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ABSTRACT

of the dissertation for the degree of Doctor of Philosophy

DEVELOPMENT AND RESEARCH OF INNOVATIVE MECHANICAL DRIVE OF RAILWAY SWITCH

Specialty: 3313.02 - Machines, equipment and processes

Field of science: - Technical sciences

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GENERAL DESCRIPTION OF WORK

The relevance of the topic and the degree of development.

Azerbaijan has played an important role in transportation between Asia and Europe, North and South since ancient times. The importance of railway transport in Azerbaijan, which has an advantageous strategic position in our time, has increased significantly. The Azerbaijan Railway participates in several international corridors (North-South, South-West, etc.) as a transit road, along which 177 stations of various nature and scale of work operate.

In order for the Azerbaijan Railway to be one of the integral, advanced parts of the global transport system, along with a number of important measures based on its reconstruction, it is necessary to ensure train speeds above 120 km/h, increase road capacity, and maximized the speed and reliability of railroad switch drives.

One of the components of the railway transport system are railroad switches, connecting and disconnecting railways to transfer vehicles from one track to another. They are also devices that limit the speed of vehicles.

Mechanical transmission mechanisms of electical drive of railway switch (RWS), starting from the first period of their serial production (1906), firms "Siemens", "Halske", firm "ALSTOM", electromechanical plant "Termotron" and other, also individual specialists Buryak S.Yu. , Reznikov Yu.M., Anders E., Soroko V.B., Seliverov D.I. and others improved them and developed promising designs.

As a result of the efforts of scientists and specialists working in this area, the dynamics of the work of RWS during its movement, etc. studied, with less energy consumption, the turning time of the point rails was reduced to *6* seconds (and even up to *3.5 s* in western countries), the force of pressing the point rail to the rail was increased to *6 kN* (*11 kN* in western countries), and for per cycle 10^6 of drive operating is provided transmission reliability conversion.

Reducing the energy intensity of the design, which is a requirement of the time, and increasing the speed are also relevant for electric switch drives.

In order to increase the capacity of railways for the production of trains, reduce the time of sorting work at stations, effectively organize shunting work, etc. One of the most important issues for RWS is to increase the speed of RWS, the efficiency of their use, the use of modern achievements in science and technology in the design of the mechanical part of electric transmissions, simplifying the design of its elements, their improvement and reconstruction within the framework of innovative requirements.

Since the thesis is devoted to the development and research of a progressive, innovative design of RWS, its topic is relevant.

Goals and objectives of the study. The aim of the work is to develop and study the design of an innovative mechanical transmission for RWS based on the design features of package reducers (PR).

The task of research is to increase the speed, reliability, maintainability and economy of the turnout, simplify its maintenance, develop an innovative mechanical transmission from reconstructed parts with a high level of unification, which increases the capacity of the railway line and analyze its operation.

In this work, it is supposed to determine the possibility of using a high-performance package-type reducer "AN" (PtRAN) in the mechanical transmission of RWS electric drive, to develop an innovative transmission mechanism on its basis, and to solve the most important theoretical and practical issues that ensure and determine the effectiveness of its work.

The mechanical transmission mechanism of the electric drive was adopted as the object of study. As a source of performance improvement, it was decided to convert the open spur gear of the existing gearbox to a closed mode and reduce the number of parts in the design as much as possible.

Research methods. In the development and research of innovative RWS mechanical drive, the basics of machin science, probability theory, the theory of dimensional chain(DCH)s and tribological sciences, structural and system studies of the mechanism of transmission of road switch and experimental methods were used.

The integrity of the results obtained is based on the use of general theoretical and empirical provisions, confirmed by the results of numerical and experimental experiments.

Almost all research methods were used in the work (observation, comparison, calculation, measurement, experimental and theoretical studies, generalization, induction, deduction, analysis, synthesis and abstraction).

The main paragraphs presented for defense. The mechanical drive of RWS innovative design; functional relations in PR; their use to improve the reliability of RWS; mathematical model of the angle of the difference between the directions of the teeth in engagement; PR optimization technique considering the gear ratio of the steps; the mechanism of influence of the errors of the elements of the keyway connection on the nature of the contact of the engaged teeth; the new construct of keyway connection; DCH of speed; mathematical models of parameters characterizing friction and wear of double sliding bearings; Choice of material for double sliding bearings.

Scientific novelty of the research. RWS's innovative mechanical drive at the Eurasian patent level; functional relationships of PR (speed, power, etc.) are defined; analytical expressions of friction work and surface wear in double sliding bearings were derived; the mechanism of the influence of the errors of the keyway connecting elements on the difference in the directions of the engaged teeth is explained, and the mathematical model of the difference in the directions of the teeth is given; in PR, the method of determining the intercenter distance with optimization has been developed; DCH of speed; measures have been developed to improve the contact conditions of the teeth in engagement (newly designed keyway at the patent level, moving the intersection center to the middle of the groove of keyway, etc.); double sliding bearings materials have been proposed for different friction conditions.

Theoretical and practical significance of research. Constructive-functional relations of PR are given; it was explains the mechanism of influence of the keyway connection on the operating indicators of the engagement; a mathematical model of the angle of the difference in the directions of the teeth in engagement is presented; PR optimization technique considering the gear ratio of the steps is offered; friction and wear on double sliding bearings are evaluated; a new cylindrical surface keyway is introduced.

Practical importance: Using the features of PR minimizes the number of constituent elements of the transmission mechanism, increases the speed and reliability of the switch drive, facilitates production and repair processes; the use of a cylindrical keyway connecting improves the meshing of the teeth, increases the service life of the gearing; the use of a sliding bearing in the shaft-pinion connection eliminates the negative impact of the keyway connection on the engagement conditions, the property of self-adjustment during engagement, provides an improvement in the quality of engagement; the material proposed for the sliding bearings ensures a long service life of the transmission mechanism

Approval and application.

- Lomonosov-2004: International conference of students, graduate students and young scientists of MSU. M.V. Lomonosov in fundamental sciences. Moscow, 2004;
- Republican scientific and practical conference "Modern problems of education in technical universities". Baкu, 2008;
- "Innovative technologies in education and science", dedicated to the 60th anniversary of AzTU. Republican SPC, Baкu, 2010;

-XV, XVI and XXII Republican scientific conferences of doctoral students and young scientists, Ваки, 2011, 2012 and 2018;

-International conference "Modern methods and technologies for the creation and processing of materials.", Belarus NSA. 2014;

- “I-st International Science and Engineering Konference”, Baku Engineering University, Baku, 29-30 november 2018;

-Republican scientific and technical conference of students and young researchers on the theme "Youth and scientific innovations", dedicated to the 96th anniversary of the national leader of the Azerbaijani people Heydar Aliyev. AzTU, Baku, 2019;

-«Advances in Science and Technology» 26th International multidisciplinary SPC, Moscow State University, Penza State University, 2020;

- Proceedings of the 7th International Conference on Control and Optimization with Industrial Applications, Baku, Azerbaijan, 26-28 august, 2020;

-Scientific seminars of the department "Mechatronic and machine design" ("Designing machine parts"), 2012, 2018, 2020, 2021;

-“Machine-building and Energy: New Concepts and Technologies” International Scientific-practical Conference, AzTU, Baku, Azerbaijan, December 2-3, 2021;

-International scientific conference “Priority directions of innovative activity in industry. Kazan: 30-31 january, 2022.

