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INVESTIGATION OF PATTERNS OF WEAR OF SURFACES OF CONNECTION "SHAFT-SLIDING BEARING"

Problem statement and its connection with important scientific or practical tasks. The search for technical solutions aimed at improving the reliability of connections of parts of ship's technical means (STM) is a topical issue in the design and operation of ship systems. The necessary level of safety and reliability is laid down at creation of ship knots and units, in the course of operation only support of the parameters set at designing due to the correct maintenance and repair that is connected with time and financial expenses is possible.

One of the ways to increase the reliability of joints of STM parts is to increase their service life by increasing their wear resistance.

There is a lot of research work aimed at improving the anti-wear properties of the surfaces of the connection "sliding bearing", their geometry, materials and technologies of their manufacture.

Design experience shows that changing the profile of the contact surfaces several hundredths of a millimeter changes the contact planes and pressures by 2 ÷ 5 times and significantly affects the distribution of internal force factors.

The purpose of this study is to determine the patterns of wear of the surfaces of the connection "shaft-sliding bearing". [1]

Key words: ship technical means, microgeometry of a surface, frictional fatigue, connection "shaft-sliding bearing", radius of curvature of a wave.

The basic material of research

Consideration of a physical wear model.

Consideration of wear in existing connections "Sliding shaft bearing" is carried out, as a rule, on the basis of a physical wear model.

Studies [3] show that an important role in wearing material plays not only the hardness of the material and load on a steam friction, but also the elastic properties of the material, the mode of operation of the part (speed, temperature), external conditions (lubricants, the environment) and structural features of the frictional node (diameter, bearing length, etc.).

Wear is characterized by a linear intensity of wear (dimensionless magnitude)

$$u_i = \frac{u}{l}, \tag{1}$$

where l is rubbing path, mm; u - the value of wear (mm) on the path of friction l.

Intensity may vary wide range from 10^{-3} to 10^{-13}. Based on the accumulated experience [5] distinguish 9 classes of wear intensity. The established classes of wear intensity are combined by the deformed solid body by the main types of contact interaction of friction surfaces [6].

Basic equation to calculate wear intensity in the case of multiple contact

$$u_i = \alpha\, k_1 \frac{F_a}{nF_r}\sqrt{\frac{h}{r}}, \tag{2}$$

where k_j is a coefficient determined by the geometric configuration and the arrangement of the height of unit inequalities on surfaces of solids (usually $k_j = 0.2$);

$\boldsymbol{\alpha} = A_A / A_R$ - coefficient of overlap; A_A - nominal contact area; A_R - actual contact area; F_A - nominal pressure on the contact; F_R - actual pressure on contact; n is the number of load cycles, which leads to the separation of the material volume from the surface layer.

But this formula is uncomfortable to use, since in this case, the intensity of wear depends on the number of load cycles. Therefore, with an engineering calculation for wear, use the following formula for elastic contact

$$u_i = \alpha\, k_1 2^{\frac{1}{2\nu}} F_a F_c^{\frac{1}{2\nu}} F_r^{t_y - 1 - \frac{1}{2\nu}} \Delta^{0,5} \left(\frac{k\ f_m}{\sigma_0} \right)^{t_y}, \tag{3}$$

where t_y is a parameter of friction fatigue curve (depends on the material of the contact parts and the type of grease, determined experimentally and takes the value of $t_y = 2 \div 10$); ν - parameter of the degree approximation of the curve of the support surface; k - coefficient characterizing the tense state on contact, and dependent on the nature of the material; for fragile materials k = 5, for highly elastic k = 3; f_m - mechanical component of friction coefficient; σ_0 - destructive stress with one-time stretching; F_c - contour pressure on contact.

Contour pressure on contact and actual pressure is determined by formulas

$$F_c = 0{,}2E^{0{,}8}\left(\frac{H_B}{R_B}\right)^{0{,}4}F_a^{0{,}2}; \tag{4}$$

$$F_r = 0{,}5E^{\frac{2\nu}{2\nu+1}}\Delta^{\frac{\nu}{2\nu+1}}F_c^{\frac{1}{2\nu+1}}, \tag{5}$$

where R_B and H_B - radius and micronage wave height.

