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ABSTRACT

of the dissertation for the degree of Doctor of Sciences

A SYSTEMATIC APPROACH TO ASSESSING THE RELIABILITY OF TRANSMISSION MECHANISMS OF MACHINES AND EQUIPMENT

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GENERAL DESCRIPTION OF THE WORK

The urgency of the problem and degree of research. The reliability of transmission mechanisms, which are one of the components of modern machines and equipment with a mechatronic structure, is one of the main factors determining their quality level. Therefore, there is a great need to apply innovative methods for quantitative and qualitative assessment of the reliability of modern transmission mechanisms of complex design.

Transmission mechanisms currently used in machinery and equipment have a complex mechatronic structure consisting of various components, and their reliability depends on a large number of random factors that arise during design, production and operation, which are often impossible to predict. Each element of the transmission mechanisms is characterized by different performance criteria and corresponding failures. Therefore, a systematic approach is required to assess the reliability of the transmission mechanisms.

The issue of assessing the reliability of transmission mechanisms has always been relevant since the beginning of their mass production and has not lost its relevance today. In the 40-50s of the XX century, methods of probability theory and mathematical statistics began to be used to assess the reliability of machines and devices, as well as their main elements. Failures that occurred at this time were treated as a random event, and the time before failure was treated as a random variable. In the 60s of the XX century, the use of experimental methods for assessing reliability became widespread. These methods were based on the study of the causes of failures, the collection and analysis of data on failures, the creation of simple models of failures.

Modern methods of assessing the reliability of transmission mechanisms of machines and equipment can be divided into two main groups. In the methods included in the first group, an analysis of failures by their type and results is carried out and a qualitative assessment of reliability is carried out. That is, based on the experimental knowledge obtained, weak elements are identified and the reliability of the system elements is approximately determined. These methods are mainly used in the process of designing machines in order to predict reliability. Such methods include the methods of "Analysis of system elements“ and FMEA – "Failure Mode and Effects Analysis".

In the second group of reliability assessment methods, an analysis of the failure rate of the object is carried out and a quantitative assessment of reliability indicators is carried out. These methods are based on the application of methods of probability theory and mathematical statistics and may include Boolean theory, failure tree analysis, Markov model, Petri nets. These methods are reflected in the studies of M. Mayer, R. Barlow, N.F. Khotialov, N.S. Streletsky, V.V. Bolotin, S.H. Babayev and other scientists.

For the quantitative assessment of the reliability of different elements of gearboxes, the model “loading – loading capacity" is widely used. The estimation of the resource under periodic loads with constant and variable amplitude based on the characteristics of fatigue strength is reflected in the studies of A.Wohler, V.Weibull, E.Khabibullin, G.Nieman, H.Winter, V.P.Kogaev, S.V.Sorensen, N.A. Makhutov and other researchers. In the works of Professor AzTU A.H. Abdullaev was studied the question of assessing the reliability of tooth mechanisms under the normal law of load distribution and carrying capacity.

An analysis of the studies on the assessment of the reliability of the transmissions of machines and equipment suggests that the system analysis of the failures of modern mechatronic transmissions and the impact on the reliability of the failures that cannot be eliminated has not been sufficiently studied and a comparative analysis of the service life of the main working elements of the transmissions according to various failure criteria has not been carried out. In addition, the existing statistical models for predicting the reliability of modern mechatronic transmissions cannot be applied precisely to the entire operating cycle, and for this purpose the development of distribution functions covering the entire operating cycle is necessary.

Based on the above, it can be concluded that a systematic analysis of failures of modern transmission mechanisms with a mechatronic structure, the development of new statistical models that allow predicting their reliability, reliability assessment taking into account failures that cannot be eliminated, a comparative analysis of the service life of the main working elements of transmissions according to various failure criteria and on this basis, increasing their resource are an urgent task.

Purpose and tasks of the work. The purpose of the dissertation is a systematic study of innovative methods and tools that allow assessing and improving the reliability of modern mechatronic transmission mechanisms of machines and equipment.

To achieve this goal, the solution of the following tasks is provided:

1. Development of a mathematical model of a multi-criteria analysis of the reliability of modern transmission mechanisms with a mechatronic structure;
2. System analysis of failures of mechatronic transmission mechanisms of modern machines and equipment;
3. Development of new statistical models for forecasting the reliability of transmissions and their elements;
4. Comparative analysis of the service life of the main working elements of gears and worm gears according to various failure criteria;
5. Development of methods for increasing the service life of gearboxes.
6. Development of more reliable, economical and compact models of transmission mechanisms with innovative design;
7. System analysis of the reliability of transmission mechanisms, taking into account failures that cannot be recoverable;
8. Assessment of the technical and economic efficiency of the conducted research.

Methods of research. The questions posed in the dissertation work were solved on the basis of theoretical and experimental research. During the research, the methods of probability theory and mathematical statistics, the Markov model and the Laplace transform were used to solve a system of differential equations derived from this model. The system of differential equations, the solution of which is impossible by the analytical method, was solved using the MATLAB program. Fatigue curves and the “loading – loading capacity" model were used to assess the reliability of mechanical elements of gear mechanisms.

The main scientific theses for the defense:

- Development of a multi-criteria mathematical model for assessing the reliability of transmission mechanisms of machines and equipment with a mechatronic structure;
- A systematic analysis of the main causes of failure of modern transmission mechanisms with a mechatronic structure and a comparative analysis of the nature of the distribution of failures;
- Development of new mathematical models for the distribution of failures of transmission mechanisms and their main working elements of machines and equipment with a mechatronic structure, more accurately reflecting the distribution of modern, as well as failures of the man-machine system;
- Analysis of the causes of gear failure, comparative analysis of their service life, taking into account the influence of various factors on the criteria of bending and contact strength;
- Development of methods to increase the service life of gear by replacing the working profile of the tooth with a non-working profile;
- Development of more reliable, economical and compact models of transmission mechanisms with a new design;
- Systematic analysis of the reliability of modern transmission mechanisms, taking into account failures that cannot be recoverable;
- Determination of the demand for spare parts during a certain period of operation on the basis of the newly proposed laws of distribution of failures.

Scientific novelty of the research. New mathematical models are proposed to more accurately reflect the distribution of failures of modern transmission mechanisms with a mechatronic structure and their main working elements, as well as a human-machine system, and the reliability of these distributions is proved on the basis of statistical data on failures obtained in practice.

For the first time, a comparative analysis of the durability indicators of gear and worm gears was carried out according to various failure criteria, taking into account the influence of a number of factors, and as a result of this analysis, ways to increase the life of the gear mechanism were investigated. The possibility of increasing the durability of cylindrical and bevel gears by 50-60%, and the durability of worm gears by 30-50% by replacing the working profiles of the teeth of the wheels with non-working profiles after a certain period of operation is substantiated.

Using the example of the steering mechanism of track-type machine, a more reliable, economical and compact model of this mechanism with an innovative design has been developed, its technical condition has been evaluated and patented.

For the first time, a systematic analysis of the reliability of transmission mechanisms was carried out, taking into account the recoverable and non-recoverable failures, and the issue of determining the need for spare parts during a certain period of operation was considered on the basis of the proposed laws of failure distribution.

Theoretical and practical significance of the work. The proposed new statistical models have theoretical and practical significance for assessing the reliability of modern transmission mechanisms of machines and equipment with a mechatronic structure and their main working elements, as well as the man-machine system.

With correct choice of geometric parameters and wheel materials at the design stage can increase the durability of cylindrical and bevel gears by 50-60%, and worm gears by 30-50% by replacing the working profiles of the teeth with non-working ones after a certain service life.

The patented steering mechanism of track-type machine can be applied in practice, being more reliable, economical and compact.

A systematic analysis of the reliability of transmission mechanisms, subject to the consideration of recoverable and non-recoverable failures, can more accurately determine the need for spare parts during a certain period of operation on the basis of the proposed distribution laws.

Approbation and application of the work. Materials of the dissertation and its separate fragments were reported and discussed at the following international and republican scientific and technical conferences and approved:

- At 9th International Conference on Machine Design and Production. Ankara, Turkey: 13-15 Yuli, 2000;
- At seminars of the Institute "Machine Components” University of Stuttgart, Stuttgart, Germany, 2004, 2008;
- At conferences entitled "Technical reliability" ("Technische Zuverlässigkeit"), organized by the German Union of Engineers (Verein Deutscher Ingenieure), Stuttgart, Germany, 2005, 2007, 2019;
- At the symposium in Dresden, Germany (Dresdner Maschinenelemente Kolloquium DMK 2007), Dresden, Germany, 2007;
- At the Republican Scientific and Technical Conference dedicated to the 60th anniversary of AzTU, Baku, AzTU, May 2-3, 2010;
- At the seminar of the Institute of Engineering and Transmission Engineering of the University of Chemnitz (IKAT), Chemnitz, Germany, 2011;
- At seminars of the Institute of "Traffic Safety and Automation" of the Braunschweig University, Braunschweig, Germany, 2013, 2015;
- At the International Scientific and Technical Conference "Intelligent Technologies in Mechanical Engineering", Baku, AzTU, 2016;
- At the seminar of the Institute "Machine Components" of the Technical University of Munich, Munich, Germany, 2019;
- At the seminar of the Department of "Machine Design” of the Azerbaijan Technical University (currently "Mechatronics and Machine Design"), Baku, 2019;
- At the Republican scientific and practical conference "Modern problems of using technological machines in construction production", Baku, AzMIU, December 20-21, 2019;

- IX International correspondence scientific specialized conference "International scientific review of the technical sciences, mathematics and computer science", Boston, USA. February 12-13, 2019.

The main provisions of the dissertation are reflected in 33 scientific articles and 1 copyright certificates (patents) of Eurasia.

The results of the dissertation can be applied in the educational process, scientific research work and in practice. In the educational process, the main results of the dissertation are included in the program of subjects taught at the Department of "Mechatronics and machine design" of AzTU. It was recommended to use the practical results of the dissertation work in increasing the service life of the transmission mechanisms produced and operated in enterprises and workshops related to "Technics Reducer Service" MMC, "Baku Bearings factory" ASC, "Azerpambig ASK" MMC, "Azerbaijan Railways" QSC.

The name of the organization in which the dissertation work is performed. The dissertation was carried out at the Department of "Mechatronics and Machine Design" of the Azerbaijan Technical University. Durability tests on an worm gears were carried out in the laboratory of the Institute "Design and Transmission Technology" (IKAT – Institut für Konstructions- und Antriebstechnik) of the University of Chemnitz, Germany.

The total volume of the dissertation, indicating the volume of structural sections. The dissertation work consists of an introduction, 7 chapters, general conclusions, a list of references and appendices. The volume of the "introductory" part of the dissertation is 17669, the volume of chapter I is 50842, the volume of chapter II is 44396, the volume of chapter III is 56654, the volume of chapter IV is 48116, the volume of chapter V is 68417, the volume of chapter VI is 68818, the volume of chapter VII is 23581 characters, and the total volume is 382283 characters.

The dissertation consists of written material compiled on 309 A4 pages, including 64 figures, 87 graphs, 23 tables, 190 titles of literary sources and appendices.

