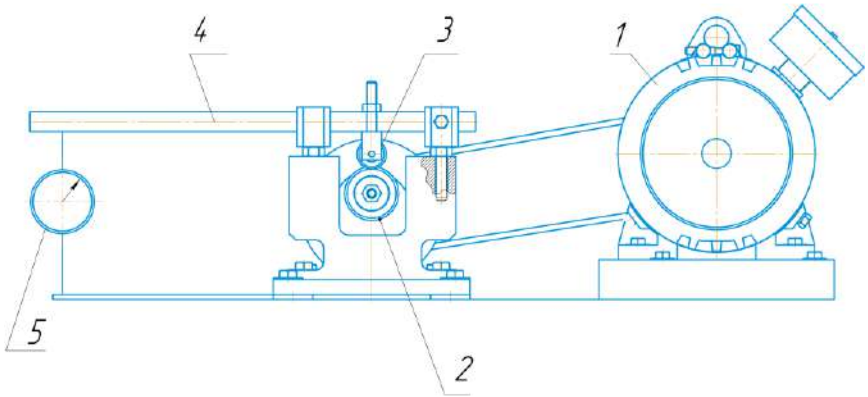
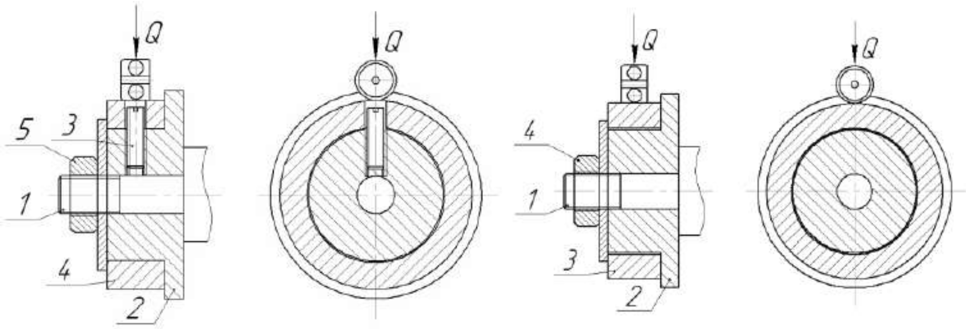


Fig. 2. Experimental setup

On the basis of the installation are located: engine 1 and working unit 2, interconnected by a V-belt transmission. The load is transmitted through the bearing 3 by means of a lever 4. The the load is controlled by an indicator 5. The tests were carried out for two schemes of internal contact of the cylinders: connecting the cylinders with a key and without a key (Fig.3).

Ring 4 (52100 steel) was put on the bushing 2 (1045 steel) and fixed with a pin 3 (Fig.3*a*). The connected parts were mounted on the shaft 1 and fixed with nut 5. Using a ball R=2.67 mm, a hole was pressed on the outer side of the sleeve 2 to register the amount of sleeve wear. An IZA-2 optical device was used to measure the indentation diameter d. The indentation depth was determined by the formula:

$$h = d^2 / 2R. \tag{3}$$



a *b*

Fig. 3. Working unit of the installation: *a*- with a key; *b*-without key

The difference between the initial indentation depth h_0 and the indentation depth after testing was taken to be equal to the linear wear of the sleeve surface u_W. The friction path of two cylinders has been determined by the dependence:

$$s_1 = 2bnt, \tag{4}$$

where b is the contact strip half-width;
n is the part rotation frequency;
t is the time of testing.

The dimensions of the contact area with internal contact of two cylinders according to the Hertz formula:

$$b = 1.128\left(\frac{Q}{lE}\frac{R_1R_2}{R_2 - R_1}\right)^{1/2}, \tag{5}$$

where Q is the part load;
R_1 is the outer radius of the sleeve;
R_2 is the inner radius of the ring;
l is the sleeve width;
E is the modulus of elasticity of the bushing material.