Azerbaijan patent (i 2021 0023 – “İşgil birləşməsi”) and Eurasian patent (№ 040109 – “Стрелочный электромеханический привод железных дорог”) were obtained for the scientific results of the dissertation work.

The name of the institution where the dissertation was carried out. Azerbaijan Technical University, department of Mechatronics and machine design.

The total volume of the dissertation with a sign indicating the volume of the structural sections of the dissertation separately.

Dissertation structure: title page (396 characters), table of contents (6369 characters), introduction (16457 characters), chapter I (45678 characters), chapter II (38628 characters), chapter III (62128 characters), chapter IV (35325 characters) , chapter V (22957 characters), conclusion (5039 characters), bibliography (26741 characters), appendices, list of abbreviations and conventions (31035 characters).

The total volume of the dissertation consists of 208 typewritten pages with 38 images, 9 graphs, 2 tables, bibliographic list of 152 titles. Excluding figures, tables, graphs, appendices and references, the thesis volume consists of 238382 characters..

MAIN CONTENT OF THE WORK

The introduction substantiates the relevance of the topic and characterizes the degree of development, indicates the goals and objectives of the study, methods, defends the main provisions, determines the scientific novelty of the study, its theoretical and practical significance, approval and application, name of the organization where the work is performed, separate and total volume of structural divisions.

The first chapter provides an overview of the research work carried out on RWS and its mechanical transmission, defines the goals and objectives of the work.

The design features and reliability of the RWS mechanical transmission mechanism, gearboxes, their main functional elements, gears, solving the issues of contact of parts, studying friction and wear of parts, increasing wear resistance were devoted to the research work of famous scientists: E.I. Danilenko, I.A. Birger, V.A. Johnson, K.B. Iosilevich, M.N. Ivanov, V.N. Kudryavtsev, G. Niemann, D.N. Reshetov, G.A. Snesarev, V. Wolf, V.Yu. Starzhinsky, F. Bowden, N.B. Demkin, V.N. Drozdov, I.N. Kostetsky, I.V. Kragelsky, A.V. Chichinadze, A.Kh. Abdullaev, A.Kh. Dzhanakhmedov, Z.Kh.

Karimov, A.M. Najafov, I.A. Khalilov and others. It is thanks to the efforts of scientists and specialists that modern achievements in these areas have been achieved. Despite the foregoing, reducing the mass capacity of the RWS, energy consumption and increasing the speed of the converter is an urgent problem.

The electrical drives of the RWS brand “СП”, their mechanical transmission system, the design and operational features of the gearboxes, including the PR, have been studied, the direction of increasing the performance of the RWS has been selected and substantiated.

Based on the results of the summary of the chapter, the relevance of the dissertation, the purpose and issues of the research are justified.

In the second chapter, the basics of developing the innovative mechanical drive of RWS are presented.

Improvement of the technical level indicators of the mechanical transmission mechanism of RWS is carried out in four directions:

1. By reducing the effect of the errors of the keyway connection elements on the directions of the engaged teeth;
2. By using the structural features of the PtRAN´s, reducing the dimensions of the reducer of RWS, the amount of processing of its parts, and the amount of service works for the structure;
3. By optimizing the distribution of the total gear ratio between the stages of the designed PR ;
4. Recommend a relatively wear-resistant material for plain bearing assemblies.

The area of the contact surface of the teeth mainly depends on the angle between the directions of engagement of the teeth. As this angle increases, the contact length and area of the teeth decrease. The difference in the directions of the engaged teeth $\Delta\boldsymbol{\beta}_0$ is determined by an arbitrary angular position in space (their tolerances) of the positions of the structural elements involved in their formation:

$$\Delta\boldsymbol{\beta}_0 = \Delta\boldsymbol{\beta}_1 + \Delta\boldsymbol{\beta}_2 + \Delta\boldsymbol{\beta}_3 + \Delta\boldsymbol{\beta}_4 + \Delta\boldsymbol{\beta}_5 + \Delta\boldsymbol{\beta}_6 + \Delta\boldsymbol{\beta}_7 + \\ + \Delta\boldsymbol{\beta}_8 + \Delta\boldsymbol{\beta}_9 + \Delta\boldsymbol{\beta}_{10} + \Delta\boldsymbol{\beta}_{11} + \Delta\boldsymbol{\beta}_{12} + \Delta\boldsymbol{\beta}_{13} = \sum_{i=1}^{n} \Delta\boldsymbol{\beta}_{\boldsymbol{i}} \,. \quad (1)$$

Where, $\Delta\beta_1$- is the deviation of the axes of the main grooves of the gear housing from parallelism, (its tolerance);

$\Delta\beta_2$ və $\Delta\beta_8$- deviations of the shaft axes from uniaxiality with the main grooves of the housing;

$\Delta\beta_3$ və $\Delta\beta_9$ - deviation of the axes of the parts of the shafts on which the gears sit from parallelism to the common axis of the shafts;

$\Delta\beta_4$ və $\Delta\beta_{10}$ - deviations of the working surfaces of the keyway of the shafts from being parallel to the axes of the shafts;

$\Delta\beta_5$ və $\Delta\beta_{11}$ - deviation of the working surfaces of the dowels from parallelism;

$\Delta\beta_6$ və $\Delta\beta_{12}$ - deviations of the working surfaces of the keyway grooves in gears from parallelism to the axes of the main grooves;

$\Delta\beta_7$ və $\Delta\beta_{13}$ - deviations of the directions of the geared teeth from parallelism to the axis of the hole of the gear shaft.

The mathematical model of the difference in the directions of the teeth (1) has two types of features: the final link $\Delta\beta_0$ is formed in the kinematic state of the component links (except for $\Delta\beta_1$); they take on a value only in the range of rotation angles corresponding to the engagement arc, and the possibility of interchangeability of the latter is not ruled out.

When the working surface of the key is not parallel to the axis of the shaft, the force perceived by it tends to be evenly distributed over the working surface, and as far as the gap δ between the contact surfaces of the gear with the shaft allows, it rotates the gear wheel until the force P is evenly distributed, and the system self-adjusts (Figure 1, a).

The contact surfaces of the teeth involved in one shaft cycle occupy different positions.

The angle between the axes of the shafts and its deviation $\Delta\beta_{14}$ are also involved in the formation of dimensional relationships between the closure link $\overrightarrow{\Delta\beta_0'}$ of DCH and the structural elements of the key connections ($\beta_{14} = 0$ is taken) (Figure 1, b; in the diagram shown: *1* - theoretical axes of shafts and gears, *2* - working surfaces of the keyways on the shafts, *3* - working surfaces of the keys, *4* - the surfaces of the working grooves of the gears), that is:

$$\overrightarrow{\Delta\beta_0'} = \Delta\vec{\beta}_4 + \Delta\vec{\beta}_5 + \Delta\vec{\beta}_6 + \Delta\vec{\beta}_{10} + \Delta\vec{\beta}_{11} + \overrightarrow{\Delta\beta_{12}} + \overrightarrow{\Delta\beta_{14}} \qquad (2)$$

Thus, the gap in the shaft-gear joint is replaced by an oblique position of the parts, the contact of the teeth occurs on one of the flanks of the teeth.