In the introduction presents role of transmission mechanisms in modern machines and equipment, the urgency of the problem and degree of research, purpose and tasks of the work, methods of research, the main scientific theses for the defense, scientific novelty of the research, theoretical and practical significance of the work, approbation and application of the work, volume and structure.

The first chapter of the dissertation presents the history of the development of transmission mechanisms used in machinery and equipment, and their classification according to the principle of operation, structure and functional characteristics, provides information about the basic requirements for the design of modern transmission mechanisms with a mechatronic structure, reflects a brief overview of existing research and modern methods for assessing their reliability.

The analysis of the reviewed literature on assessing the reliability of transmission mechanisms allows us to conclude that, despite a large number of studies in this area, the system analysis of failures of modern transmissions with a mechatronic structure and the effect of recoverable and non-recoverable failures on reliability has not been sufficiently studied, a comparative analysis of the durability of the main working elements of transmission mechanisms according to various failure criteria has not been carried out. In addition, the existing statistical models used to predict the reliability of transmission mechanisms cannot be accurately applied to the entire cycle of operation, and for this it is necessary to develop distribution functions covering the full cycle of operation.

Transmission mechanisms are one of the main components of modern machines and equipment with a mechatronic structure, and the correct assessment of their reliability, increasing their durability and maintainability are of great importance.

Taking into account all the above, the issues of system analysis of failures of modern transmission mechanisms, the development of new statistical models that allow predicting reliability, reliability assessment taking into account failures that cannot be eliminated, comparative analysis of the durability of the main working elements of transmissions according to various failure criteria become relevant.

The second chapter is devoted to the study of failures of mechanical, hydraulic, electrical, etc. transmission elements and their causes, the determination of the main reliability indexes based on theoretical and statistical methods, the construction of a structural diagram of the system analysis of the reliability of transmission mechanisms and the definition of the main areas of research.

Analysis of failures of modern transmission mechanisms with a mechatronic structure shows that the main failures of mechanical elements wear, fatigue can be predicted and calculated, but failures of the nature of hydraulic, electrical and electromagnetic elements are stochastic and cannot be predicted. Therefore, a systematic approach is required to assess or predict the reliability of the transmission mechanism as a whole. In order to obtain more accurate results in assessing the reliability of transmissions, it becomes necessary to analyze failures during operation. Fig. 1 shows the main types of failures that occur in transmission systems and the corresponding methods of reliability analysis.

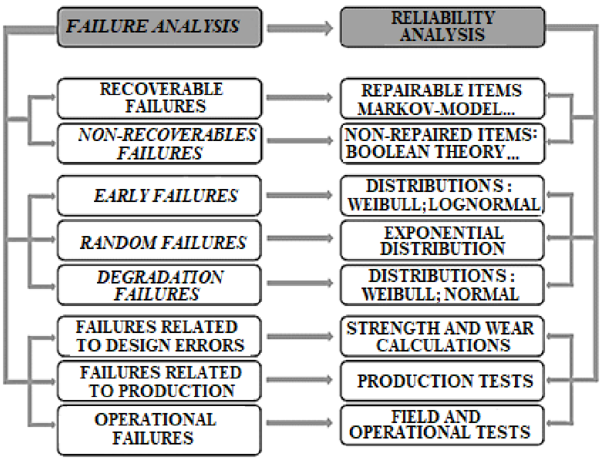

Fig. 1. Failure analysis and reliability assessment methods

Failures of transmission mechanisms of machines and equipment occur mainly during operation. In many cases, these failures are due to the fact that the main working parts lose their operability from the effects of wear, fatigue, vibration and corrosion after prolonged operation. Accidental overloads, human-machine system errors and control system defects can also sometimes cause failures.

Based on the above, the block diagram of the system analysis of the reliability of transmission mechanisms of modern machines and equipment can be described as shown in Fig. 2. It is very important to choose the right method for assessing the reliability of the transmission mechanism and the statistical model for the recoverable and non-recoverable failures. Here it is necessary to take into account the nature of failures and in what period of operation they most often occur.

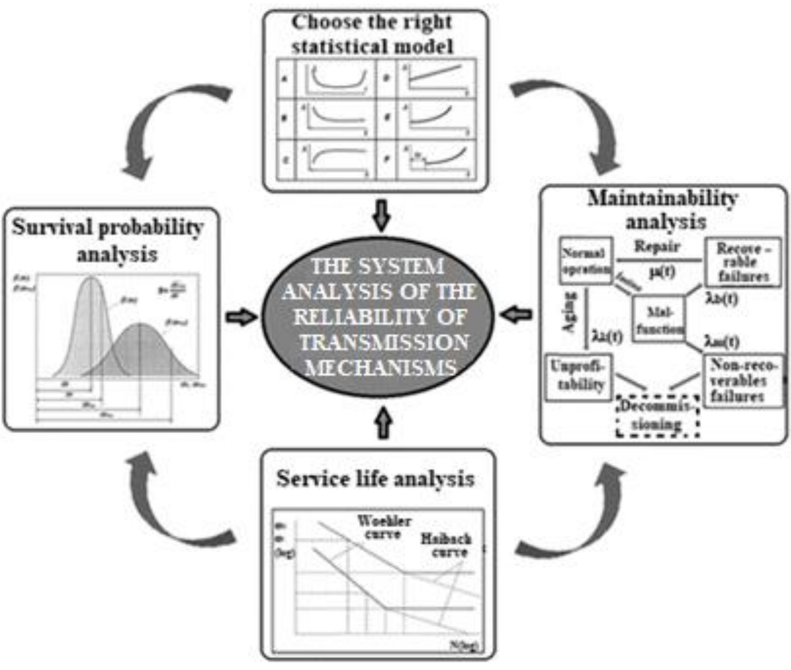

Fig. 2. Block diagram of the system analysis of the reliability of transmission mechanisms

Another area of research is the system analysis of recoverable and non-recoverable failures. In the existing literature, these failures are considered separately for different objects, and cases of accounting for both failures for the same object have not been considered. The transmission mechanisms are maintainable, that is, after failure, these failures are eliminated in most cases, and operability is restored. But in some cases, there may also be such failures that the restoration of operability becomes impractical from a technical point of view or due to a decrease in profitability. This may be due to the fact that the main working elements have become completely unusable as a result of an accident or wear. Therefore, in order to increase the availability factor of the machine fleet and provide the necessary spare parts, it is important to take into account failures that cannot be eliminated.

It may also be of particular importance to conduct a comparative analysis of durability according to various criteria for the operability of the elements of the transfer mechanism. Such an analysis can play a role, in particular, in increasing the durability of gears. As a result of contact stresses, the working surfaces of the teeth, as a rule, fail. And bending stresses can lead to tooth breakage on both surfaces. In gears with higher bending strength, it is possible to increase the life of the gear mechanism by replacing the working surfaces with non-working surfaces after a certain period of operation. Therefore, a comparative analysis of the durability of gears according to the criteria of contact and bending strength can be of great practical importance.

The third chapter of the dissertation is devoted to the analysis of existing statistical models that are most often used in assessing the reliability of transmission mechanisms, the development of new distribution functions that allow more accurately predicting reliability indexes throughout the entire service life, and testing their application in practice.

The existing failure distribution functions, although they have a number of advantages, in most cases do not allow taking into account the failures of modern transmission mechanisms during the entire period of their operation. To this end, it becomes more expedient to use a superposition of the current laws of the distribution of failures. For the distribution density of a statistical model consisting of a superposition of exponential and normal distributions, the following expression can be used:

$$f(t) = \frac{k}{T} e^{-\frac{t}{T}} + \frac{1-k}{\sqrt{2\pi}\cdot\sigma} e^{-\frac{(t-T)^2}{2\sigma^2}}, \qquad (1)$$

here, the first term of the formula is equal to the exponential distribution with the failure rate λ_0=1/T, and the second term is equal to the normal distribution with the mathematical expectation T, the mean square deviation σ. The coefficient k (0≤k≤1) reflects the role of these two classical distributions in the statistical model. In Formula (1), when k=0, it turns into a pure normal distribution, and when k=1, it turns into a pure exponential distribution (Fig. 3). from the graph of the dependence of the failure rate on time at different values of k, it can be seen that the new model combines the advantages of both classical distributions, and eliminates the disadvantages. Unlike the exponential distribution, the new model takes into account an increase in the failure rate during the wear period, while unlike the normal distribution, it takes into account failures that occur during the early failures period and normal operation period.

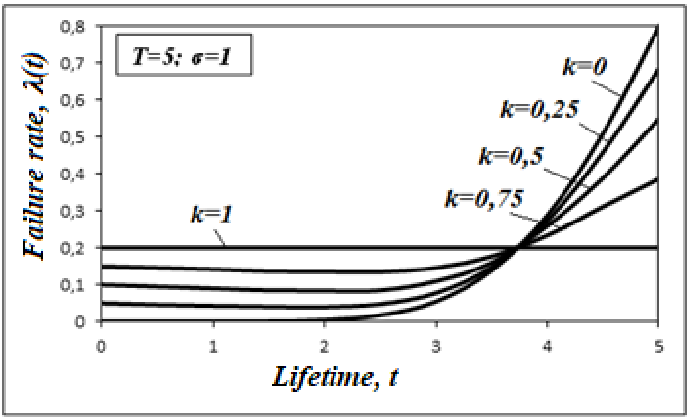

Fig. 3. Dependence of the failure rate on time at different values of the coefficient k

A number of failures of the transmission mechanism and its individual elements are sudden. These include the failure of parts and assemblies whose failures cannot be predicted in advance, for example, sealings, electromagnets, electronic control devices. For the failure rate in these cases, the following three-parameter distribution was proposed:

$$\lambda(t) = \lambda\left[1 + (\alpha - 1)e^{-\beta t}\right]. \qquad (2)$$

Where λ is the value of the failure rate in the normal period of operation; α is a shape parameter of distribution and takes into account the value of the failure rate (λ_0) at the start of operation, i.e. $\alpha=\lambda_0/\lambda$; β is a coefficient that takes into account the length of the first period of operation (the period of early failures) (t_f in Fig.4).

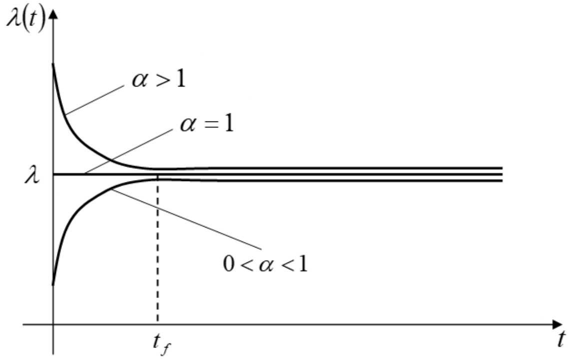

Fig. 4. Failure rate for three-parameter distribution

As can be seen from the graphs in Figure 4, the nature of the change in the failure rate over time, depending on the value α, is different. When α>1, the failure rate gets a high rating at the beginning of operation, and then remains constant throughout the entire service life. Such a change in the failure rate may be due to design errors and manufacturing defects. This circumstance can also be applied when assessing the reliability of the man-machine system. Because a novice machinist makes more mistakes until he reaches the required level of professionalism. And as soon as professionalism reaches the required level, the failure rate becomes constant. At α<1, the failure rate increases from a given minimum to any value, and then remains constant throughout the entire service life.