Considered calculation dependencies allow to trace the influence of various factors for wear. For many types of processing, including for the finished surfaces, $\nu = 2$.

For non-ceremonial surfaces, the specific load is nonlinearly affected by wear, and more significantly on the surface with a small area of contact, that is, without wavy. Wavy of surfaces significantly reduces the nonlinear nature of this dependence. For the finished surfaces, the intensity of wear is directly proportional to the specific load.

The modulus of elasticity significantly affects the wear of the material, and, for materials with the same strength to the gap its increase leads to an increase in wear. A larger range of changes in the degree corresponds to the contact of a rough surface without wavy, and more absolute value of this indicator takes place for a rough and wavy surface. The unambiguous relationship between the modulus of elasticity and the intensity of wear experimentally to establish it is difficult to establish, because there is a connection between the elastic modulus and friction properties, in particular, with the coefficient of friction, as well as with strength characteristics.

The property of imperfect elasticity takes into account the coefficient of hysteresis losses, whose knowledge is important for predicting the wear of the finished surfaces.

Increasing the absolute values of strength characteristics always positively affects wear resistance. The more σ_0, the stronger material with a single rupture, and the more t_y, the greater the number of cycles is required to separate wear particles.

The intensity of wear is strongly dependent on the coefficient of friction and from stronger properties. This connection is ambiguous, since the coefficient of friction depends on the elastic properties of the material, roughness of surfaces, specific load and parameters characterizing the molecular interaction on the contact.

The effect of microgeometry of the surface to wear is also quite significant. Neglecting roughness can lead to an error in several orders of magnitude to determine the intensity of wear.

Thus, formulas (1) - (5) are empirical character and contain many interrelated parameters, practically each of which has a fairly large spread of values. This leads to numerous mistakes in determining the wear of elastic-dissipative mechanical systems and indicates the imperfection of the physical model of wear.

Transition to a statistical wear model

Many ship mechanical systems during operation undergo a very large number of load-discharge cycles ($10^8 \div 10^{10}$), it is impossible to take into account the physical model based on the physical model, so it is proposed to use a statistical model.

The developed model of wearing a wear of a complete picture of wear for 100 ÷ 200 steps to change the load. In this case, the actual picture of external forces during the entire work time is replaced with a cycle of 5 ÷ 10 passes from the minimum load to the maximum and vice versa, taking into account the probability of each load level. In order to formalize the load of the system, it is necessary to conduct its field tests at various loads, based on which the graph of the load dependence on time (Fig. 1) is being built.

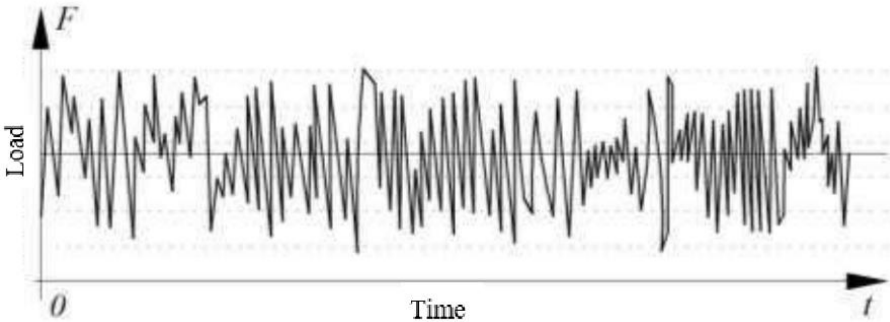

Fig. 1. Schedule of load dependence on system time

Based on this graph, the entire range of load changes can be divided into several degrees, and determine how often the graph intersects a line of one or another load level. The total number of such intersections will significantly exceed the number of load cycles. Now it is possible to build an empirical frequency distribution function that determines the dependence of the relative frequency of loading from the load level:

$$p_i = \frac{n_i^H + n_i^P}{\sum_k (n_k^H + n_k^P)}, \tag{6}$$

where k is the load number; n_i^H, n_i^P - the number of intersections of the i-th level of load when loading the system and its unloading; $\sum(n_i^H + n_i^P)$ - general number of intersections.

