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STUDY ON THE DETERMINATION OF THE TECHNICAL RESOURCES FOR TOOTHED GEAR MECHANISMS OF MARINE AND AVIATION COMMUNICATION SYSTEMS

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Abstract— The reason for failure of most moving parts of technical systems in the machine-building and electronics industry, transportation and household equipment can be attributed to the wear and tear of their working surfaces, which are in relative motion. Therefore, the forecasting of the technical resources for toothed gearing is done by examining failures, determined in accordance with the wear criterion and involves the use of methods for calculating the wear of gear teeth (metal or plastic).
In this article, we propose the use of formulas, incorporating the parameter of intensity of wear due to friction that, even under external stationary conditions, appears to be a variable for the studied interval of time. The subject of the study is a toothed gearing of a communication system of the type P-859, which is used in marine and aviation transport systems.

Index Terms— forecasting of technical resources, gear transmission mechanisms, wear intensity

1 Introduction

THE wear intensity of the toothed gear mechanisms J, is determined by the ratio between the value of their wear Δ_H, and the length of the friction path L_{TP}, i.e. [1; 2; 3; 4]:

$$J = \frac{\Delta_H}{L_{TP}} \quad (1)$$

The use of this parameter J (that in general changes dynamically by nonlinear law) introduces an undefined error in the calculation of the toothed gear mechanisms, because the wear on the gear tooth profile is assumed to be uniform in trivial literature [1; 4].

In operation [4], it has been shown that in the tooth engagement process, the sliding velocity changes at different points of the profile, and in the engagement strip, it equals zero. In view of the fact that the sliding speed happens to be one of the main parameters, determining the magnitude of the wear, then the wear for each point of the section of the tooth profile shall be a variable quantity. Under working conditions [5] as a wear parameter is accepted the wear rate, which is generally expressed as a ratio between the magnitude of wear of the toothed gear and the time interval during which the wear occurs. Therefore, it is necessary to select a parameter that will more accurately characterize the wear process.

Solution of the problem: If exactly identical toothed gears (called standardized toothed gears) are manufactured from different materials, then under identical testing regimes, their wear rate will be different. This is determined by the properties of the materials used, as the wear rate of the standardized toothed gears is referred to as standardized base rate. It is an accurate characteristic of the materials used for the manufacturing of the gears and can serve as a parameter, defining the wear process.

As it is well known from the theory of operational reliability, the wear of the parts can be divided into three distinct stages: development stage, normal operation and stage of wearing. From the point of view of the technical resources T_{PKE}, the period of normal operation of the toothed gear mechanisms that spans until the end of the technical operation (TE) and the failure-free wear of the components of the technical systems (TC), is of particular interest. During this time period, the technical resources T_{PKE} are determined by [6], according to:

$$T_{PKE} = \int_0^{T_{OTP_i}} P_{БP}(t).dt \quad (2)$$

where: T_{OTP_i} is the time interval for performing of i - number of overhauls and current repairs; $P_{БP}(t)$ - probability for failure-free operation of the technical system (TC) (in this particular case – a marine communication system).

The probability of failure-free operation of the TCs is in turn determined by the "Basic law of reliability" for restorative technical systems as defined in literature [6]:

$$P_{БP}(t) = \exp\left[-\int_0^{T_{PKE}} \omega(t).dt\right] \quad (3)$$

where: $\omega(t)$ is the intensity of the failure rate of TC (in this case, a recovered toothed gear mechanism of a marine communication system).

In the field of normal wear and tear of toothed gears, according to the theoretical and experimental tests performed in [3], the wear intensity and wear rate are constant and do not depend on operating regimes.

Consequently, if we limit ourselves to examining only the normal operation of the toothed gear mechanism of P-859, shown in Fig. 1 [7], then the wear process can be represented as a straight line, inclined to the abscissa (x-axis) at an angle that depends on the quality of the work as well as the wear rate. [2].

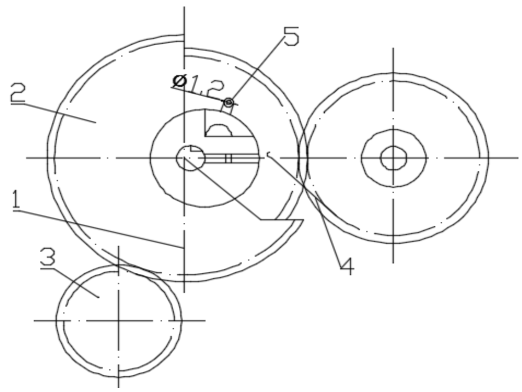

Fig. 1. Toothed gear mechanism of a marine and aviation communication system P - 859

The beginning of the line corresponds to the end of the development stage, while its end corresponds to the transition from normal operation to the period of active aging ("catastrophic" wear).