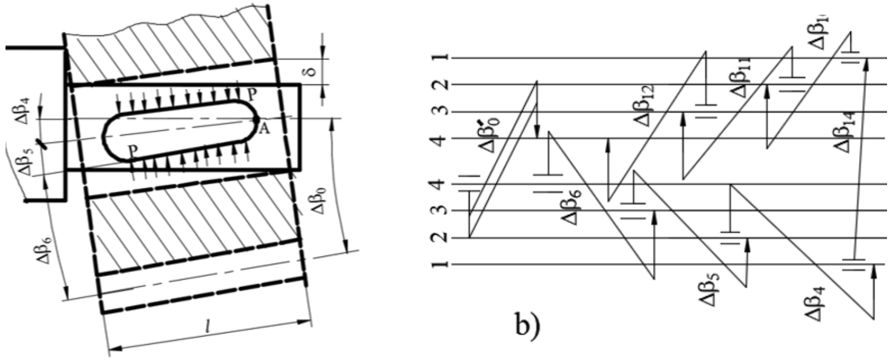

Figure 1. Schemes of the effect of the position of the key joint on the position of the engaged tooth (a) and the angular size relationships of the key joint elements (b)

The mathematical model T_0 of the tolerance for the difference in the directions of the engaged teeth, based only on the tolerances of the elements of the key connection, is as follows:

$$T_0 = \sqrt{\frac{3}{2}\left(\sum_{i=4}^{6} T_i^2 + \sum_{i=10}^{12} T_i^2\right)}\,. \qquad (3)$$

where T_i– link tolerance number *i*.

The relatively widely used (medium size) shaft-key-gear connections were investigated by numerical experiment and it was found that only the difference in the angles of the direction of the geared teeth due to possible deviations of the elements of the key connection, the deviation of the surface of the keyway from the parallelism of the axes (the accuracy of k.p.-10) was taken more than twice.

With theoretical engagement, the linear contact of the teeth is assumed on a plane passing through the axes of the shafts, corresponding to the pitch circle *1-1* (Figure 2).If in real engagement the deviation of the working surface of the key from being parallel to the shaft axis is Δ_1 along its length ℓ, and the deviation of the working surface of the keyway of gear wheel from the axis is Δ_2, then the direction of the tooth is formed depending on the state of the working connection (Δ_1+Δ_2) (Figure 2, *A, A₁*). The distance (or convergence) of the points of contact of the engaged teeth with each other along their length is formed in two directions along the *X* and *Y* axes, in the *Q* and *P* planes, taking into account the theoretical line of engagement.In accordance with the two positions of the key with a difference of 90^0 degrees from each other on the shaft, the errors of non-parallelism in the horizontal plane Q in the direction of the X axis are up to a certain angle- α_0 of the gear wheel axis (Figure 2, *A*), and in the plane P up to the angle- β_0 leads to distortions (Figure 2, A_1).

The generatrix *1-1* of the engaged driven tooth *N* in the engagement zone, in the engagement position, rotates through an angle α and takes the state *2-2*, and not the theoretical state. An error similar to the direction error of the tooth occurs. The teeth of the engagement "approach" each other in a horizontal plane along their length from point b to d. As a result, while their contact is in the areas around point *d*, their largest gap is created around point *b*.

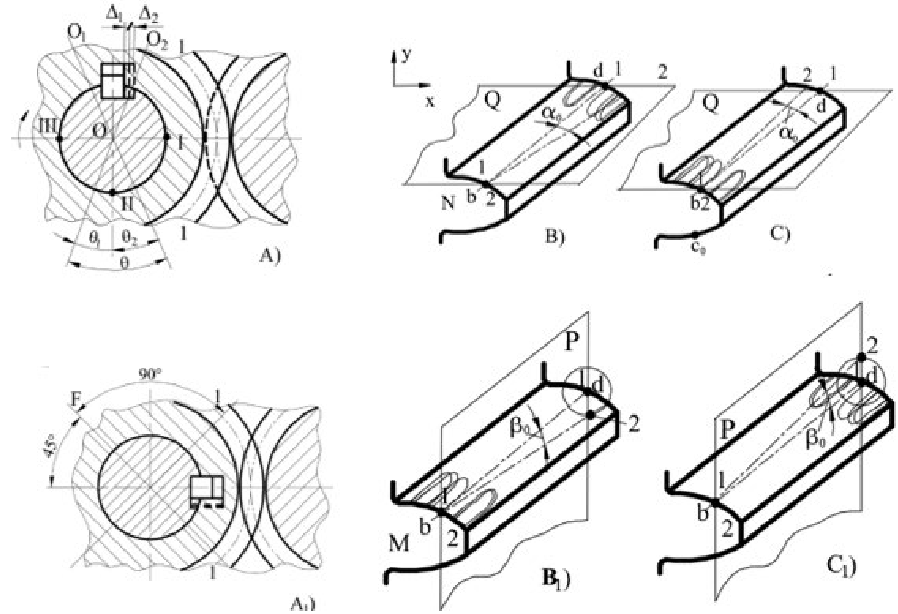

Figure 1. Scheme of the key connection and the direction of the teeth

The tooth number *N* enters the engagement when the axis of the key is turned to a certain angle θ, and exits the engagement at the position of the key OO_2 (Figure 2, A).The shape and area of the contact surface are formed parabolically on the scale of elastic deformations depending on the difference in the directions of engagement of the teeth. Since the tooth rotates smoothly continuously in engagement, the parabolic contact surface also smoothly slides from the top to the base of the tooth (and vice versa in the leading tooth) (Figure 2, *B* and B_1).

When the key joint is rotated by *90°*, the contact of the tooth *M* occurs with the same regularity on the vertical plane *P*, and in intermediate positions with a smooth transition from one position to another.

In the other two extreme cases of the key joint (when the shaft is rotated by *180^0, 270^0*), the action of the key on the clutch mechanism has a similar character.

When the axes of the key and shaft deviate from parallelism in the opposite direction, points *d* and b (Figure 2) change their functions, and the top of the cone is already at point *d*. That is, the position of the center of intersection of the axes of the key and the shaft also affects the nature of the engagement.

The task of improving the performance of the mechanical transmission mechanism by reducing the influence of the key joint on the direction of the tooth is solved in three directions:

- with the use of transmission mechanisms without a key joint (PtRAN);
- by improving the key joint;
- by controlling the position of the center of the intersection of the axes of the key and the shaft.

The location of the intersection center in the middle of the key groove provides a decrease in the amplitude of the "buildup" of the cone by half compared with the case when it is at the end of the groove.

Therefore, the design of an innovative RWS mechanical drive should be based on the above principles.

The third chapter is devoted to the development of an innovative mechanical drive RWS and the study of its structural and functional relationships.

RWS has developed an innovative mechanical drive with a three-stage gear train, which minimizes the negative impact of the key joint on gear engagement, based on the features of PtRAN's, and ensures high operational manufacturability and high production (Figure 3).

The construction and working principle of the innovative transmission mechanism is as follows: The rotational motion is transmitted from the reversible electric motor (1) through the coupling (2) to the drive shaft (7) located on the rolling bearings (13) in the gearbox housing (3), and from it to the gear wheel (9) with the shaft coupling (8) (Figure 3); rotational movement is carried out from the gear wheel (9) to the left side of the double-crown block with a freely rotating assembly structure placed on sliding bearings (15, 16) on the shaft axis (14) and to the gear wheel (19) and the left steel of the friction device, rigidly associated with the disc (18), while the right side of the block with friction gear is brought to the drive gear (21), rigidly connected to the steel friction disc (20); the driven gear (11) of the double-crown block, placed on the sliding bearing (10), transmits the movement to the heavily loaded driven gear (23) through the driven gear (12); the output end transmits rotational motion to the main shaft (25) and thus to the rack-pinion mechanism (5) of the DDD electromechanical transmission through the gear wheel (23) mounted on the rolling pillows (22) in the gearbox housing with the cam clutch (24). When the torque reaches the threshold value in the rack-pinion train (at the point rails of the RWS), the friction gear is turned off and the movement on the subsequent steps stops. If necessary, by tightening (or loosening) the nut (26), the value of the friction torque between the steel disks of the friction device is adjusted.