In order to obtain a more universal distribution, which also allows us to express an extreme distribution, the following expression for the failure rate was proposed:

$$\lambda(t) = \lambda\left(1 + \alpha \cdot e^{\beta t}\right). \qquad (3)$$

Where λ is the value of the failure rate at the beginning of operation. at small values of λ, the failure rate in the initial period of operation is also small; α is the coefficient of the distribution form; β-shows the time-dependent change in the failure rate.

The dependences of the failure rate on time can be obtained in accordance with the two cases shown in Figure 4, when α>0, β<0 and α<0, β<0. When α>0 and β>0 in expression (3), the failure rate increases monotonically, starting from any minimum value (Fig.5). These can still be considered as partial solutions for the distribution of extreme values. This distribution of failures is typical for most machine parts, such as transmission mechanisms, which gradually fail as a result of wear, fatigue, corrosion or aging.

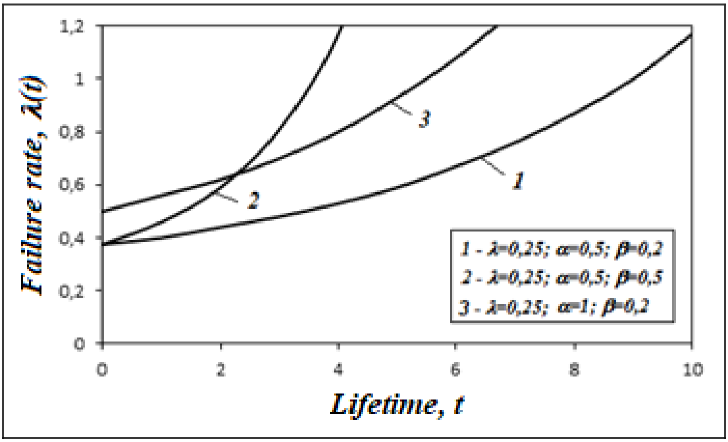

Fig. 5. Increased failure rate at α>0 and β>0

Statistical data on deviations as a result of tests conducted on diverse equipments were used to apply and evaluate the proposed distribution. To assess the degree of compliance of the distribution with the test results, the χ^2–Pearson criterion was applied and the possibility of using this distribution in assessing reliability was confirmed.

The fourth chapter is devoted to the analysis of the system reliability of the transmission mechanism at the design stage. Modern gear mechanisms have a rather complex mechatronic structure and consist of a large number of elements with different operating principles. Each element of the transmission mechanism is characterized by different operability criteria and failures. Therefore, the structural scheme of the system analysis of the reliability of the transmission mechanism based on the ”load - load capacity" model can be described in accordance with figure 6.

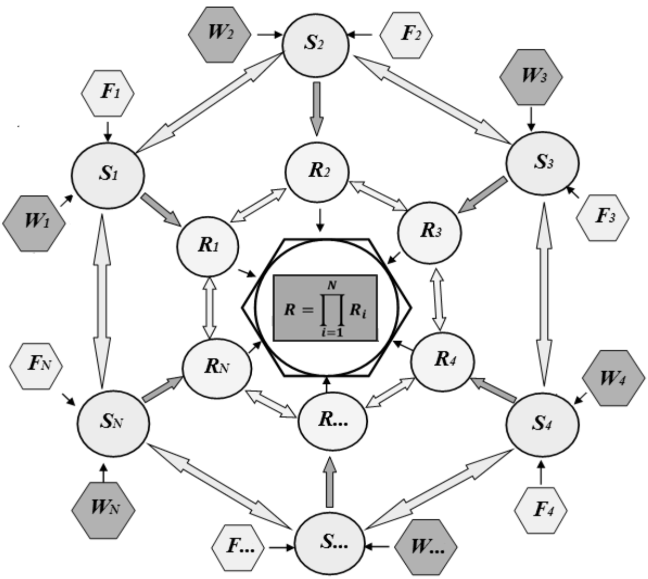

Fig. 6. Structural scheme of the system analysis of the reliability of the transmission mechanism

Based on this assumption, the mathematical model of the system analysis of the reliability of the transmission mechanism can be analytically expressed in the form of the following system of equations:

$$\begin{cases} \varphi_1(l_{11}, l_{12}, ..., x_{11}, x_{12}, ..., S_1, R_1) = 0 \\ \varphi_2(l_{21}, l_{22}, ..., x_{21}, x_{22}, ..., S_2, R_2) = 0 \\ \dots\dots\dots\dots\dots\dots\dots\dots\dots\dots \\ \varphi_i(l_{i1}, l_{i2}, ..., x_{i1}, x_{i2}, ..., S_i, R_i) = 0 \\ \dots\dots\dots\dots\dots\dots\dots\dots\dots\dots \\ \varphi_N(l_{N1}, l_{N2}, ..., x_{N1}, x_{N2}, ..., S_N, R_N) = 0 \end{cases} \qquad (4)$$

Where l_{ij} are the input parameters of the i-th element (geometric, physical, etc.); x_{ij} are external factors affecting the i-th element (load, force, strength etc.); S_i is the reserve coefficient of the i–th element according to the criterion of the considered operability; R_i is the survival probability of the i–th element according to the criterion of the considered operability; N is the number of elements of the transmission system.

Most of the mechanical elements of the gear mechanisms of machines (gears, bearings, shafts, etc.) they are more susceptible to failure as a result of loss of fatigue strength after prolonged exposure to alternating strength. Gears can fail both under the action of bending and contact stresses. The effect of these stresses can have different characteristics of change depending on time. Even when the transmission mechanism is under the action of a static external load, during one full cycle of the gear wheel, each of its teeth engages at least once, and at this time the contact and bending stresses change periodically. That is, even when the transmitted torque is constant, the gears are exposed to periodically changing bending and contact stresses.

Fatigue curves are used to assess the reliability of parts affected by variable stresses. Fig. 7 shows fatigue curves for bending and contact stresses of gears.

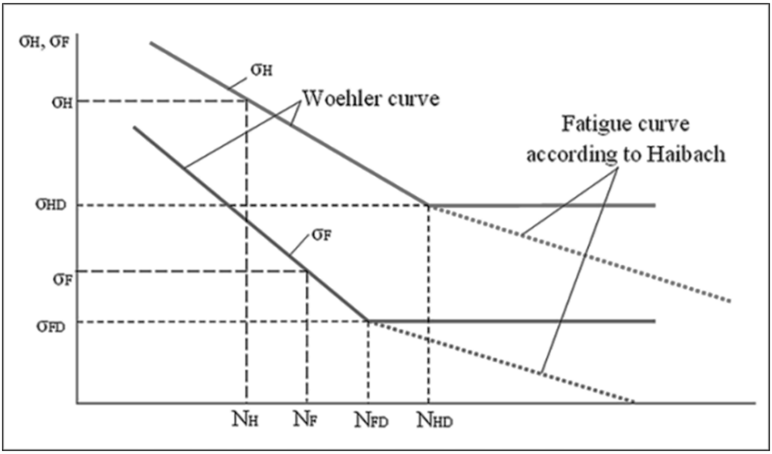

Fig. 7. Fatigue curves

When determining the resource of gears, the Miner-Haibach hypothesis is used at loads less than the endurance limit and the Wohler hypothesis at high load values. Assuming that the values of the actual and limit stresses are subject to a normal distribution, a graphical interpretation of determining the probability of failure at constant load based on fatigue curves can be described as in Fig. 8. For each variable load mode, it is necessary to determine the resource and the probability of failures (Fig. 9). The lifetime of the gear according to any criterion of operability at such a load can be calculated using the following expression[1]:

$$N_\Sigma = N_D \frac{\sum_{i=1}^{j+n} h_i}{\sum_{i=1}^{j} h_i \left(\frac{\sigma_i}{\sigma_D}\right)^k + \sum_{i=j+1}^{j+n} h_i \left(\frac{\sigma_i}{\sigma_D}\right)^{2k-1}}. \qquad (5)$$

Where σ_i - values actual stresses (σ_H or σ_F); j - the number of load conditions provided that ($\sigma_F \geq \sigma_{FD}$; $\sigma_H \geq \sigma_{HD}$); n - the number of load

[1] Naunheimer H. Fahrzeuggetriebe. Lehrbuch/H. Naunheimer, B. Bertsche, G. Lechner. – Berlin Heidelberg: Springer-Verlag, -2007. 710 S.

conditions provided that ($\sigma_F<\sigma_{FD}$ and $\sigma_H<\sigma_{HD}$); h_i-the difference between load cycles, according to figure 9, corresponding to each load conditions $h_i=N_i-N_{i-1}$.

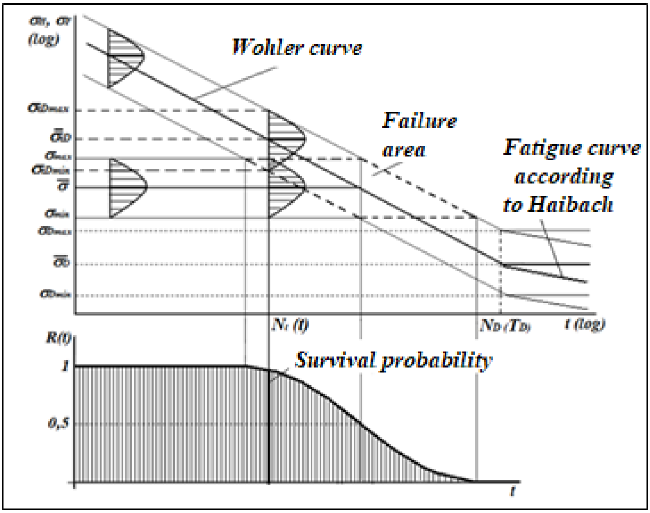

Fig. 8. Graphical interpretation of determining the probability of failure at constant load

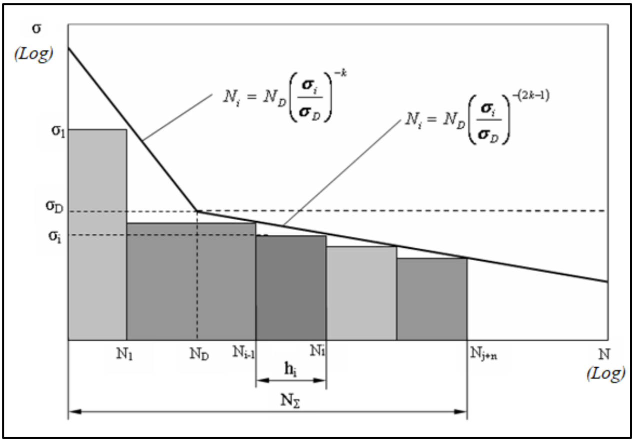

Fig. 9. Definition of resource in variable load

Based on the presented methodology, the issue of predicting the reliability of the cylindrical gear transmission of the winch (Fig. 10) with a load capacity of F=10 kn, lifting speed V=0.8 m/sec, drum diameter d_b=0,6 m. The gear material was selected steel grade 45X with a surface hardness HRC 45-50, core hardness HB 269-302 and the hardness of the driven gear was selected HB269-302. Taking the coefficient of variation of the load equal to 5%, the survival probability of the gears of a cylindrical spur gear depending on its service life is determined.