The linear nature of the wear of the toothed gear components allows, by virtue of taking consistent measurements of the thickness of the worn-out gear teeth during the course of a fixed number of cycles, to determine the base wear rate K_{uS} according to the formula:

$$K_{uS} = (S_{i-1} - S_i).\Delta N \tag{4}$$

where: S_{i-1}, S_i is the current value of the teeth thickness, measured along the chord of the gear (as shown in Fig. 2), respectively at the beginning and end of the testing; ΔN - number of cycles of the gear wheel

In the case of gears, as the unit of measurement of T_{PKE}, is used the "cycle", which is equal to one revolution of the gear wheel. Since the wear rate of the gear teeth, occurring during one working cycle of the gear mechanism is negligible, it can not be technically measured by the modern measuring instruments. By taking into consideration the linear nature of the gear teeth wear, it is possible to determine its wear value over several cycles ΔN .

During the time period of normal operation of the toothed gear mechanism of the P-859, as shown in Fig.1, the values of the wear rate K_{uS} remain approximately the same. The change curve of these values can be subjected to statistical processing and analysis, during which the following parameters of standardized wear rate K_{uS} are obtained:

Mathematical expectation $\overline{K}_{u0}$ calculated according to the formula

$$\overline{K}_{u0} = \frac{1}{L}\sum^{L} K_{ui} \tag{5}$$

Root mean square deviation σ_K defined by:

$$\sigma_K = \sqrt{\frac{1}{L-1}\sum^{L}\left(K_{ui} - \overline{K}_{u0}\right)^2} \tag{6}$$

Coefficient of variations ν of the base wear rate K_{uS} as determined according to the formula:

$$\nu = \sigma_K / \overline{K}_{u0} \tag{7}$$

The base wear rate K_{uS} can be calculated in different sections of the teeth of the respective gear in the transmission, as demonstrated in Fig. 1; however, this is not absolutely necessary. The main purpose is to determine the particular section, where the wear is at its maximum, since the dynamics of motion transmission by virtue of a toothed gear is influenced by the maximum wear rate [2]. According to the conducted research, as reported in literature [5; 6], the greatest wear occurs at a distance of (0,3÷0,5) mm from the teeth tips (m – module of engagement).

It is natural to find in the specific mechanisms of the toothed gear differences from the standardized structural and technological parameters; the conditions and regimes of operation of their wheels may be completely different from those under which the standardized toothed gears have been tested. Therefore, the wear rate differs as well from the base rate. The taking into consideration of the structural, technological and operational features is accomplished by the introduction of corresponding coefficients A_K, A_T и A_E, determined by virtue of the multifactorial experiment [4]. In this way, the wear rate of the particular gear shall be calculated as follows:

$$K_И = \overline{K}_{u0}.A_K.A_T.A_E \tag{8}$$

The availability of wear parameters, obtained as a result of the testing of the toothed gears, allows for the forecasting of the necessary technical resources (TP). According to [2], the wear parameters of the gear mechanisms under normal conditions, i.e in the event of energy equilibrium, practically do not depend on the friction mode: load, speed, temperature, etc. If the wear parameters are calculated for such regimes, then the TPs are determined by the threshold limit value of wear of the teeth and the rate of this wear, dependent on the wear parameters, load, structural and technological characteristics and operating conditions. Taking into account the fact that the destruction of one tooth results in the destruction of the whole gear, the formula for calculating its TP (measured in number of cycles) will look as follows:

$$N = \frac{[\Delta_u]}{(K_{u0} + u_p \sigma_k).(q + u_p \sigma_q).A_K.A_T.A_E} \tag{9}$$

where: $[\Delta_u]$ - limit value of wear of the teeth along the considered section (its determination is given in [4]); u_P - quantile of normal distribution; q - normal pressure at the contact area of the conjugated teeth (the methodology for its calculation is mentioned in [4]); σ_q - root mean square deviation of the normal pressure at the contact of the conjugated teeth.