A comparison of the proposed transmission mechanism with electromechanical drives of the SP series, modern turnouts (gear ratio, input and output torques, etc.) and numerical tests are carried out. It has been determined that the proposed transmission mechanism has a number of advantages over the RWS type "СП" transmission mechanisms.

The transmission mechanism uses a patented key joint to compensate for the angular errors of the gear elements The keyway has a key with a design that the seat is composed of round parts, with a suitable design hole on the shaft and a prism slot on the gear wheel, and the hole of the gear wheel is adopted in a saddle shape to ensure that the axis of the gear wheel rotates at some angle around two spatial axes perpendicular to the axis of the shaft[1].

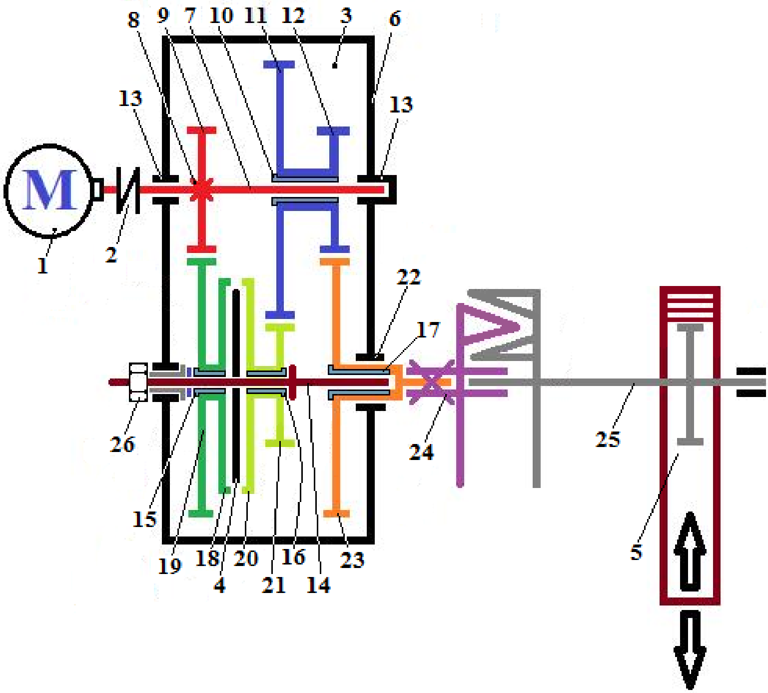

Figure 3. Scheme of the innovative three-stage gear transmission mechanism of electromechanical drive of RWS

When the gears begin to mesh, the system self-adjusts, the system becomes the one that provides the least power consumption. The result is smooth engagement, longer gear life and increased overload capability.

The transmission mechanism- PR developed for RWS, is new, and the structural-functional relations (size, speed, etc.) affecting the mechanism are of interest.

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[1] Patent, ixtira, İ 2016 0053., AR SM və Patent üzrə DK, 29.06.2016.

In PR, three groups DCH's (functional, auxiliary and free) were defined, their designation was given, DCH of speeds were developed as a type of functional connection. A closed loop of interconnected velocities that determines the velocities of several parts is called DCH of velocity . The DCH of speed of the gear transmission is represented as a three-link closed loop. (Figure 4).

Changing the speed in DCH of speed for *k* number transmission stage is given by the following formula:

$$\omega_k = \omega_1 \frac{u_k - 1}{u_1 \cdot u_2 \cdots u_k} \tag{4}$$

Where u_k - is the gear ratio of stage number k. The second product will be called the velocity change coefficient and denoted as Ks. The velocity change coefficient, is generally determined by the following expression:

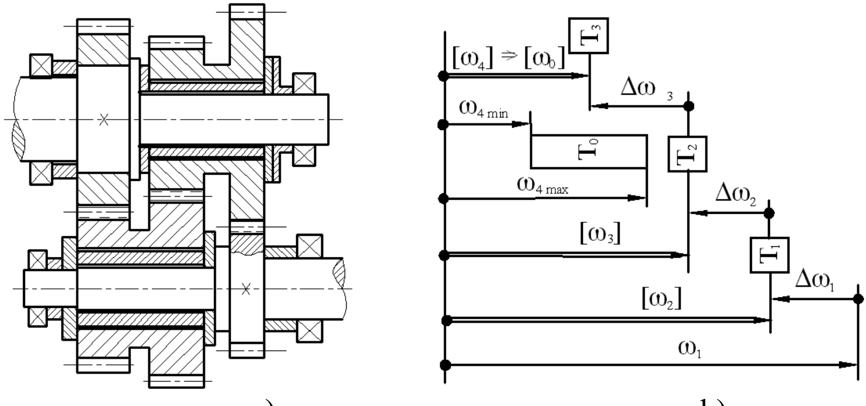
$$K_s = \frac{u_k - 1}{u_1 \cdot u_2 \cdots u_k} \tag{5}$$

On DCH of speed, output shaft speed ω_4 is the closing link, $\omega_4 \Rightarrow [\omega_0]$ (Figure 4). $\Delta\omega_1, \Delta\omega_2$ and $\Delta\omega_3$ are decremental links representing the change in speed at the corresponding stages I, II and III.

Regardless of the number of stages, PR's have the same center distance. The dimensions and accuracy of its elements are not standardized. Therefore, in PR, the ratio between the center distance and the radial clearance *[Δ]* of the gear train is of great importance. The functional radial DCH of intercenter distance is presented in three variants (Figure 5).

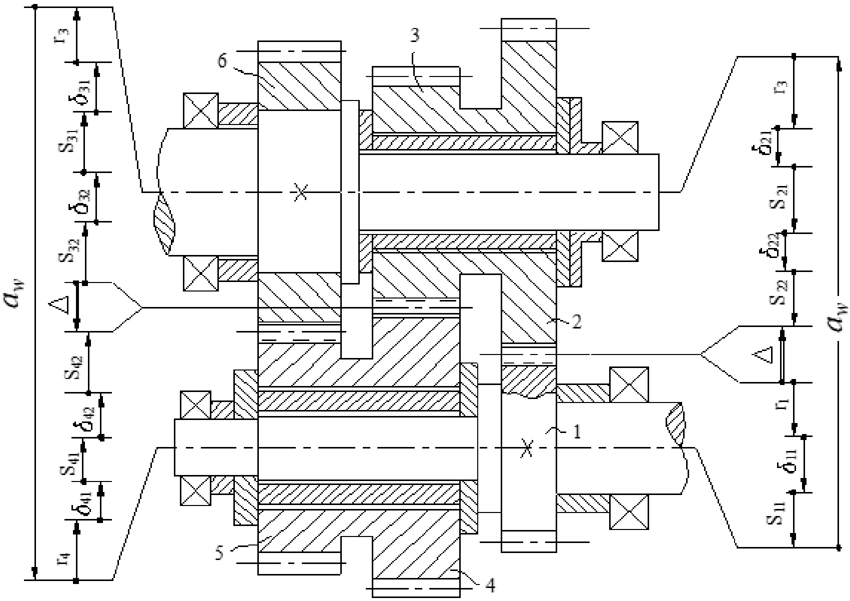
Both in gear transmission equipped with a double sliding bearings (Figure 5, a):

$$\Delta = \overrightarrow{a}_w + \overleftarrow{r}_3 + \overleftarrow{\delta}_{31} + \overleftarrow{S}_{31} + \overleftarrow{\delta}_{32} + \overleftarrow{S}_{32} + \overleftarrow{r}_4 + \\ + \overleftarrow{\delta}_{41} + \overleftarrow{S}_{41} + \overleftarrow{\delta}_{42} + \overleftarrow{S}_{42} \quad . \tag{6}$$



a) b)

Figure 4. Scheme of a gearing three-stage PR (a) and velocity formation of the closing link DCH (b)



a) b)

Figure 5. Some schemes of DCH in three-stage PR

where r- are the radii of the pitch circles of the gears, S and δ- are the distance and tolerances between the matching elements, the indices are the symbol - number of the element.