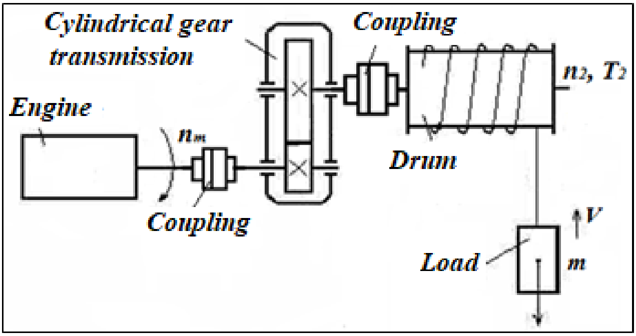

Fig. 10. Transmission mechanism of the winch

Calculations have shown that the probability of failure of the drive gear as a result of bending stresses is greater. This is due to the fact that the surface hardness of the material of this wheel is great. And on the driven wheel there was a high probability of failure as a result of contact stresses. Calculations have proved that the weakest element of the transmission is the driven gear wheel. Therefore, the overall degree of reliability is determined precisely by the possibility of survival probability due to the contact strength of the driven wheel. To increase the contact strength of the gear, it is necessary to choose a material with a higher hardness or change the type of heat treatment. The dependence of survival probability of a cylindrical gear reducer on time is shown in Figure 11.

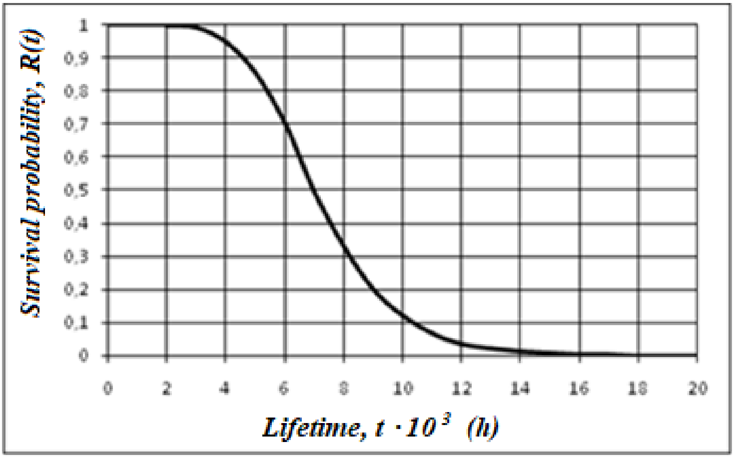

Fig. 11. The dependence of survival probability of a cylindrical gear reducer on time

One of the main methods of increasing the reliability of machines and their drive systems in the design is to improve the design. In recent years, significant research and innovative projects have been carried out in this direction at the Department of "Machine Design" (currently "Mechatronics and Machine Design") of Azerbaijan Technical University. The proposed transmission mechanisms of the new design have great advantages in their compactness, high performance and reliability. The compactness of the design is achieved by reducing the number of shafts in the proposed packet transmission mechanisms. In the presented dissertation work, the designs and qualitative indicators of the steering mechanism of track-type machine were analyzed, a kinematic scheme of the transmission mechanism of a new design was developed, which has higher reliability and low metal consumption and received a Eurasian patent. The purpose of the invention is to increase the gear ratio, reliability, efficiency and reduce weight and overall dimensions. To achieve this goal, a four-stage mechanism for steering mechanism of track-type machine was proposed (Fig.12).

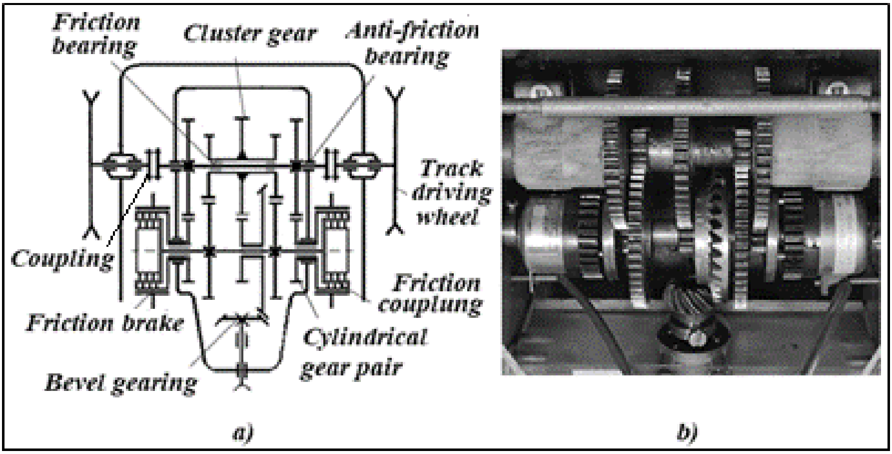

Fig. 12. Four-stage mechanism for steering mechanism of track-type machine

The proposed design of a four-stage two-flow gearbox for steering mechanism of track-type machine has better drive characteristics compared to currently existing mechanisms and allows you to increase the gear ratio. However, since most wheels of the gearbox have the same geometric dimensions, the manufacturability of production and unification are ensured, special requirements for the accuracy of manufacturing and installation of the structure are not imposed, the degree of reliability of the mechanism increases due to the reduction of two intermediate shafts and two pairs of rolling bearings.

In order to verify the operability of the proposed innovative design, a laboratory sample of a four-stage two-flow mechanism for turning tracked vehicles was designed, developed and tested. Based on the results of the preliminary assessment, the following were determined:

- the new design of the steering mechanism of the four-stage two-stream transmission works normally and fulfills its functional task;
- due to the increase in efficiency factor, fuel consumption is reduced;
- the overall mass of the mechanism is reduced;

- it is possible to increase the gear ratio by increasing the number of the wheel of the gearbox rotation mechanism;

In the fifth chapter, the issue of comparative analysis of the lifetime of gears according to various failure criteria is considered. Failures occurring in gears are generally associated with contact and bending stresses occurring in the teeth (Fig. 13).

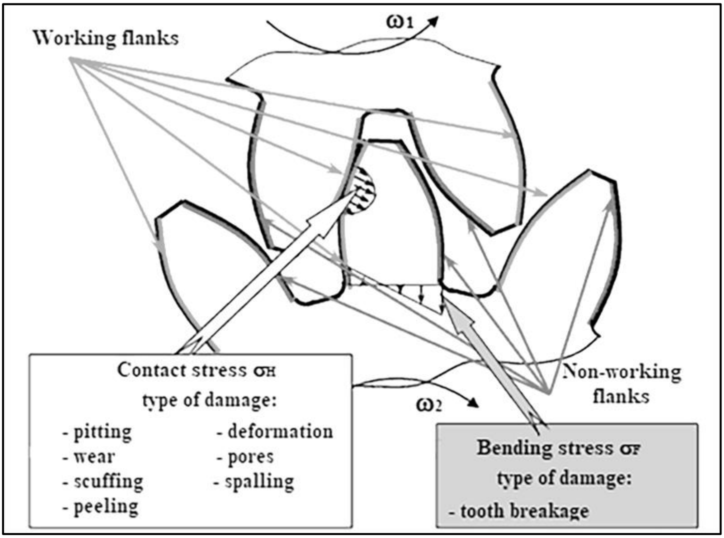

Fig. 13. Effective stresses and types of gear teeth damage

The most common cases of gear failure in the gears of modern machines and equipment are tooth breakage as a result of loss of fatigue strength after prolonged exposure to bending stresses and the susceptibility of the working surfaces of the teeth to pitting as a result of fatigue from exposure to contact stresses[2]. In the existing literature, the calculation of the durability of gears according to various failure criteria is compiled separately, and a comparative analysis is not

[2] Иванов М.Н. Детали машин. Учебник/М.Н. Иванов, В.А. Финогенов. – Москва: «Высшая школа», - 2008. – 408 с.

carried out. Therefore, these studies do not draw any conclusions about what the probability of failure is higher in the designed gear and which failure may occur earlier. Tooth breakage is more dangerous, and it leads to failure not only of the wheel in question, but also of the entire transmission. And failures associated with contact stresses arising on the working surface of the teeth do not always lead to a complete failure of the gear.

If during operation the gears of the gear mechanism rotate only in one direction (in most machines and equipment this circumstance prevails), then there may be cases of failure caused by the impact of contact stress is associated with damage occurring on the working surfaces of the teeth. And the non-working opposite profiles of the teeth are not exposed to contact stresses and no damage occurs on these surfaces (Fig.13). On the contrary, bending stresses affect both the working profile and the non-working profile. Therefore, it is possible to increase the durability of the gear mechanism by replacing the working profiles, relatively damaged as a result of tooth fatigue, with non-working ones after a certain period of operation. But it should be borne in mind that if the bending strength of the teeth is not high enough, then this operation does not matter. This is due to the fact that bending stresses can lead to breakage of both working and non-working tooth profiles. Thus, it is possible to increase the durability of gears by the proposed method only when the bending strength of the teeth is sufficiently high. Therefore, a comparative analysis of durability indexes for contact and bending stresses for gears is of particular importance.

Studies have shown that bending stresses are more dangerous in reverse gear loading compared to contact stresses. Thus, high bending strength is of great practical importance for both one-sided and two-sided loading of gear teeth. Therefore, it is of particular importance to conduct a comparative analysis of the durability of gears according to two different failure criteria - the strength condition for bending and contact stresses. Such an analysis can play an important role in the correct choice of wheel material, module and width coefficients in the design of new gearboxes. To conduct a comparative analysis of the resource of gears according to two different failure criteria, a dimensionless value K_{FH}, called the service life ratio, was used:

$$K_{FH} = \frac{N_F}{N_H}. \qquad (6)$$

where N_F – estimated limit number of loading cycles of alternating bending stresses σ_F until complete fracture (tooth breakage) occurs; N_H - estimated limit number of loading cycles of alternating contact stresses σ_H until complete fracture (pitting) occurs.

The number of loading cycles N_F and N_H at stresses less than the endurance limit is determined using fatigue curves based on the Miner-Haibach hypothesis, and for large stress values based on the Wohler hypothesis (Fig. 7). And for variable load, they should be determined based on the formula (5). As already mentioned, variable bending stresses are considered more dangerous for gears. The condition $K_{FH}>2$ is necessary in order for the bending stress resource to be greater in comparison with the contact stresses when the teeth are loaded bilaterally. In the case of unilateral tooth loading, durability can be increased by replacing the working surface with a non-working surface after a certain period of operation, if the condition $K_{FH}>2$ is met. Therefore, the study of how the K_{FH} coefficient varies depending on various factors, such as gear speed, material characteristics, modulus and width coefficient, is important.