In the technical operation and maintenance of toothed gears, different measuring methods are used in order to determine their wear [2]. In the case of testing the wear of the toothed gear mechanisms, shown on Fig. 1 during the process of technical operation of the P-859 system, it is appropriate as deemed by the authors, to use the method of normal. According to this method, the base rate of wear for the normal K_{uW}, for the size of the roller K_{uM} and wear rate for the mass of the wheel K_{uG}, are calculated as follows:

$$K_{uW} = (W_{i-1} - W_i)/\Delta N \tag{10}$$

$$K_{uM} = (M_{i-1} - M_i)/\Delta N \tag{11}$$

$$K_{uG} = (G_{i-1} - G_i)/\Delta N \tag{12}$$

where: W is the value of the normal; M - size of the roller; G - mass of the gear (the indices correspond to the operating time of the wheel over the course of N_{i-1} and N_i cycles).

The wear of the toothed gear also regulates the traction of the wheels in the gear mechanism in the course of the P-859's technical operation. This dictates the necessity to determine the relationship between the wear rates K_{uG} and K_{uS}. This relationship is defined by using a formula for the mass G_i of the worn-out material of the gear wheels in the process of operation [5]:

$$G_i = \Delta_i.b_W.L_{TP}.\gamma \tag{13}$$

where: γ - density of gear material; L_{TP} - path of friction; b_W - width of the gear tooth rim

We substitute formula (13) in formula (12), as we assume that the wear along the thickness of the gear tooth is uniform. This results in the following:

$$K_{uG} = b_W.\Delta_u.\gamma \cdot \frac{L_{TP,i} - L_{TP,(i-1)}}{\Delta N} \tag{14}$$

By taking into account that $\Delta_u = S_{i-1} - S_i$, we can record formula (14) in the following way

$$K_{uG} = b_W.\gamma.[L_{TP,i} - L_{TP,(i-1)}].K_{uS} \tag{15}$$

After solving the formula (15) for K_{uS}, the result is:

$$K_{uS} = \frac{K_{uG}}{b_W.\gamma.[L_{TP,i} - L_{TP,(i-1)}]} \tag{16}$$

Since the initial value of the mass of the toothed gear wheels can be accepted as the starting point of the reading, then $L_{TP,(i-1)} = 0$.

Therefore, the base wear rate of the toothed gear K_{uS} shall be calculated according to the equation:

$$K_{uS} = \frac{K_{uG}}{b_W.\gamma.L_{TP,i}} \tag{17}$$

Equation (17) is valid in this version only in the event of uniform wear of the teeth along the whole profile of the gear wheel. In the case of uneven wear of the toothed gear of P-859 system (that happens in the event of change in the supply voltage of the marine communication system), the authors propose that the friction path $L_{TP,i}$ measured in *mm* during the i-cycle of operation be represented by a polynomial function of the current time t, which looks as follows:

$$L_{TP,i}(t) = a_0 + a_1.t + a_2.t + ... + a_m.t^m \tag{18}$$

where: $a_0, a_1, a_2, ..., a_m$ are coefficients, defined by a system of equations for m - number of points from the graph of changes of $L_{TP,i}(t)$.

This system of equations shall appear as follows:

$$\left| \begin{array}{l} L_0 = a_0, \\ L_1 = a_0 + a_1.(t_2 - t_1) + a_2.(t_2 - t_1)^2 + ... + a_m.(t_2 - t_1)^m, \\ L_i = a_0 + a_1.(t_i - t_{i-1}) + a_2.(t_i - t_{i-1})^2 + ... + a_m.(t_i - t_{i-1})^m, \\ L_m = a_0 + a_1.(t_m - t_{m-1}) + a_2.(t_m - t_{m-1})^2 + ... + a_m.(t_m - t_{m-1})^m. \end{array} \right. \tag{19}$$

The final formula for determination of the base wear rate of the toothed gear mechanism during friction K_{uS} is given by:

$$K_{uS} = \frac{K_{uG}}{b_W.\gamma.(a_0 + a_1.t + a_2.t^2 + ... + a_m.t^m)} \tag{20}$$

4 Conclusion

As a result of the conducted study, the following conclusions can be drawn:

1. In this report, a practical formula has been derived for the technical resources of toothed gear mechanism of a marine communication system. These resources are determined by the limit value of wear of the teeth and the wear rate that depends on the wear parameters, load, structural and technological characteristics and operating conditions.
2. A mathematical relation has been defined, applicable to the wear rate of the gear mechanism, along the teeth section and

the mass of the gear wheel, in the case of uneven wear that is typical of marine systems.

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