With the help of the indicated DCH, it is possible to solve various design issues, including the dimensions and allowances of the elements that make up the center-to-center distance. To ensure efficiency at the design stage, it is important that one of the components of the link in (6) be accepted as a regulatory link.

All PR stages assume the same center distance. The total number of gear ratios is divided equally between the stages. However, such a division of the transmission number has no basis. The optimization of the design of the PR in terms of transmission number is carried out in the example of a three-stage transmission.

Optimization conditions and restrictions are assumed. Based only on reports of resistance to contact stresses of the teeth, a condition is determined that provides the minimum dimensions of the gear span length (L) (Figure 5):

$$L = a_w + 0{,}5(d_{w4} + d_{w6}) \Rightarrow \min \ , \qquad (7)$$

where d_{w4} and d_{w6}- are the diameters of the corresponding gears.

The values of the total number of gears (u) at the steps (u_1, u_2 and u_3) are determined from the assumption that the number of gears varies along the geometric series and its total coefficient φ is taken ($\varphi = 1{,}02;\ 1{,}06;\ldots;\ 2{,}00$), is done:

$$u_3 = \frac{\sqrt[3]{u}}{\varphi} \ ; \qquad u_2 = \sqrt[3]{u} \ ; \qquad u_1 = \varphi\sqrt[3]{u} \qquad (8)$$

In the first approach, it is assumed that the same material is used for the corresponding parts at all steps, the tooth width factors take the same value($K_a = K_{a1} = K_{a2} = K_{a3}$, etc., where the numbers in the indices indicate that the parameter belongs to the corresponding step), all efficiency - (η_i) are assumed to be equal to one.

The general expression for PR length is obtained by applying the current report expression[2] to the gear contact stress in spur gears:

$$L_1 = B\left(\frac{\sqrt[3]{u}}{\varphi}+1\right)\cdot\frac{\sqrt[3]{\varphi^2 u\sqrt[3]{u}}}{1+\sqrt[3]{u}} + B\left(\frac{\sqrt[3]{u}}{\varphi}+1\right)\cdot\sqrt[3]{\varphi^2\sqrt[3]{u}} +$$
$$+B\cdot\frac{\left(\sqrt[3]{u}+\varphi\right)\sqrt[3]{\varphi^2 u\sqrt[3]{u}}}{1+\varphi\sqrt[3]{u}} \Rightarrow \min \qquad (9)$$

Here, since the factor B does not contain the parameter φ, it can be ignored during optimization ($B = f(K_a\,, T_i, \eta_{1,} K_{H\beta}, \psi_a, [\sigma]_H)$).

Using the «EXEL» program, the value of the product φ was determined, which provides the smallest optimal length of three-stage PtANR's with a gear ratio of $u=27$ and $72,3$.

With a value of $u=27$ of the full gear ratio, the shortest length of a traditional gearbox is provided at $\varphi=1$, and the shortest length of a three-stage PtANR is provided at $\varphi=1,06$. At the same time, the length of the PtANR is reduced by *1,4%* compared with the uniform distribution of the total number of gears between the stages, the length of the PtANR is provided by *~17,9%* compared to the traditional gearbox.

The ability to perform its task PR, mainly depends on the operation of the nodes of the double slip cushions. It is important to study the tribological regimes in the nodes in order to provide favorable values for the tribological parameters of the PR friction nodes.

The fourth chapter is devoted to issues of friction and wear of parts of multi-stage PR's.

The service life of the PR is mainly limited by surfaces that change their size and shape due to the conditions and friction regime, which differ sharply. Among such surfaces, which are classified into five groups during operation, the double sliding bearing included in

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

group II, since it is free in the working position, can rotate at different speeds depending on the operating conditions (Figure 6).

A feature of the double slide bearing is the ability to control the friction mode (bearing wall thickness ***t***, frequency of its rotation, materials of tribological pairs, etc.) and change the relative position of the sliding by controlling the friction conditions. If the wear intensity in a tribological pair is relatively high due to the sliding velocity, then it is possible to significantly reduce the speed by dividing it into two tribopairs and create more favorable conditions for friction. This issue is investigated on the plan of velocities formed in friction pairs belonging to the driving shaft (as well as the driven shaft) (Figure 6).

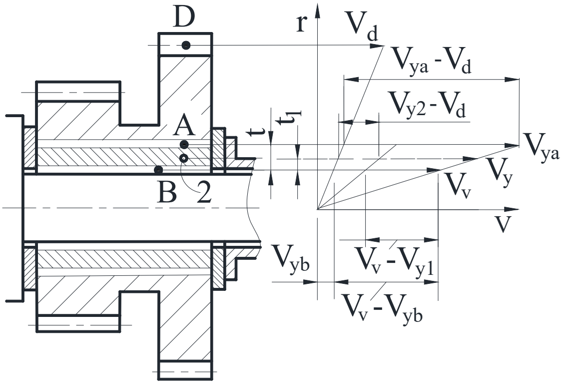


Figure 6. Speed plan for double plain bearings on a drive shaft

Relative slip occurs only on the contact surfaces of the shaft with the bearing (the area of point B), the relative slip speed (RSS) is equal to $(V_v - V_{yb})$ (V_v and V_d are the circumferential speeds of the shaft and gear, respectively).

Relative slip occurs only on the surfaces of the bearing-gear (in the areas of point A) and the RSS is like $(V_{ya} - V_d)$.

Relative slip occurs on both contact surfaces (both in the region of points A and B). In this case, the total RSS is divided into two parts, and if the cushion wall thickness is equal to t_1, then the RSS at point B is equal to $(V_v - V_{yb1})$, and at point A, the RSS is equal to $(V_{ya1} - V_d)$. Adjustment of the bearing wall thickness t_1 and the friction coefficient according to the speed plan (Figure 6) makes it possible to reduce the magnitude of the largest RSS in the friction units. Because in this case

$$V_v - V_{y1}) < (V_v - V_{yb}) < (V_{ya} - V_d) \quad , \qquad (10)$$

and $$(V_{y1} - V_{d1}) < (V_v - V_{yb}) < (V_{ya} - V_d) \quad . \qquad (11)$$

where $V_{y1},\ V_{yb}, V_{ya}$, etc. are the speeds of the shaft, bearing and gear at the respective points.

The probability of the implementation of each of these options is determined by the ratio of the values of torques (friction work) resulting from the friction forces generated at the joints. At this value, the wall thickness t_1 => min should be considered the most economical option.