Based on the presented methodology, calculations were made for various types of transmissions in order to study the influence of various parameters on the service life ratio. For the drive wheel of a rectilinear cylindrical wheel drive with an inter-center distance of a_w=160 mm, the gear ratio u = 4, the dependence of the service life ratio on the speed of rotation of the gears, the module and the coefficient of the width of the gear wheel in diameter is shown in Figures 14-17. As can be seen from figures 14 and 15, in gears made of both improved steels and hardened steels, the lifetime ratio K_{FH} decreases with increasing peripheral speed *V*.

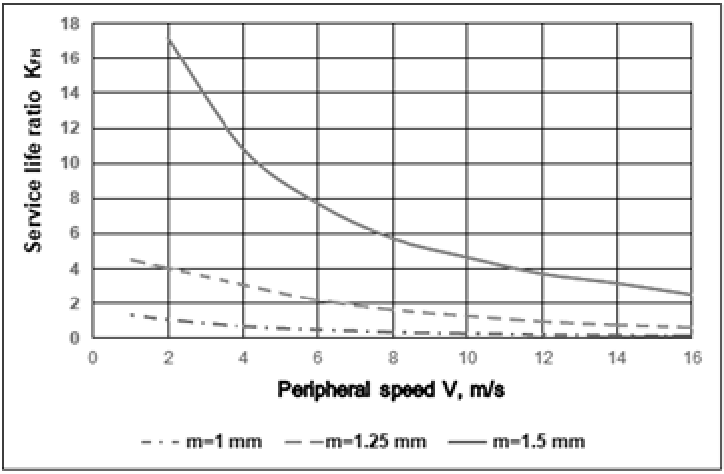

Fig. 14. Service life ratio for the pinion of the straight-tooth spur gearing according to peripheral speed for different modules (for gears made of improved steel 34CrMo4, center distance 160 mm, tangential force 3125 N)

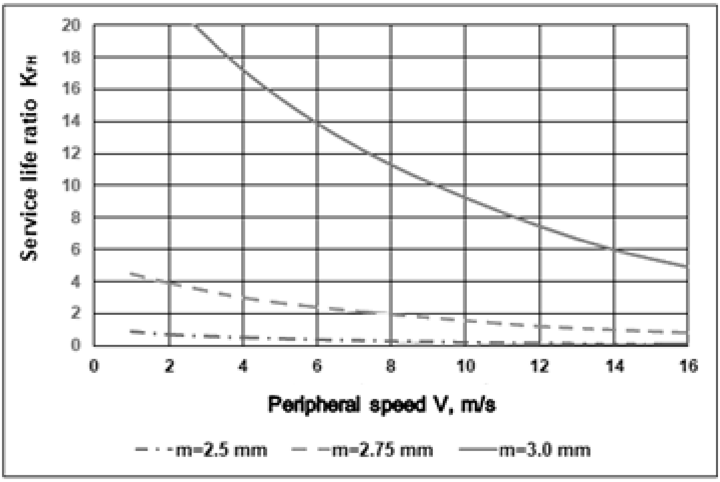

Fig. 15. Service life ratio for the pinion of the straight-tooth spur gearing according to peripheral speed for different modules (for gears made of improved steel 16MnCr5, center distance 160 mm, tangential force 7800 N)

Module m and tooth width factor *b/d* also have a significant influence on the lifetime ratio. For gears made of improved steel (34CrMo4, with σ_{FD}=520 N/mm^2 and σ_{HD}=530 N/mm^2)[3] is K_{FH} >2 for modules ab 1.25 mm and a tooth width factor of 0.3 (Fig. 16). For gears made of hardened steel (in this case, 16MnCr5, with σ_{FD}=860 N/mm^2 and σ_{HD}=1470 N/mm^2) the situation changes significantly. Due to a heat treatment, the pitting resistance of the teeth is increased more than the tooth strength for bending stresses, which is why the service life ratio decreases significantly (Fig. 17). Tooth flank changing would only be useful for modules from 3 mm and tooth width factor from approximately 0.35 (or for modules *m*>2.75 mm and tooth width factor *b/d*> 0.4).

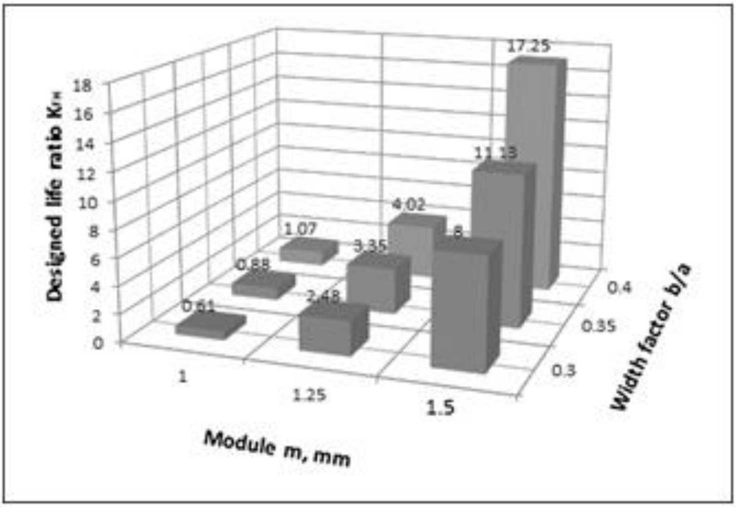

Fig. 16. Service life ratio for the pinion of the straight-tooth spur gearing for different modules and width factors (for gears made of improved steel 34CrMo4, center distance 160 mm, tangential force 3125 N)

[3] Decker K.-H. Maschinenelemente: Funktion, Gestaltung und Berechnung. Lehrbuch/K.-H. Decker. - München Wien: Carl Hanser Verlag, - 2018. 677 S.

In large-sized gear reducers, an increase in durability from an economic point of view is of particular importance. Therefore, a comparative analysis of the lifetime indexes of the teeth of the wheels according to various failure criteria for a new type multi-stage gearbox for sucker-rod pumping units was also considered.

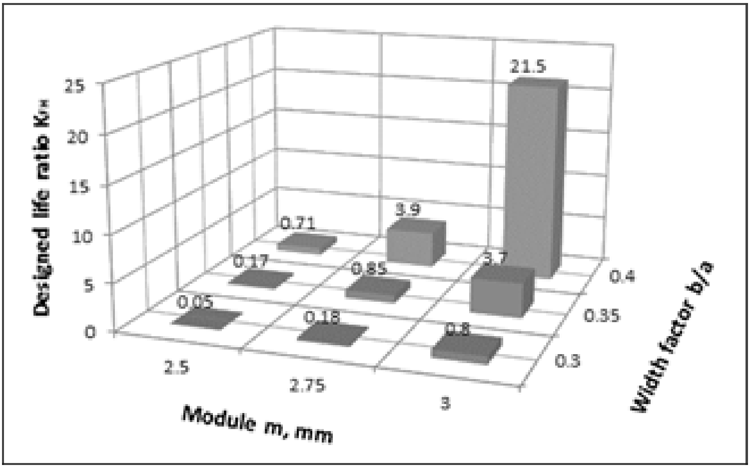

Fig. 17. Service life ratio for the pinion of the straight-tooth spur gearing for different modules and width factors (for gears made of hardened steel 16MnCr5, center distance 160 mm, tangential force 7800 N)

The gearbox was developed at the Department of Machine Design (currently Mechatronics and Machine Design) of AzTU. The gearbox (Fig.18) consists from only two shafts and gears mounted on them. The drive gear is mounted rigidly to the drive shaft by means of keyed joint. The heavily loaded driven gears are also symmetrically fixed to the driven shaft by means of keyed joint. Intermediate double-crowned gear and three-crowned gear blocks mounted on sliding bearings rotate freely on shafts. The main advantages of the proposed design are compactness, reliability, manufacturability and high gear ratio.

Depending on the torque on the output shaft and the required gear ratio, various modifications of the presented gearbox of a new type were designed. A comparative analysis of the lifetime indexes of the gears of a two-line three-stage gearbox with a center distance of a_w=315 mm and a total gear ratio of u = 91,1 is considered. The gear ratio of each stage is 4,5. Because of the low-speed gear stage takes up the greatest load, the calculations were performed for the gears of this stage. Other gear parameters (for example, number of teeth, modules, width factors and mechanical characteristics of materials) are varied for comparative analysis.

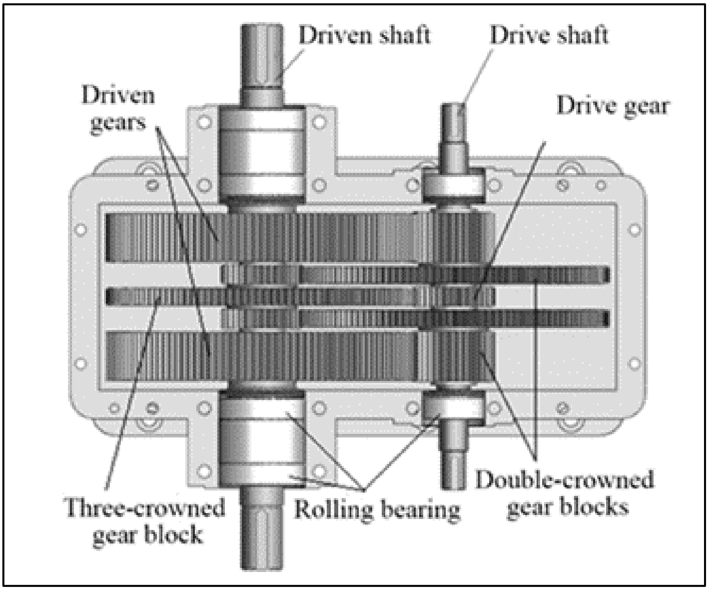

Figure 18. Two-line three-stage spur gear reducer

The result of calculations showed that for gears made of improved steel 34CrMo4 (σ_{FD}=520 N/mm^2 and σ_{HD}=530 N/mm^2), with a modulus m>2 mm and the facewidth factor ψ_{ba}>0,225 (or m>2,5 mm and ψ_{ba}>0,2) the K_{FH} factor is greater than 2. Calculations carried out for gears made of hardened steel 16MnCr5 (σ_{FD}=860 N/mm^2 and σ_{HD}=1470 N/mm^2) showed that starting for the values of module m>5,5 mm and the facewidth factor ψ_{ba}>0,25 (or with values of m>6 mm and ψ_{ba}>0,225) the factor K_{FH} is greater than 2. This is due the fact that during hardening, the contact strength of the teeth increases significantly more than the bending strength.

Currently, two- and three-stage cylindrical gear reducers with Novikov transmission are used on sucker-rod pumping units in the oil industry of Azerbaijan and a number of CIS countries. Similar studies were carried out on one of Novikov's three-stage gear reducers of the ЦЗНШ-450-28 brand used on sucker-rod pumping units. The total gear ratio of the gearbox under study is u = 64.57, and the rated torque is 28 kN·m.