The tests were carried out at two loads of 100N and 200N. To study the effect of a lubricant on wear, three lubrication modes were used: dry contact; Fiol-3; Fiol-3 with copper powder additives (1% by weight). The results of tests for a load of 100 N without lubricant are shown in Table 1. The tests were carried out with the following initial data: R=2.67mm; n= 650 rev^{-1}; R_1= 35mm; R_2= 35.5mm; Δ=0.5 mm.

Table 1

Test results without lubrication at a load of 100 N

t, min	d, mm	h, mm	u_w, mm	S, mm·10^5
-	1.929	0.20	-	-
100	1.83	0.159	0.008	0.3
200	1.85	0.155	0.012	0.6
300	1.83	0.152	0.015	0.9
600	1.79	0.15	0.02	1.8
900	1.75	0.144	0.025	2.5

The results have been obtained similarly for different modes of lubrication and loading. The dependence of wear on the friction path conditions is shown in Fig. 4 (Q=100N).

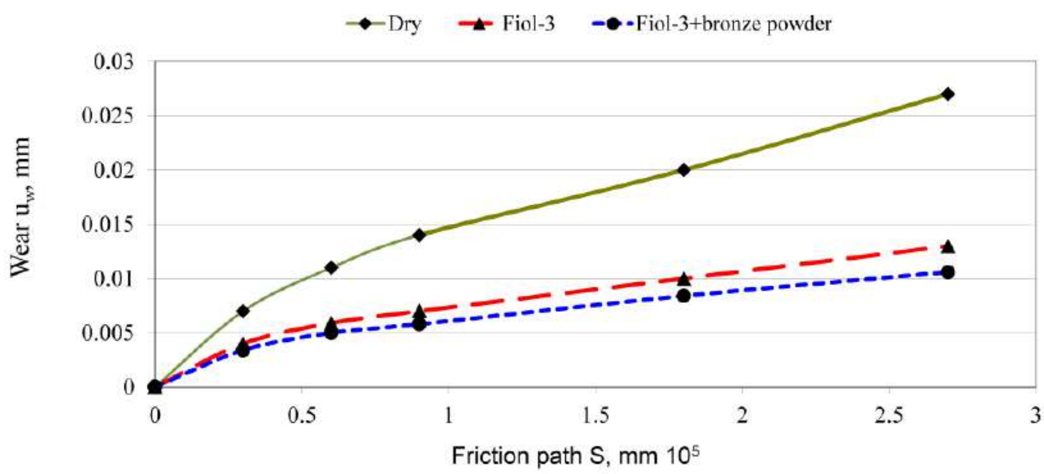


Fig. 4. Wear on the friction path with non-free contact of the cylinders (Q = 100 N)

Wear tests with free contact of the cylinders were carried out according to the scheme in Fig. 3b also for two loads and three modes of contact lubrication.

The tests were carried out with the initial data: R=2.67mm; n= 650 rev^{-1}; R_1= 34.7 mm; R_2= 35.7 mm; Δ=1 mm. The wear on the friction path with free contact of the cylinders for various lubrication conditions is shown in Fig. 5 (Q = 100 N).

