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DEFINITION OF PERMISSIBLE CONTACT STRESS DURING OPERATION OF HARMONIC PRECESSIONAL DRIVES WITH ROLLING ELEMENTS

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Abstract : *The development of modern engineering causes appearing of new priority tasks to solve, constant increase of power and decrease of mass and dimensions parameters, concentrated in individual machines and units, take their places among them. Taking into consideration the fact mentioned above, we have created a new progressive kind of transmission mechanisms – the harmonic precessional drives with rolling elements. The value of contact stress has the defining importance for long lasting working ability of the transmission mechanisms while operating with nominal values of workloads.*

In following article generalized method of definition of permissible contact stress for master links of harmonic precessional drives with rolling elements which occur in workloads transfer process is presented, based on the analysis of publications about harmonic precessional drives with rolling elements. Represented method considers specifics of geometry of transmission, used materials, surface hardness of master links and necessary durability of transmission.

Keywords*: harmonic drive, periodic race groove, rolling elements, slipping motion, precession.*

1. INTRODUCTION

The great number of requirements to transmission mechanisms made by different customers mainly includes increase of reliability and durability, efficiency and load ability together with decrease of mass, dimensions and manufacturing cost.

It is getting more and more difficult now to meet the requirements by partial modernization of the traditional gear transmissions. That's why the problem of making new kinds of mechanical transmissions and mesh systems is really actual.

Nowadays there is the active research and development in sphere of the movement transmission mechanisms using new type of meshing instead of gear meshing. These mechanisms transmit the torque using rolling elements, balls and rollers, which are set in the periodic race grooves. And the article is focused on this topic.

2. LITERATURE REVIEW

The first mechanisms, using rolling elements, set in the periodical race groove, instead of the gear transmission for transferring the torque, appeared in patent literature in the beginning of the last century. [1-4]. But the mechanisms didn't have practical usage and weren't wildly spread in equipment at that time. It was connected with the absence of technical ability to produce and control the periodical race grooves of high quality. The development of science and technology, appearance of CNC machines, quality control centers, computer technology etc. help to increase practical interest to the drives with rolling elements due to the higher efficiency and economy in comparison with traditional gear transmissions.
In previously published sources that contains information about harmonic precessional drives with rolling elements (in following text we use abbreviation HPDRE) [5-8] the recommendations about determination of permissible contact stress, appearing while mechanisms operating, that in complex consider the influence of the materials of which HPDRE elements are made. In the mentioned sources advantages of the precession mechanisms, possible constructive design, mechanism kinematics, and power distribution in the meshing are described.
Methods of strength calculations of mechanical drives with rolling element other constructions [9-11] can't be used for HPDRE because of the great difference in drive kinematics and main elements geometry (significant difference of space trajectories of periodic race grooves, and also units constructions of the main parts).
Thus, the given information is new, and it hasn't been researched in the works of the authors.

3. MATERIAL OF INVESTIGATION

The goal of this research is the development of generalized methods of definition of permissible contact stress for master links of single-stage harmonic precessional drives with rolling elements which occur in workloads transfer process. This methods should consider specifics of geometry of transmission, used materials, surface hardness of master links and necessary durability of transmission.
On fig. 1 original design of harmonic drives with rolling elements is presented. The transmission operates as follows: Input shaft 4 initiates wobbling motion of precessional wheel 9 by means of eccentric oblique shaft neck by the rotation. Due to this motion rolling balls 11, which are accommodated in separator 12, engaging with periodic race grooves formed in precessional wheel 9 and idler wheel 3. Precessional wheel 9 and idler wheel 3 have complementary spherically curved opposed surfaces on which periodic race grooves are formed. As the precessional wheel 9 nutates, the balls 11 successively cam the element 9 rotationally by engaging the walls of the groove. Precessional wheel 9 rotates with reduction around its own geometrical axis due to difference of periodicity of periodic race grooves. Gear ratio is determined by of periodicity of periodic race grooves on idler wheel 3 and precessional wheel 9. Rotary motion of precessional wheel 9 is transmitted to circular plate 13 which fixedly seated on the output shaft 7 by means of a pins 14 which are constantly engaged with situated in precessional wheel 9 spherical bearings 15. The engagement of pins 14 and spherical bearings 15 allows the transmition of the rotary motion of precessional wheel 9 without transmitting of his wobbling movement. The patent on this mechanism was received.