Based on the proposed methodology, the values of the service life ratio were determined under various load conditions. The service life ratio of the gears of all stages of the gearbox under the influence of the rated load received a very high rating. This situation is repeated in overload modes that are many times higher than the rated load. For the gears of the intermediate stage of the gearbox and the low-speed stage, the values of the service life ratio under heavy loads are noticeably reduced, but even with a load exceeding the nominal three times, the condition K_{FH}>2 is fulfilled. From this it can be concluded that the probability of failure under the influence of bending stresses of these wheels is very small if design and production errors are not made and the operating rules are fully observed. The failure of teeth in most cases is associated with damage caused by contact stresses acting on the working profile of the tooth.

A comparative analysis of service life according to various failure criteria was also carried out for worm-gear. Worm-gears are very widely used in many areas of mechanical engineering, because they have a large gear ratio, high kinematic accuracy, smooth and silent operation compared to other mechanical transmissions. One of the disadvantages of these gears is the high cost of their material and manufacture. Therefore, the implementation of preventive measures to improve the reliability of worm-gears is important.

Studies show that the cases of failure of the elements of a worm-gear are closely related. For example, worm-gear wheel made of bronze, the manifestations of damage resulting from scuffing of the working surface with increasing temperature subsequently disappear with increasing wear. The same can be said about pitting. Putting as a rule, reaches its maximum after prolonged operation, after which the process of intensive wear of the surface begins. As a result of absorption, the thickness of the teeth decreases, and the strength of the teeth decreases due to bending stresses. As a result of the action of bending stresses, the palate in most cases occurs on the teeth most susceptible to eating. It is established that the service life of worm-gear to failure in most cases can be divided into three stages[4]. Fig. 19 graphically shows the stages of failure on worm-gear wheel, which is first putting and then wearing.

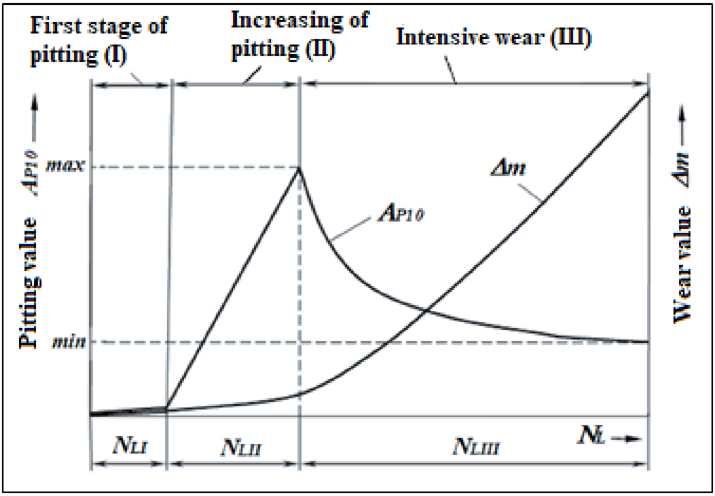

Fig. 19. The stages of failure on worm-gear wheel

The first stage of operation ends with the appearance of pitting on worm gearwheel, the length of this stage depends on the actual contact

[4] Rank, B. Versuche zur Grübchentragfähigkeit von Schneckengetrieben: Untersuchungen an Zylinder-Schneckengetrieben - Grübchenbildung, Verschleiß/B. Rank. – München: TUM. FVA-Forschungsvorhaben 12/IV. Heft 494, -1996. 286 S.

stresses arising on the working surface of the tooth and the sliding speed. At loads less than the rated load, the first stage of pitting can last quite a long time. At the end of this stage, the area damaged as a result of pitting is about 2% of the working surface of the tooth. After the appearance of the first occurrence of pitting, the process of its development accelerates and the second stage of operation begins. This stage is characterized by the rapid spread of pitting on the working surfaces of the tooth. Usually, the pitting does not cover the entire work surface. Studies show that after the pitting area covers 50-60% of the working surface, an increase in the wear intensity on the surface leads to its gradual decrease. After that, the third stage of operation begins. This stage is characterized by a sharp increase in the intensity of wear of the working surface and a reduction in the pitting to minimum. The third stage, as a rule, leads to tooth breakage under the influence of bending stresses, based on its weakening as a result of wear. The stages of occurrence of failures in the teeth of a worm gearwheel are reflected in Figure 20, a. As can be seen from the figure, damage to the tooth surface occurs, as a rule, in its working profiles. And non-working opposite profiles is practically not susceptible to damage after prolonged operation. Therefore, at the end of the second stage of operation, it is possible to increase the durability of the worm gear drive by replacing the working profiles with non-working ones. As can be seen from Figure 20, b, with this method, the resource increase will be approximately in the amount of N_{LI} + N_{LII}. Thus, to estimate the relative increase in service life time, the following formula can be used:

$$\Delta L(\%) = \frac{N_{LI}+N_{LII}}{N_{LI}+N_{LII}+N_{LIII}} \cdot 100\%. \qquad (7)$$

Using the presented methodology, calculations were performed on various values of the gear ratio (u) and the number of cycles (n_1) of the worm gear with an inter-center distance a_w=160 mm, the torque on the shaft of the worm wheel T_2= 4900 Nm. The material of the worm was adopted hardened steel grade 16MnCr5 (surface hardness 58÷62 HRC), and the material of the worm wheel CuSn12Ni2-C-GZ (σ_{HD}=520 N/mm^2), obtained as a result of casting by centrifugal method.

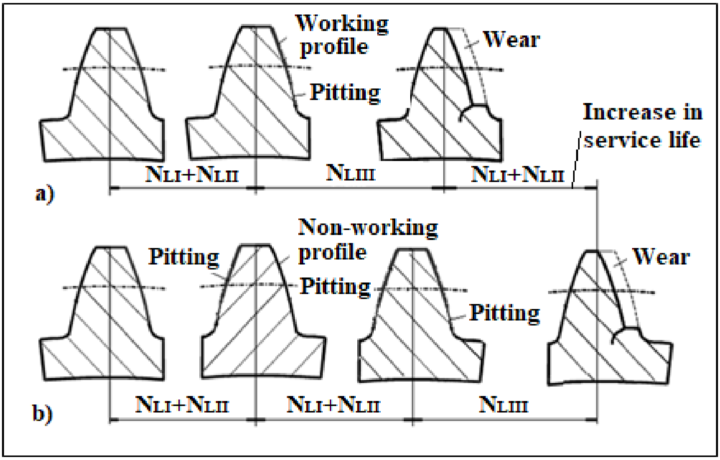

Fig. 20. The stages of occurrence of failures in the teeth of a worm gearwheel

The values of other parameters required for calculations are set according to the German standard DIN 3996. The results of the calculations are shown in Fig. 21. As can be seen from the figure, as the number of worm shaft cycles increases, the values of the relative increase in service life time also increases. This is due to the fact that at low worm speeds, the stages of formation and intensive pitting increase (N_{LI} and N_{LII}) are short, and the process of intensive wear (N_{LIII}) begins quickly. Calculations show that by replacing the working profiles of the wheel teeth with non-working profiles, when the rotation speed of the worm shaft n_1= 3000 min^{-1}, it becomes possible to increase the transmission life by 40÷55%. Obviously, the gear ratio also has a significant impact on the relative increase in service life. With a large gear ratio, the possibility of increasing the resource becomes greater. At u=20.5, it is possible to increase the resource by 25-45% depending on the rotation speed, and at u=50 - by 30-55%.

One of the main conditions that with the proposed method it is possible to increase the life of the worm gear is the absence of the possibility of tooth breakage under the influence of bending stresses. Intensive wear of the working surface of the tooth at the third stage can lead to a decrease in its thickness and, consequently, breakage as a result of loss of bending strength. For this reason, a comparative analysis of service life according to the criteria of bending and contact stresses of the worm wheel tooth is of particular importance.

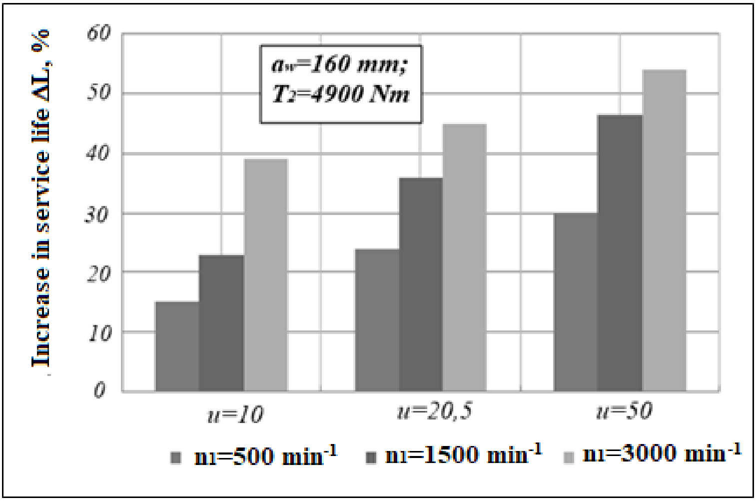

Fig. 21. The relative increase in service life time

For a worm gear with an axial distance a_w=160 mm, a gear ratio u=20.5, the values of the K_{FH} parameter are set for various values of the torque (T_2), module (m) and the number of cycles (n_1) of the worm shaft. It is established that with an increase in the load on the worm wheel, the value of the service life ratio decreases. But at nominal load limits ($T_2<T_{2max}$), the value of this parameter is greater than 2 with a

module m=5.0 mm. At large values of the module (m=6.25 mm and m=7.87 mm), the K_{FH} parameter becomes quite high. And at values of the modulus m<5 mm, bending stresses become quite dangerous and the probability of tooth breakage from the effects of these stresses becomes higher. The rotation speed of the worm shaft (n_1) also has a fairly large effect on the service life ratio K_{FH}. Under heavy loads ($T_2>T_{2max}$), the K_{FH} parameter decreases when n_1 increases.

Analysis of tests worm gears with different geometric dimensions at different values of kinematic parameters and loads shows that the stage of intensive wear of the working surface of the worm wheel is preceded by the stage of pitting formation and its propagation over the surface. In many cases, this stage can be quite large. The duration of all stages of operation depends on the load and the number of cycles. But in all cases, the service life ratio gets quite large estimates. This also allows, based on the assumption put forward, to increase the life of the worm gear by replacing the working surface of the teeth with a non-working surface after a certain period of operation.

In order to study the effect of changing the direction of the load on the bending strength of the teeth of a worm wheel made of bronze alloy, tests were carried out with the participation of the author of the presented dissertation work in the research laboratory of the Institute of Engineering and Transmission Engineering of the University of Chemnitz (IKAT), Germany. Experimental tests were carried out on a universal pulse installation with a servohydraulic testing machine of the UPM 140/95 brand (Fig.22). The device allows you to check the strength of the teeth for bending of the worm wheel of various sizes in a relatively short time.

The tests were carried out on worm gears with an axial distance of a_w=100 mm and a_w=160 mm, gear ratios u=20,5 and u=20. The material of the toothed crown of the worm wheel was bronze of the CuSn12Ni2-C-GZ brand, obtained as a result of casting by centrifugal method, and the material of the worm was steel of the 16MnCr5 brand. In order for the force on the axis to be perceived only by the test tooth, the adjacent teeth are milled. In order to check the bending strength of a tooth that has undergone wear, the tooth profile is cut off with a special milling machine with the size of the permissible wear limit. During all the tests carried out, it was recorded that the working and opposite tooth profiles do not fail even after load cycles of $5 \cdot 10^6$ when loaded in both directions. Teeth whose thickness has been reduced to the permissible wear limit (δ_{WD}=5,2 mm) have lost their bending strength before the end of the trial period and as a result have failed.