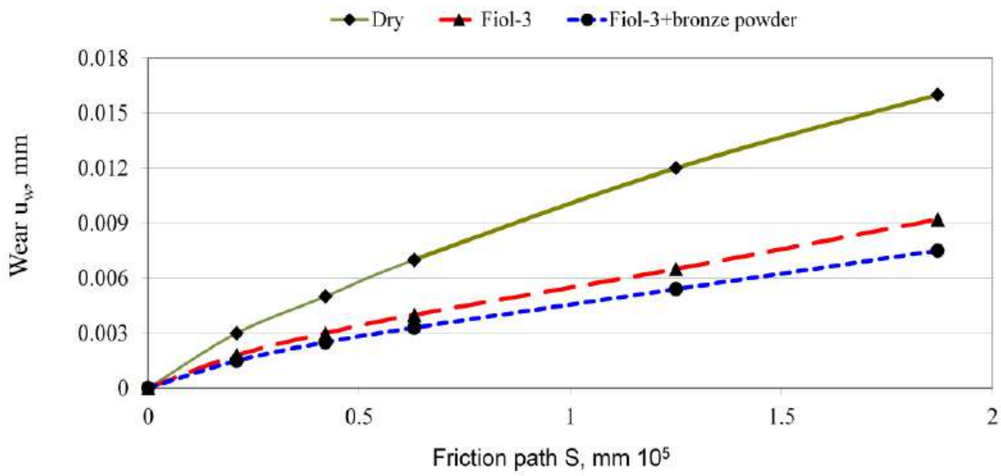


Fig. 5. Dependences of wear on the friction path with free contact of the cylinders (*Q* = 100 N)

Results of estimated wear resistance at internal contact of cylinders with slip

For the analysis of the wear of machine parts, you can use a simple wear model:

$$u_w = k_w \sigma^m S, \tag{6}$$

where u_w is the current wear;
k_w, m are the parameters of the wear model determined from the experiment; σ is the contact pressure;
S is the friction path at wear points, determined by calculation or from experiment.

To determine the parameters of the wear law (6), it is proposed to use the obtained results of tests for cylinder wear under various lubrication modes. The wear law parameter m is determined by the results of the wear values and contact pressures obtained at two values of the external load: 100 N and 200 N according to the dependence:

$$m = \frac{\lg(u_{w1}/u_{w2})}{\lg(\sigma_1/\sigma_2)}, \tag{7}$$

where Δu_{w1}, Δu_{w2} are the values of the wear of the part at various loads Q;
σ is the maximum contact pressure at a given load according to the Hertz formula:

$$\sigma = 0.418\left(\frac{QE}{l}\frac{R_2 - R_1}{R_1 R_2}\right)^{1/2}, \tag{8}$$

Wear model parameter k_w:

$$k_w = \frac{u_{w2}}{\sigma^{m_2} S_2}. \tag{7}$$

The results of wear tests and parameters of the law of wear of cylinders for various lubrication conditions are presented in Table 2.

Table 2

Results of studies of cylinders wear with slip

Lubricant	With key			Without key		
	Dry	Fiol-3	Fiol-3 + bronze	Dry	Fiol-3	Fiol-3 + bronze
b (100N), mm	0.23	0.23	0.23	0.16	0.16	0.16
b (200N), mm	0.32	0.32	0.32	0.23	0.23	0.23
S_{900}(100N), mm·10^5	2.7	2.7	2.7	1.87	1.87	1.87
S_{900}(200N), mm·10^5	3.75	3.75	3.75	2.7	2.7	2.7
σ_{100}, MPa	15			19.5		
σ_{200}, MPa	199			285		
m	1.84	2	2.04	1.8	1.76	1.78
Kw, MPa^{-1}·10^{-9}	18.4	8.85	7.2	7.1	4.3	3.52

Thus, in contrast to the known approaches, a theoretical and experimental method is proposed for determining the wear of cylinders with internal contact, taking into account the friction path from the slip value. A method for identifying the parameters of the wear law based on the results of laboratory tests is also proposed. For the further development of ideas about the mechanism of slip, it is necessary to carry out experiments to determine the slippage of uncoupled cylinders and to study experimentally the influence of interference and clearance on the amount of slippage and wear.

The task is to determine the wear of the hub shaft in the connection of the rolling ball bearing of the suspension hub of the vehicle. The initial data: load on one wheel is 3000 N; diameter of the bearing ring is d=34 mm; ring thickness is 5 mm; the clearance Δ = 0.05 mm. The slip in the contact is taken to be equal to the value of the relative clearance: $\psi=\Delta/R=0.003$. The friction path in one wheel revolution: $S1=2\pi R\ \psi=0.32$. With a wheel diameter of 500 mm, the wheel makes an 640 revolutions per 1 km of run. Then the friction path for 100,000 km of run will be: S=0.32·640·100,000=2.05·107 mm. The calculation of the hub shaft wear was carried out to the formula (7). Two modes of lubrication were considered: Fiol-3 and grease with copper powder. The parameters of the wear (7) were taken from Table 2. The contact pressure was taken to be 5 MPa. The results of calculating the wear of the hub shaft are shown in Fig. 6.