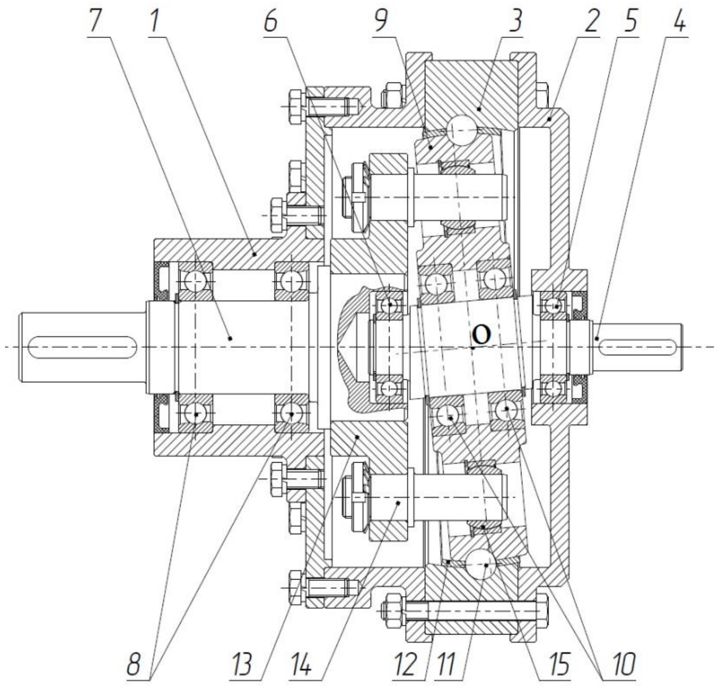

Fig.1 Construction of single-stage harmonic precession drive with rolling elements: 1 – assembled reduction gear casing, 2 – reduction gear cap, 3 – idler wheel, 4 – input shaft, 5 – radial bearing, 6 – radial bearing, 7 – output shaft, 8 – radial-thrust bearing, 9 – precessional wheel, 10 – radial-thrust bearing, 11 – ball, 12 – separator, 13 – circular plate, 14 – pin, 15 – spherical bearing, O – centre of precession

Parametric representation of path of periodic race grooves for single-stage harmonic precession drive:

$$\begin{aligned} x &= (\cos\alpha \cdot \sin\gamma - \sin\alpha \cdot \cos\gamma \cdot \cos\beta) \cdot D_b \\ y &= (\sin\alpha \cdot \sin\gamma - \cos\alpha \cdot \cos\gamma \cdot \cos\beta) \cdot D_b, \\ z &= \cos\gamma \cdot \sin\beta \cdot D_b \end{aligned} \tag{1}$$

where α – precession angle, α = [0..(2z ± 2)π], deg;

β – nutation angle, deg (β = 0.5 – 9 according to analytical analysis);

γ – angle of self-rotation, $\gamma = \alpha z / (z \pm 1)$, deg;

D_b – basic diameter – the diameter of the spherical surface, on which the path of motion of centers of the rolling elements are situated, mm.

In harmonic precession drive with rolling elements periodic race groove and rolling elements generate kinematic pair of higher degree. In this kinematic pair elements are conjugated by point. Elements of kinematic pair mounted with ability of rotation, slipping motion, rolling relative to each other.

The functioning of harmonic precession drive depends on forced friction force which appears on contact area and generates a torgue. This torgue generates motion of output link. In this case the rolling can be accompanied by significant slipping motion. The slipping motion is caused by the moment of rolling resistance, moment of inertial force as well as difference of path length of periodic race grooves which are performed on precessional and idler wheel and path length of motion of centers of the rolling elements. The last factor is caused by difference of periodicity of periodic race grooves and this is the most significant factor by its effect on value of slipping motion.

According to [6] slipping motion exercise a significant influence to kinematic pair of higher degree cycle, on its durability. In addition to loss of durability slipping motion is impacting negatively on coefficient of efficiency of mechanism

In order to determine the value of slipping motion (path length of slipping motion to total path length ratio) caused by difference of path length periodic race grooves and length path of motion of center of the rolling element, we should calculate length of these pathes. To calculate the length of an arbitrary curve, this curve should be replaced with broken line that contains all breakpoints. The sum of lengths of all broken lines is taken for the length of a curve. There is a formula:

$$L = \int_{0}^{(2z\pm2)\pi} \sqrt{(x')^2 + (y')^2 + (z')^2}\, d\alpha, \qquad (2)$$

where $x = x(\alpha)$, $y = y(\alpha)$, $z = z(\alpha)$ – periodic race groove point data.

The more smaller breakpoints we will consider, the more accurate will be the length of a curve.

Influence of periodicity of periodic race grooves and numbers of rolling elements to path of motion of centers of the rolling elements is shown on fig. 2.
L – path length of motion of centers of the rolling elements;
L_1 – path length of periodic race grooves formed on the wheel with lesser periodicity of periodic race grooves;
L_2 – path length of periodic race grooves formed on the wheel with greater periodicity of periodic race grooves.