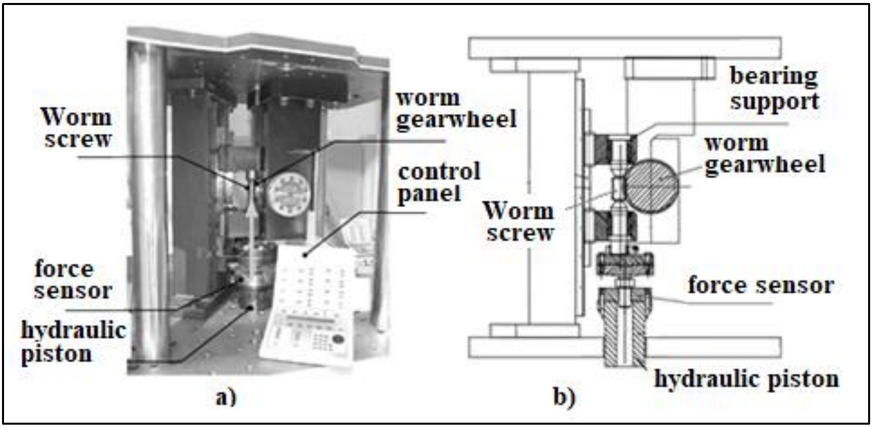

Fig. 22. Servohydraulic testing machine of the UPM 140/95 brand

Due to the fact described above, it can be stated that the lifetime reserve of most gearboxes is not fully utilized. Thus, almost a double increase in the service life of gear transmissions is possible if the working flanks of the gears would be changed after a certain operating time. In order to make the working capability of the rear flanks of the teeth usable, namely in order to realize the activation of the previously hardly loaded rear flanks and thus to increase the service life of the gearboxes, the tooth strength for bending stresses should be way more than the pitting resistance of the teeth. The design, function and operating conditions of the spur and bevel gears are very different. Therefore, the method of changing the tooth flanks for each gear must be selected in detail according to the design solution or working method of the machine. In general, the change of the tooth flanks can be realized with the following procedures:

- Change of direction of rotation of the electric motor;
- The method of complex replacement of gearboxes;
- Inversion of the gear-wheels;
- Complete reversal of shafts with wheels;
- Combined method.

In the sixth chapter, the issue of assessing the reliability of transmission mechanisms of machines and equipment by the criterion of maintainability is considered. The maintainability of machines and equipment, as well as their structural elements, is one of the main factors determining the level of their reliability. Evaluation of the maintainability indexes of a technical system is of great technical and economic importance in matters of proper organization of repair and maintenance work during operation, minimizing downtime during repair, proper management of spare parts.

In many cases, it becomes important to determine reliability indexes by the criterion of maintainability of a single transmission mechanism of a machine or equipment. It can be a gear reducer, coupling, etc. Most often these units are manufactured by a specialized manufacturer, and it is the reliability indexes of this unit that matter to the manufacturer. For the operating company, both the reliability indexes of the machine as a whole and the reliability of its individual components supplied by various manufacturers are of interest. This is especially important from the point of view of assessing the demand for spare parts during the repair process.

In the presented dissertation work, for the first time, the issue of determining reliability indexes, taking into account the recoverable and non-recoverable failures of the transmission mechanism, was considered. In the existing literature, it is accepted that all failures that occur when assessing the reliability of restored technical systems can be recoverable. In reality, this approach does not always reflect reality, that is, failures that occur during the operation of most technical devices cannot always be recoverable. Sometimes such failures occur during operation that they cannot be recoverable and the technical system is decommissioned. For example, cars and their transmission systems belong to recoverable systems. Most of the failures that occur during their operation can be recoverable with the help of repairs, but if accidental events occur (for example, an accident, fire, unprofessional operating, etc.). as a result, such failures may occur, the elimination of which will be impossible. Therefore, in the course of assessing the reliability of such systems, a separate analysis of recoverable and non-recoverable failures is an important issue.

The influence of the aging process should also be taken into account when assessing the reliability of the systems being restored. Since the restoration of the operability of many elements of the system after a certain period of operation as a result of their wear and loss of strength fatigue requires high costs, then due to the conditions of economic effect, it becomes necessary to decommission it.

Thus, for a more reliable assessment of the reliability of machines and equipment whose operability is being restored, it is important to analyze failures of various nature separately and select a more suitable mathematical model for each case. In transmission systems with restoration of operability, consideration of three different processes separately, depending on the nature of failures and whether they are eliminated or not, can play an important role in assessing reliability. These processes are reflected in the diagram of the graph of cases in figure 23. As can be seen from the figure, the first process is associated with eliminated failures. This process is characterized by the failure rate of the recoverable failures $\lambda_b(t)$ and the repair rate $\mu(t)$. When the occurrence of failures corresponds to process II, that is, their elimination is impossible for various reasons, this leads to the complete decommissioning of the technical system. This process is characterized by the failure rate $\lambda_m(t)$. Due to the aging of the parts during operation, the loss of fatigue strength and wear, the profitability of the installation decreases, its operation becomes economically unprofitable and is decommissioned. This process is characterized by the failure rate $\lambda_k(t)$.

In the simplest case, the Markov model can be used to assess reliability when the failure rate does not change depending on time. The object to be restored during operation can be in two cases: S_0 - the object is fully in operation in the case of operable and S_1 - the object is under repair to eliminate the failure. In addition, as a result of failures that cannot be eliminated, the device may be decommissioned and converted to S_2 after it becomes completely unusable or becomes obsolete and loses its effectiveness. When we assume that the failure and repair rates are constant, these parameters can be assumed as follows: $\lambda_b(t)=\lambda_b$, $\mu(t)=\mu$, $\lambda_m(t)+\lambda_k(t)=\lambda_c$.

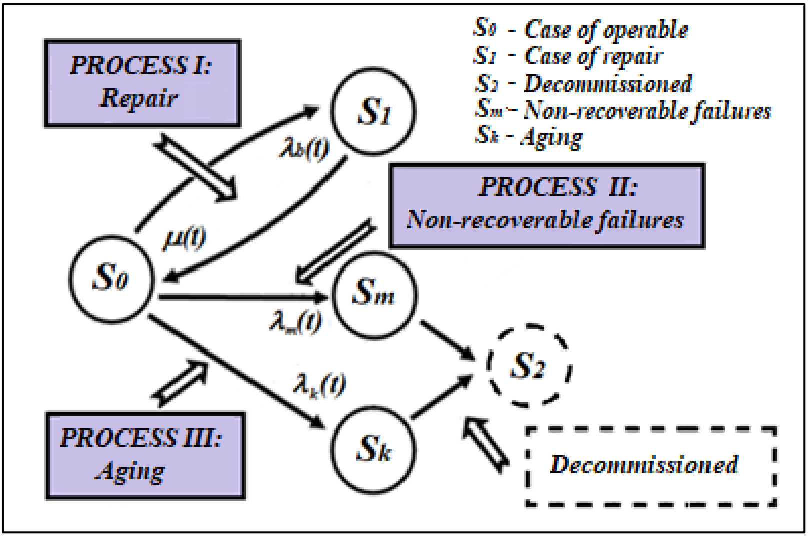

Fig. 23. Graphs of occurrence and elimination of failures

If the probability of finding a gear reducer in each state is denoted $P_0(t)$, $P_1(t)$ and $P_2(t)$, respectively, then the Kolmogorov system of differential equations for the Markov model of the system under consideration can be written as follows:

$$\begin{cases} \frac{dP_0(t)}{dt} = -(\lambda_b + \lambda_c) \cdot P_0(t) + \mu \cdot P_1(t) \\ \frac{dP_1(t)}{dt} = \lambda_b \cdot P_0(t) - \mu \cdot P_1(t) \\ \frac{dP_2(t)}{dt} = \lambda_c \cdot P_0(t) \end{cases} \quad (8)$$

Since the technical object in question can always be located only in one of three different cases, the sum of the probabilities of its being in these situations at any time of operation is equal to one. With this in mind, we can write the following normalizing condition:

$$P_0(t) + P_1(t) + P_2(t) = 1. \quad (9)$$

Since at the time of the start of operation, the gear reducer is usually in the operable state, the initial conditions can be recorded:

$$P_0(0) = 1 \text{ and } P_i(0) = 0,\ for\ i = 1, 2. \quad (10)$$

The system of differential equations (8), taking into account the normalizing (9) and initial (10) conditions for determining of the state probability of reducer, was solved using the Laplace transform. On the basis of the presented methodology, the dependence of the probability of failure of the gear reducer with the failure rate λ_b=0.5 years^{-1}, the rate of recovery μ=1 year^{-1} was established at different values of the rate of non-recoverable failures. The calculation results are graphically presented in Figure 24. As can be seen from the graph, the reliability of the gear reducer in question depends significantly on the rate of non-recoverable failures.

The issue of assessing the reliability indexes of the system (Fig.25, a), which is most often found in the drives of machinery and equipment and consists of three components – the engine, coupling and gearbox, taking into account the non-recoverable failures, is considered. We are considering also took into account the possibility of a coupling failure as a result of gearbox malfunctions. In addition, the total failure of the entire system as a result of the failure of all three components was taken into account.

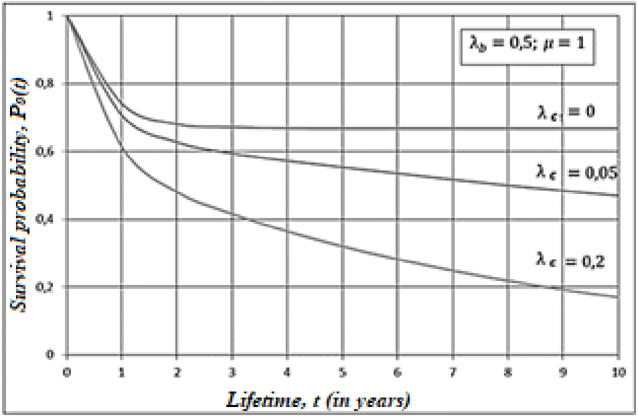

Fig. 24. The survival probability of the gear reducer on the time

Taking into account all these cases, it is possible to show a graphical representation of the Markov model of the three-component system under consideration in accordance with Fig. 25, b. Where λ_{23} is the failure rate of the coupling as a result of the gearbox failure, and μ_{23} is the repair rate of both components together. Based on the presented Markov model of the three-component system under consideration, a system of Kolmogorov differential equations can be written as follows:

$$\begin{cases} \dfrac{dP_0(t)}{dt} = -(\lambda_1 + \lambda_2 + \lambda_3 + \lambda_4) \cdot P_0(t) + \mu_1 \cdot P_1(t) + \mu_2 \cdot P_2(t) + \mu_3 \cdot P_3(t) + \mu_{23} \cdot P_{23}(t) \\ \dfrac{dP_1(t)}{dt} = \lambda_1 \cdot P_0(t) - \mu_1 \cdot P_1(t) \\ \dfrac{dP_2(t)}{dt} = \lambda_2 \cdot P_0(t) - \mu_2 \cdot P_2(t) \\ \dfrac{dP_3(t)}{dt} = \lambda_3 \cdot P_0(t) - (\mu_3 + \lambda_{23}) \cdot P_3(t) \\ \dfrac{dP_{23}(t)}{dt} = \lambda_{23} \cdot P_3(t) - \mu_{23} \cdot P_{23}(t) \\ \dfrac{dP_4(t)}{dt} = \lambda_4 \cdot P_0(t) \end{cases} \tag{11}$$

The analytical solution of the system of differential equations (11) is quite difficult. Therefore, with the help of the Matlab program, a system of differential equations (11) was solved, taking into account normalizing and initial conditions, and the dependences of the probabilities of states that the system receives were established (Fig.26). The dependence of reliability on time was established (Fig.26, b), also taking into account failures that cannot be eliminated at different values of the failure rate and recovery of the main components of the system under consideration. It can be seen from the results obtained those non-recoverable failures of transmission mechanisms have a significant impact on their reliability and do not allow maintaining the availability constant, leading to its gradual decrease. Therefore, for a more accurate assessment of the demand for spare parts, it is important to take into account failures that cannot be eliminated.