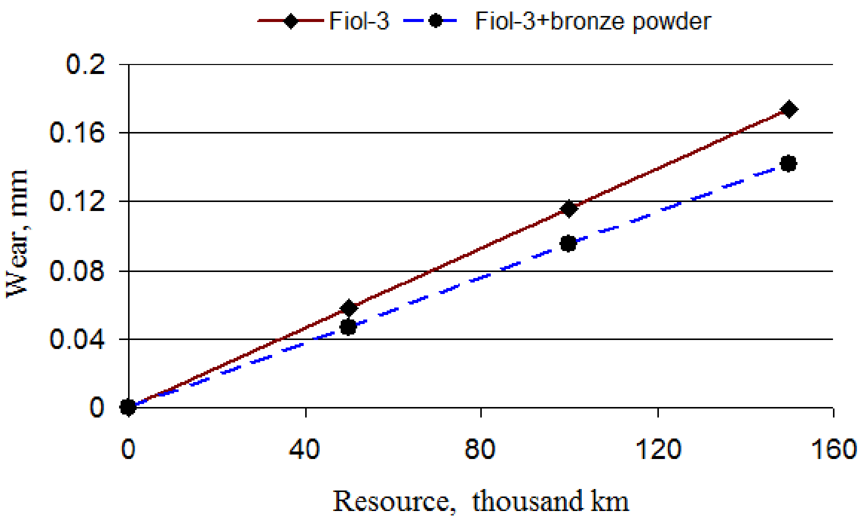


Fig. 6. Dependence of the hub shaft wear on the vehicle resource

The calculation results show the efficiency of using copper powder as an additive to a lubricant.

Conclusions

Developed a test procedure for wear during internal rolling of a cylinder.The developed technique is used to assess the effectiveness of lubricants and additives from bronze powder to increase the wear resistance of the interface. As a result of tests, it has been found that: wear of cylinders with a key is 1.35 times more than the wear of cylinders without a key, which is explained by a correspondingly large friction path; lubrication with Fiol reduces the wear of the joint by 1.81 times; lubrication with Fiol with the addition of bronze powder reduces wear in comparison with lubricant without the addition of additives by 1.2…1.8 times; the overall reduction in wear when using Fiol with bronze powder is estimated 2.3…2.1 times.

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Диха О.В., Маковкін О.М., Посонський С.Ф. Вплив змащування на тертя та знос підшипників кочення автомобіля

Метою роботи є розробка обчислювального та експериментального підходів для визначення зносостійкості агрегатів тертя з внутрішнім контактом циліндрів із ковзанням. Наукова новизна полягає у врахуванні ковзання для розрахунку шляху тертя та зносу циліндрів із внутрішнім контактом та запропонованого методу визначення параметрів закону зносу за результатами випробувань. Практичне значення мають запропоновані методи впливу навантаження, ковзання та умов змащення ресурсу для проектування агрегатів тертя. Розглянуто залежності для визначення шляху тертя для внутрішнього кочення циліндрів. Запропоновано конструкцію експериментальної установки для вивчення тертя та зносу циліндрів із ковзанням. Проведені експериментальні дослідження: шляху тертя; зносу поверхонь як із шпонкою, так і без шпонки; вивчено вплив мастильних матеріалів на знос. Форма моделі зносу пропонується для визначення ефективності методів підвищення зносостійкості. За результатами випробувань реалізовано метод визначення параметрів закону зносу. Результати показують ефективність мідного порошку як добавки до мастильного матеріалу. Встановлено, що знос циліндрів із ключем більший, ніж знос циліндрів без ключа через різні шляхи тертя. Наведено практичний приклад визначення зносу валу втулки автомобіля за допомогою шаблонів зносу.

Ключові слова: пара тертя, підшипник, параметри зносу, змащення, добавка, підвісна втулка, лабораторне випробування