Analytical expressions are given to determine the actual RSS in the friction units of the drive shaft **(Figure 6):**

I. Relative slip occurs only in the region of point *B*. In this case, when the serial number of gears in multistage PR's is *n* - even (and when n is odd, then u_{n-1}), then RSS, V_s:

$$V_s = 2 \cdot 10^{-3} \pi\, r_v\, \mathrm{n}_1 \left(1 - \frac{1}{u_1 \cdot u_2 \cdot\cdot\, u_n}\right), \frac{m}{min} \quad . \qquad (12)$$

where r_v and n_1 are the shaft radius and rotation frequency, respectively.

II. Sliding occurs only along section A, then RSS, V_s':

$$V_s' = 2 \cdot 10^{-3} \cdot \pi \cdot n_1\, (r_v + t) \left(1 - \frac{1}{u_1 \cdot u_2 \cdot\cdot\, u_n}\right), \mathrm{m/min} \quad (13)$$

III. If the relative slip occurs in both areas of points *A*, V_{sa} and *B*, V_{sb}, then with the thickness t1 of the cushion wall (Figure 6):

$$V_{sb} = 2 \cdot 10^{-3} \pi\, r_v\, (\mathrm{n}_{v1} - \mathrm{n}_{y1}) \ , \qquad \mathrm{m/min}\ , \quad (14)$$

$$V_{sa} = 2 \cdot 10^{-3} \pi\, (r_v + t_1)(\mathrm{n}_{y1} - \mathrm{n}_{d1})\ , \quad \mathrm{m/min}\ . \quad (15)$$

In the friction units of the driven shaft, a similar speed plan is created and the speed changes are investigated.

The work of friction is investigated to evaluate the work of friction units PR. Of course, although the dependence of wear during friction on the work of friction is approximately $U=f(A)$, it can be considered possible to use it in a comparative relative assessment of wear.

The friction work generated in the (second) friction connection on the drive shaft A_2, the friction work in the shaft-sliding bearing connection A_{21}, and the friction work generated in the sliding bearing -gear wheel connection A_{22} are set and compared. The question of the designer and operation is to provide the most convenient of them. If

$$f_1 r_v > f_2(r_v + t) \tag{16}$$

then slippage theoretically occurs at the junction of the bearing-gear, otherwise at the junction of the shaft-bearing, because A_{21}<A_{22}. If

$$f_1 = f_2\left(1 + \frac{t}{r_v}\right) \tag{17}$$

then slippage occurs in both joints. It is difficult to fulfill Condition (17), but this is the most convenient case. Because in this case, the part subject to the greatest wear is the double sliding bearings, and it is more cost-effective to replace it during repairs.

As a result of the work of friction, up to size of radial wear of the contact surfaces, the gears are displaced towards the axis of the shaft, the side clearance between the engaged teeth increases and the smoothness of the engagement of the teeth deteriorates. Therefore, it is important to determine and compare the radial wear of the contact surfaces. For this, the wear mass is considered to be equivalent to the work of friction.

If the drive shaft rotates by an elementary angle $d\varphi_1$ (in radians), then the sliding bearing and the gear block rotate by an angle $d\varphi_2$ (Figure 7, a), $d\varphi_1 > d\varphi_2$. Formula of work done by the friction force F in the first friction unit of the drive shaft during the operation of the shaft τ:

$$A_\tau = \int_0^{120\pi n_1 \tau} (P_2 + P_3)\, f_1 \cdot r_v (d\varphi_1 - d\varphi_2) \quad (18)$$

P_2 and P_3 are radial forces acting on the first friction unit on the drive shaft.

Taking the wear rate (Specific wear) U_0, the specific mass of the material γ, the radial wear of the assembly is found:

$$\Delta = \frac{60(P_2 + P_3) \cdot f_1 \cdot n_v \cdot \tau \cdot U_0}{\ell \cdot \gamma} \left(1 - \frac{1}{u_1 \cdot u_2}\right). \quad (19)$$

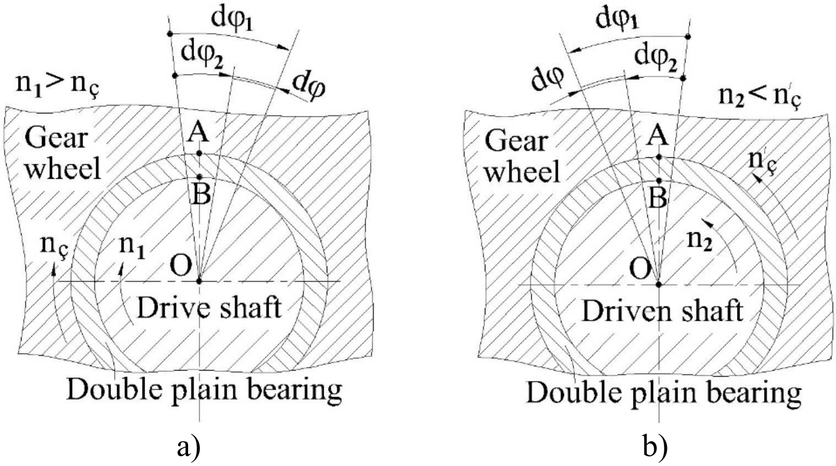


Figure 7. Schemes for determining radial wear in nodes

For the case when the relative rotational motion occurs only in the bearing-gear assembly (point *A*, Figure 7, a), the radial wear was determined from the work of friction during the time τ and compared with expression (19). It has been established that the radial wear of surfaces meeting under the same relative sliding conditions depends on the sliding conditions and does not depend on the place of the relative sliding motion.

Using a similar technique, radial wear is determined in the sliding bearing shaft-axle (driven shaft) assemblies (Figure 7, b):

$$\Delta_1 = \frac{60(P_1 + P_2)f_0 \cdot n_1}{\ell_1 \cdot \gamma}\left(\frac{1}{u_1} - \frac{1}{u_1 \cdot u_2 \cdot u_3}\right) U_0 \tau \quad . \qquad (20)$$

A variety of friction modes in the plain bearing units causes uneven wear of the units. In this time, the replacement of worn parts leads to low repair efficiency, their service life is not fully used in the parts replacement cycle.

The efficiency of using PR is ensured by uniform and minimal wear on the nodes. Under this condition:

$$\frac{(P_2 + P_3)u_3}{\ell} - \frac{P_1 + P_2}{\ell_1} = 0 \quad . \qquad (21)$$

This condition can be met even by controlling the materials of tribological pairs. For this, the following condition must be met:

$$\frac{(P_2 + P_3) \cdot f_1\, u_1 \cdot U_0}{\ell \cdot \gamma} = \frac{(P_1 + P_2)f_0 \cdot U_0^1}{\ell_1 \cdot \gamma} \qquad (22)$$

One of the factors affecting the wear resistance of friction units is the roughness of the contact surfaces. It is proposed to form an uneven surface with roughness formed on the surface during normal wear of their operation on rubbed surfaces during the production process.

The fifth chapter presents experiments on the choice of material for a double sliding bearing, their results and interpretation of the results.