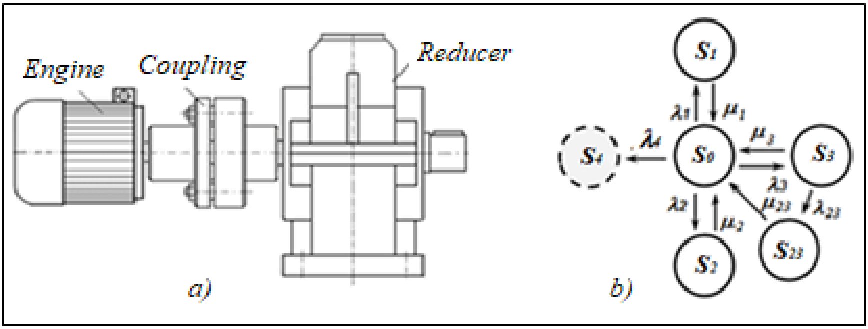

Fig. 25. Markov model of the three-component system

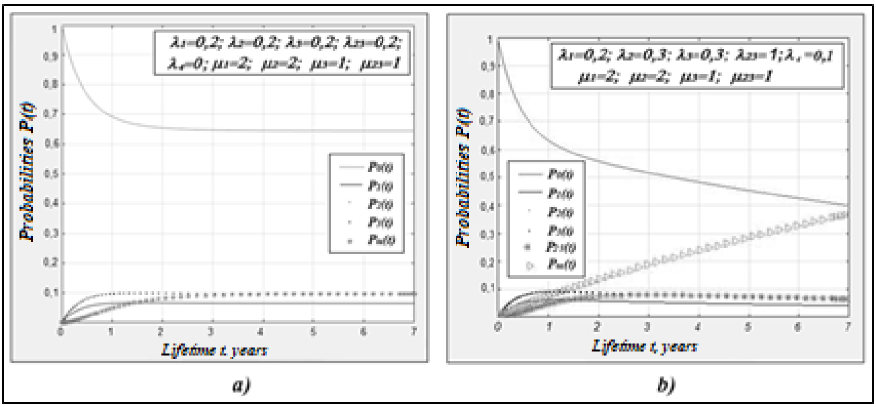

Fig. 26. Probabilities of states of the three-component system

The seventh chapter is devoted to the assessment of the technical and economic efficiency of the research. Since the transmission mechanisms of machines and equipment belong to technical systems whose operability is being restored, it is necessary to store the necessary number of spare parts in a warehouse in order to reduce downtime during repairs. The number of spare parts required for any part or assembly may vary depending on the failure rate at different stages of operation. Therefore, the correct determination of the dependence of the failure rate of parts or assemblies on which spare parts will be stored on time and based on this, maintaining the required number of spare parts at certain stages of operation is of great economic importance. For example, it is more appropriate to determine the number of spare parts required by months, quarters, years or repair cycles. Taking into account the number of necessary spare parts noted during a certain period of operation (month, quarter, year, etc.)), the number of necessary spare parts can be determined as follows[5]:

$$m_i = n \cdot N \cdot \lambda(t_i) \cdot \Delta t_i,\ if\ \lambda(t_i) \cdot \Delta t_i \leq 0{,}2; \qquad (12)$$

or

$$m_i = n \cdot N \cdot \left(1 - e^{-\lambda(t_i)\cdot\Delta t_i}\right),\ \text{if}\ \lambda(t_i) \cdot \Delta t_i > 0{,}2. \qquad (13)$$

Where Δt_i is the operational period for which the number of spare parts is determined; λ(ti) is the mathematical expectation of the failure rate in the considered operational period.

Based on the presented methodology, the number of spare electromagnetic couplings required by years during operation of N= 300 transmission mechanisms during T= 10 years was determined.

Currently, various types of sucker-rod pumping units are being operated in the oil-producing industries in Azerbaijan. In these installations, gear reducers with Novikov gearing are mainly used. Tab. 1 shows the brands of gear reducers most commonly used on

[5] Труханов В.М. Надежность технических систем типа подвижных установок на этапе проектирования и испытания опытных образцов. Монография/В.М. Труханов. Москва: «Машиностроение», -2003. - 320 с.

sucker-rod pumping units, the prices of their sale on the market and the amount of savings achieved in case of an increase in durability by about 50% due to the replacement of the working profiles of gears with non-working profiles. Considering that thousands of sucker-rod pumping units are operated in the oil and gas production departments operating in the Azneft, the cost of the economic effect achieved by increasing the durability of gear reducers can be estimated in millions of manats.

If the mutual change in the direction of movement of the descent and ascent escalators in shopping malls and metro stations does not cause any problems in the organization of passenger movement, then the replacement of the working profiles of gears with non-working profiles can be carried out in the presented way. Worm gearboxes are currently used on escalators installed by Tyssen Krupp at Baku Metro stations. Tab. 2 shows the brands of worm gearboxes used on escalators of Baku Metro stations, market prices and the amount of effect achieved with an increase in durability by about 50%. Considering that the gearboxes in question are quite expensive, increasing their durability is of great economic importance.

Tab. 1. Evaluation of economic efficiency achieved by increasing lifetime by 50% in gearboxes of sucker-rod pumping units

The brands of gear reducers	The prices of gear reducers on the market		Expected profit as a result of resource increase by 50%, in manats
	In US dollars [79]	In manats at the exchange rate (17.11.2021)	
Ц2НШ-450	16000	27200	13600
Ц2НШ-750	18000	30600	15300
Ц2НШ-315	12000	20400	10200
Ц2НШ-560	19000	32300	16150

Tab. 2. Evaluation of the economic efficiency achieved by increasing the resource by 50% in worm gearboxes used on escalators

The brands of worm gearboxes	The prices of worm gearboxes on the market		Expected profit as a result of resource increase by 50%, in manats
	In Euro	In manats at the exchange rate (17.11.2021)	
SOG-250	15000	28500	14250
SOG-320	17000	32300	16150

A comparative analysis was carried out of a new steering mechanism of track-type machine, developed and patented at the Department of "Machine Design" (currently "Mechatronics and Machine Design") of AzTU with existing gear turning mechanisms. In the newly proposed design, the number of shafts, gears, rolling bearings is reduced, and thus the total number of transmission elements is reduced from 38 to 32. This, of course, leads to a sufficient reduction in the dimensions, weight of the structure, as well as the cost of its manufacture. For both structures, an analysis of reliability indexes was carried out, suggesting that the main structural elements were manufactured by the manufacturer at the required technical level.

Assuming that the failures of the elements in both the traditional and the newly proposed design of the gear mechanism do not depend on each other, the overall reliability index was determined in accordance with the case of sequential connection of the elements. The calculations carried out showed that the reliability index of the new design is 5.4% higher than that of the traditional one. From this it can be concluded that the new proposed design of new steering mechanism of track-type machine allows not only to reduce the overall dimensions, weight and cost of its manufacture, but also to increase reliability indexes.

RESULTS AND PRACTICAL RECOMMENDATIONS

1. The proposed distribution, consisting of a superposition of exponential and normal distributions, while preserving their positive properties, is able to eliminate their disadvantage, which makes it possible to more exactly assess the reliability of machines and equipment;
2. To assess the reliability of the transmission mechanisms of modern mechatronic structures, the degree of compliance of the proposed failure distribution function with the test results of real equipment was checked, and the results were positive. The proposed failure distribution function can also be applied to the assessment of the reliability of the man-machine system;
3. In order to predict the required number of spare parts during a certain period of operation or repair cycle in order to update parts and components of transmission mechanisms of machines and equipment by choosing the correct distribution of failures of these components and determining the dependence of the failure rate on time, it is necessary that the storage of only the required number of spare parts ensures minimal storage costs, and the availability of spare parts in warehouses always leads to an intensification of the production process and an increase in labor productivity;
4. The service life ratio of gears according to the criteria of bending and contact stresses depends on the geometric and kinematic parameters of transmission, mechanical characteristics of materials. Increasing the module and wheel width allows you to increase the service life ratio;
5. Replacing the working profiles of the teeth with non-working ones after a certain period of operation, ensuring the necessary strength for bending stresses by the correct choice of geometric parameters and materials, increases the lifetime of the gears;
6. To increase the lifetime of cylindrical and bevel gears by 50-60%, and the lifetime of worm gears by 30-50%, it is possible by replacing the working profiles of the teeth of the wheels with non-working

profiles after a certain period of operation, if the lifetime according to the criterion of bending strength is sufficiently greater than the lifetime according to the criterion of contact strength

7. Calculations have proved that with an increase in lifetime by 50% by replacing working tooth profiles with non-working ones after a certain period of operation, it is possible to save 13-16 thousand manats when operating Ц2НШ brand gearboxes used on rocking machines, and 14-16 thousand manats when operating SOG brand worm gearboxes used on subway escalators;

8. Comparison of the new steering mechanism of track-type machine, developed and patented at the Department of ”Machine Design” (currently ”Mechatronics and Machine Design") of AzTU with the steering mechanisms of existing planetary gears, showed that the new design allows to reduce overall dimensions, weight and cost of manufacture, as well as to increase reliability to 5.4%;

9. Non-recoverable failures of drive components, the operability of which is being restored, seriously affect their reliability indexes and do not allow them to maintain a stable availability factor, lead to its gradual decrease. Therefore, for a more exactly assessment of the demand for spare parts, it is important to take into account failures that cannot be eliminated.;

10. As a result of calculations, it was proved that the intensity of interdependent failures of drive components does not significantly affect the reliability of the system, but the influence of the corresponding recovery intensities is quite large.

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Author's participation in published works.

Works [2, 3, 5, 8, 10, 11, 13, 15, 16, 17, 18, 19, 20, 21, 22, 23, 27, 31, 32, 33] are done alone by the autor.

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