Using the formula (6) and a graph (Fig. 1), you can determine the number of loads corresponding to the given time of operation, and to construct a load distribution diagram that practically always represents a normal distribution law (Fig. 2).

The amount of wear for one finite element is determined by the formula

$$w_{эл} = \frac{w_i}{A} \sum_{i=1}^{k} (F_i^H n_i^H + F_i^P n_i^P), \tag{7}$$

where A is the area of the contact surface of the finite element; F_i^H, F_i^P - friction forces when loading and unloading; w_i - intensity of wear.

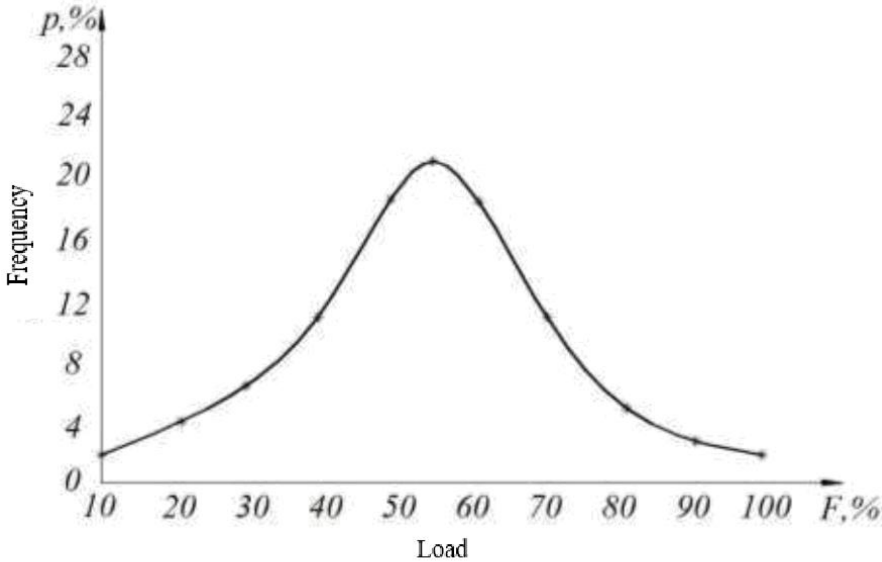

Fig. 2. Load statistical distribution

Given the frequency distribution function (6), formula (7) acquires a look

$$w_{эл} = \frac{w_i N_{цикл}}{Ap_{cp}} \sum_{i=1}^{k} (F_i^H n_i^H + F_i^P n_i^P) p_i \,. \qquad (8)$$

Thus, the proposed statistical wear takes into account the history of the system's load, which is definitely a necessity in connection with the non-conservative nature of the friction forces and change their direction with cyclic load.

Conclusions

1. In mathematical terms, a characteristic feature of contact problems is that they are reduced to the study of boundary problems for systems of

differential equations of mechanics of a continuous medium with mixed boundary conditions. Analytical decisions are obtained only for a very limited range of tasks, which resulted in widespread use of numerical methods and, in the first place, the method of finite elements as the basic method of most existing computing systems.

2. Formulas that describe the physical model of wear, are empirical character and contain a lot of interdependent parameters, practically each of which has a large solution of values. This leads to numerous mistakes in determining the wear of systems "Sliding shaft bearing" and indicates the imperfection of the physical model of wear.

3. When cyclic loads you need to use the proposed statistical wear of wear, which, unlike the physical model, takes into account the system of loading the system, which is definitely necessary in connection with the non-conservative nature of the friction forces and change their direction for cyclic load.

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