Experimental examples $\Phi\text{P}_r - 1{,}8$; $\Phi\text{P}_r - 2{,}3$; $\Phi\text{P}_r - 2{,}9$. It is made from auxiliary bearings and cylinders of rotary compressors, elevator motor bearings and bronze materials MS-58 in a mold with a diameter of Ø7 mm and a length of 30 mm (Figure 8) of the following composition:

1. Auxiliary bearing material: Fe-100% iron powder;
2. Cylinder material: 0,5 % C; 3 % Cu və 96,5 % Fe ;
3. Tunegraphite (elevator motor bearing): 0,8 % C, 36 % Cu, 4 % Sn, rest bronze Бр010С1,5Ц (10 % Sn, 1,5 % Pb, up to 1%- Zn, and rest Cu);

4. Bronze MS-58 (Turkish material).

Figure 8. Test samples

The flat side surface of specimens with the indicated materials and the cylindrical surface of a rod specimen made of steel 38X2MЮA with a diameter of *Ø40* mm were taken as a friction pair.

Experiments were conducted under the following conditions: at $V_{sh} \Rightarrow 0{,}1$ *m/s*; *0,3 m/s*; *0,5 m/s*; *0,7 m/s* values of friction speeds (corresponding to the rotation frequency of the shaft sample),at *P=>1,0 MPa*; *2,5 MPa*; *4,0 MPa*; *5,5 Mpa* values of the specific pressure (SP) on the contact surface and lubricating the friction zone with oil I-40 "Industrial". For all samples, the friction track length was taken as *2,25 km*.

The number of parallel experiments was two, and according to their results, sometimes three, the reliability of the results obtained was ensured within 95% accuracy.

The wear of the stick samples was measured both by mass and by linear (direct and indirect) dimensional change.

Table 1 shows the average results obtained in the wear tests of samples under various friction modes. Consideration of the dependences of specimen wear (*U*) on the friction mode (*V, P and M* - material) (*U=f(V, P, M)*) shows that specimen wear in the accepted range of friction conditions differs by a total factor of *5,85*, and each material has a different sensitivity to changes in friction conditions.

Table 1.

The test pieces wear

Ordinal number	Relative sliding speed, *m/s*	The test pieces wear, *mm*															
		The materials of test pieces															
		Auxiliary bearing material				Cylinder material				Tunegraphite				Bronze MS-58			
		Specific pressure on the contact surface, *MPa*															
		1,0	2,5	4,0	5,5	1,0	2,5	4,0	5,5	1,0	2,5	4,0	5,5	1,0	2,5	4,0	5,5
1	0,10	1,24	1,21	1,41	1,81	0,74	0,85	0,84	1,13	0,61	0,68	0,48	0,75	0,52	0,59	0,51	0,51
2	0,30	1,31	1,32	1,46	1,87	0,86	1,01	0,98	1,49	0,60	0,73	0,67	1,07	0,71	0,85	0,77	0,78
3	0,50	1,51	1,79	1,87	2,43	1,06	1,15	1,41	1,83	0,72	0,82	1,13	1,48	0,82	0.95	0,99	1,21
4	0,70	2,15	2,20	2,49	2,81	1,22	1,55	1,76	2,36	1,05	1,09	1,51	2,17	1,24	1,21	1,19	1,57

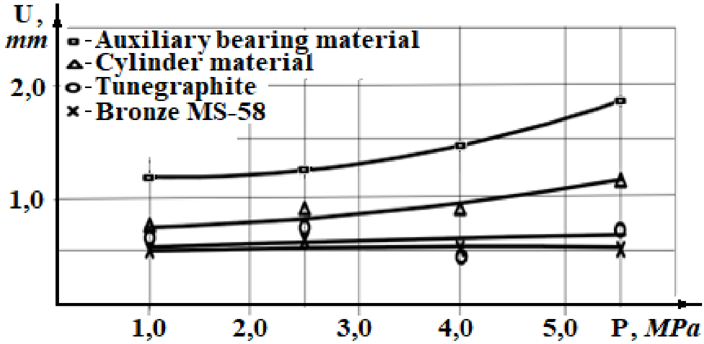
It is recommended to evaluate the change in the wear resistance of materials depending on the operating conditions using the sensitivity coefficient for material wear to the friction mode. Dividing the greatest wear of the material (U_{max}) by the least wear (U_{min}) in a given range of changes in the elements of the sliding mode is called the coefficient of sensitivity of the material to friction conditions ($\mathrm{K_h}$):

$$K_h = \frac{U_{max}}{U_{min}} \tag{23}$$

The coefficient of sensitivity to friction conditions for the studied materials was: auxiliary bearing material – *2,32*; cylinder material: *2,81*; tunegraphite: *4,52*; bronze MS-58: *3,08*. Thus, although the material of the auxiliary bearing showed the lowest wear resistance, its sensitivity to changes in friction conditions was also low.

Influence of SP on sample wear. Although the dependences of sample wear on SP (*U=f(P)*) generally follow a similar pattern, they differ sharply in quantity under different relative slip conditions:

When testing specimens from tunegraphite and bronze materials MS-58 at a relative sliding velocity of *0,1 m/s*, the dependence of wear on SP did not manifest itself and remained practically stable (Graph 1).

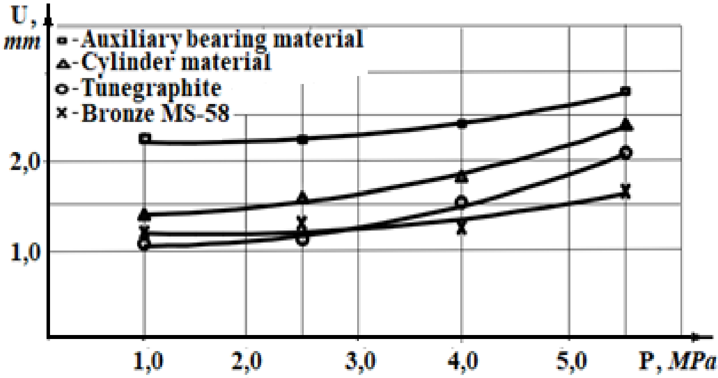


Graph 1. Dependence of specimen wear on SP at relative sliding velocity V=0,1 m/s

The dependence of the materials of the cylinder and the auxiliary bearing on the SP, despite the large range of variation of the latter (*P=1,0÷5,5 MPa*), changes in a relatively small range, along a poorly expressed parabolic curve. At a pressure of *P=5,5 MPa*, the wear of the samples increased and reached *1,81 mm* and *1,13 mm*, respectively. The consumption increased approximately *1,5* times. The wear of the sample from bronze material MS-58 even decreased somewhat *(~15%)* at high pressures. In terms of the friction condition, the cylinder material had the highest sensitivity coefficient ($K_h = 1{,}53$), and the lowest sensitivity coefficient ($K_h = 1{,}16$) - bronze material MS-58.

Dependence *U=f(P)* is studied at relative speeds of *0,3 m/s* and *0,5 m/s* according to the results presented in Table 1.

The wear of a sample made of bronze MS-58 at a relative sliding speed of 0,7 m/s is not affected by changes in SP up to P = 4 MPa. Up to the value of SP *P ≈ 2,8 MPa*, tunegraphite is the least affected by wear, and at values *P > 2,8 MPa*, bronze MS-58 is affected (Graph 2).



Graph 2. Dependence of specimen wear on SP at relative sliding velocity V=0,7 m/s

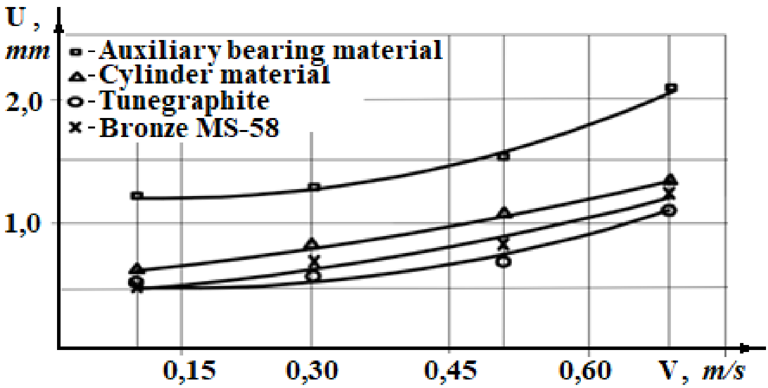
The tunegraphite material was found to be more sensitive to wear ($K_h = 2{,}07$) and the auxiliary bearing material had a relatively low sensitivity ($K_h = 1{,}3$). At a value of specific pressure *P=1,0 MPa*, the wear of the material of the auxiliary bearing was *2,05* times higher than the wear of highly wear-resistant tunegraphite, and at a value of *P=5,5 MPa*, the wear was *1,79* times higher of bronze MS-58.

So, for bushings of double plain bearings in contact with gears and shafts, made of steel material operating at speeds up to *42 m/min*, with a SP of *1,0-5,5 MPa*, it is more convenient to use bronze and tunegraphite materials.

Influence of RSS on wear. At the value of SP *P=1,0 MPa*, the wear of samples from all materials varies according to the curvilinear law (*U=f(V)*) (Graph 3).

The wear of specimens made of MS-58 bronze, tunegraphite, and cylindrical materials, their wear resistance were close at all values of RSS, but differed by approximately *30%.*

However, the wear of material of the auxiliary bearing received greater values than the wear of the other three materials, and differed by almost two times. Thunegraphite turned out to be the least worn material.

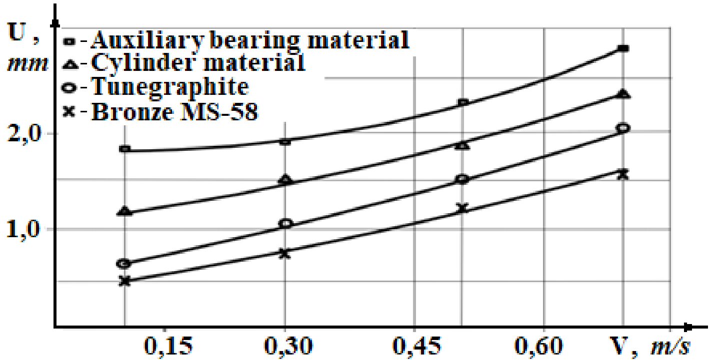


Graph 3. Dependence of wear of rod specimens on RSS (SP P=1,0 MPa)

The material of the cylinder ($K_h = 1{,}65$) has the highest resistance to wear in the RSS range, and bronze MS-58 ($K_h = 2{,}38$), which has a higher sensitivity to frictional mode - wear, provides the least resistance. The coefficients of sensitivity to the friction mode of the material of the auxiliary bearing and tunegraphite were in intermediate positions, $K_h = 1{,}73$ and $K_h = 1{,}72$, respectively.

The influence of RSS on wear at SP values *P=2,5* and *P=4,0 MPa* was also performed according to the results presented in Table 1.

At a value of SP *P=5,5 MPa*, the specimen made of bronze MS-58 showed the highest wear resistance, the material of the auxiliary bearing showed the lowest wear resistance, as in other friction modes (Graph 4). In contrast to the lower SP, the wear of a sample made of tunegraphite material under friction conditions at SP of *5,5 MPa* was also higher than that of a sample made of bronze material MS-58, by *1,47* times at an RSS value of *0,1 m/s*, *1,37* times higher at *0,3 m/s*, *1,22* times higher at *0,5 m/s*, and *1,38* times higher at *0,7* m/s.



Graph 4. Dependence of wear of rod specimens on RSS (SP P=5,5 MPa)

It was almost impossible to determine the wear of the central shaft sample.

According to the results of experiments, when the value of the SP is *1,0; 2,5* and *4,0 MPa*, and at all RSS values, tunegraphite, which has a relatively high wear resistance, and MS-58 bronze material in a SP value of more than ~ *5,0 MPa*, are recommended for use in PtANR's.

CONCLUSION

1. DCH's of keway connections of gears and a transmissions were compiled, a change in the position of the contact surfaces of the engaged teeth was found, depending on the position of the keyway connection, and its mechanism was explained [5], [8], [10], [19].

The design of a cylindrical keyway connection, which allows to exclude the negative impact of the keyway connection on the quality of engagement, is based on the lowest energy consumption of the system when the gears are engaged and which provides self-adjustment, was developed at the level of the Azerbaijan patent and is recommended for use [27].

2. The work of railway transport is studied, an innovative design of an electromechanical drive with PR is presented, which improves the quality of manufacture and operation of the RWS, improves the technical level of the mechanical transmission mechanism in four directions at the level of a Eurasian patent, and is recommended for use [1], [2], [23], [28], [29], [30].

3. DCH's, formed in the PR, are classified according to the functional feature and the mechanism of formation: functional DCH are divided into three types: DCH of gear and transmission parameters, DCH's of speed and DCH's of power. The essence of DCH of speed, formation mechanisms, their construction and calculation rules are given. To evaluate and compare the change in the closing link of DCH of speed, the concepts of the coefficient of change in speed (K_s) and the measure of change in speed (K_V) are introduced [13], [14], [15].

4. The radial DCH of the accuracy parameters of gears and stages of the PR are compiled, the DCH formed in the gear shaft assemblies with keyed connections and the shaft-bearing-gear with a double plain bearing are compared, it is found that the number of DCH constituting links in the plain bearing assemblies is much higher, despite that they are interchangeable, a high closing link accuracy is ensured [6], [7], [8].

5. Optimization of the PR in terms of the dimensions of the structure was carried out and a method was developed for determining the favorable distribution of the total number of gears between the stages. In a three-stage PtANR with a total gears ratio i=27, the optimal distribution of the gear ratio between the stages is ensured by reducing its length by 1,4% compared to its uniform distribution and by ~17,9% compared to the corresponding traditional reducer [11], [12], [25].

6. The surfaces of PR's that have relative movement and are subject to wear are classified. Formulas (12), (13), (14) and (15)) are given to determine and compare the relative velocities of the friction surfaces of a double sliding bearing and the dependence of a favorable variation of the friction coefficients on the nodes, formulas ((16), (17)) for their relationship with the speed of friction. A mechanism for controlling the occurrence of relative slip on the contact surfaces of the shaft-bearing or bearing-gear is disclosed [9].

7. The values of radial wear in the friction units ((19), (20)) are determined and compared. The radial wear of the surfaces of the elements on the units of double plain bearings depends on the sliding conditions and does not depend on the place of the relative sliding movement (sliding along the inner or outer surface of the bearing). As one of the sources of ensuring the reliability of the gearbox, it is recommended to ensure the equality of the wear for the nodes, for which a mathematical model (23) is given [24].

8. The initial production roughness of rubbing surfaces is recommended to be taken equal to the roughness formed during normal period of wear [4].

Recommended for use in PR are double plain bearings made of tunegraphite, which has a relatively high wear resistance At SP of 1,0, 2,5 and 4,0 MPa and at all RSS values and from bronze material MS-58 at a SP of more than ~5,0 Mpa. To assess the change in the wear resistance of materials depending on the operating conditions, the coefficient of wear sensitivity (23) of the material to the friction mode is recommended [17], [18], [20] , [26].

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