Initial data for calculation:
D_b – basic diameter , D_b = 100 мм;
z_1 – periodicity of periodic race groove formed on the wheel with lesser periodicity of periodic race groove, $z_1 \in [10; 100]$;
z_2 – periodicity of periodic race groove formed on the wheel with greater periodicity of periodic race groove, $z_2 \in [10; 100]$;
n_{re} – number of rolling elements, $n \in [10; 100]$.

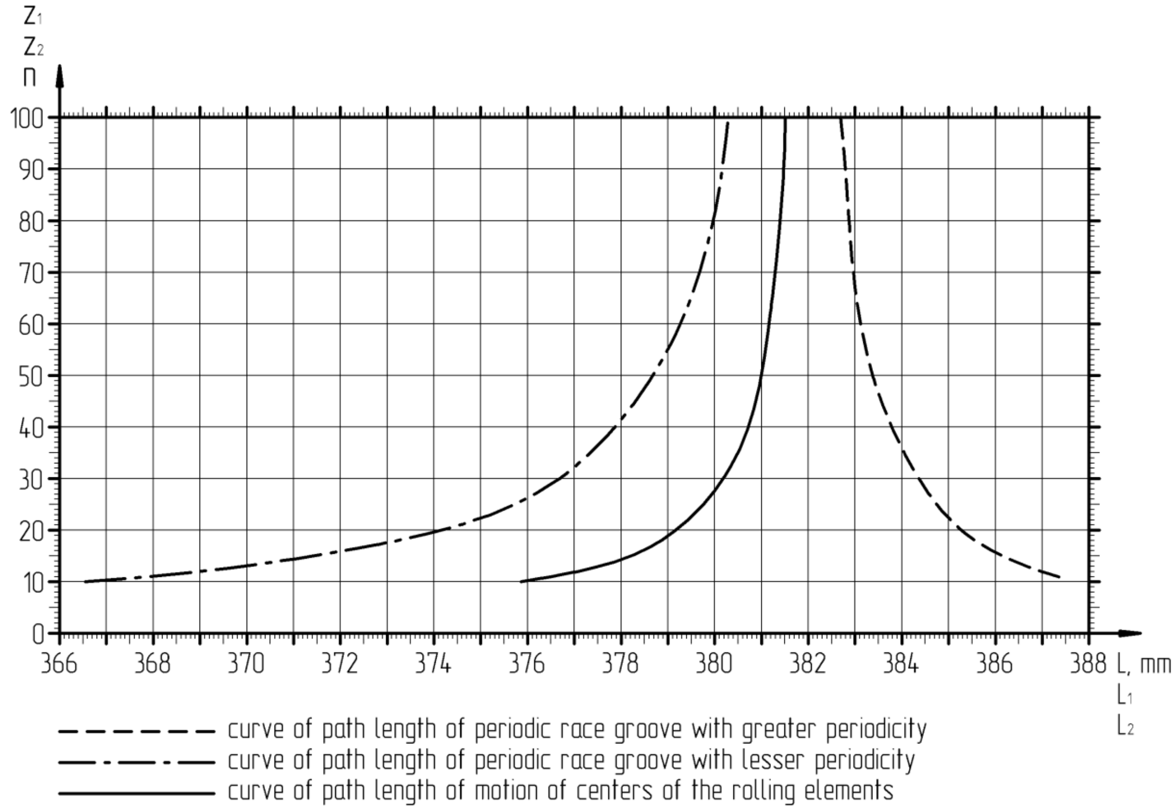

Fig. 2 Curve of dependency of path length of periodic race grooves and motion of center of the rolling elements from periodicity of periodic race grooves

The curves have a linear dependence on basic diameter. Variation of path length proportionally is proportional of basic diameter.

On fig. 3 curve of dependency of value of slipping motion which is generated by difference of path length of periodic race grooves and path length of motion of centers of the rolling elements depending on periodicity of periodic race grooves conjugating wheels and number of rolling elements is presented.

The value of slipping motion is determined by formulas:
slipping motion in engagement ball and periodic race groove formed on the wheel with lesser periodicity

$$W_{L1} = \frac{L - L_1}{L} \cdot 100\% ; \quad (3)$$

slipping motion in engagement ball and periodic race groove formed on the wheel with greater periodicity

$$W_{L2} = \frac{L_2 - L}{L_2} \cdot 100\% . \quad (4)$$

Average of slipping motion in engagement ball and periodic race groove is determined as arithmetical mean:

$$W_L = 0.5 \cdot (W_1 + W_2) . \quad (5)$$

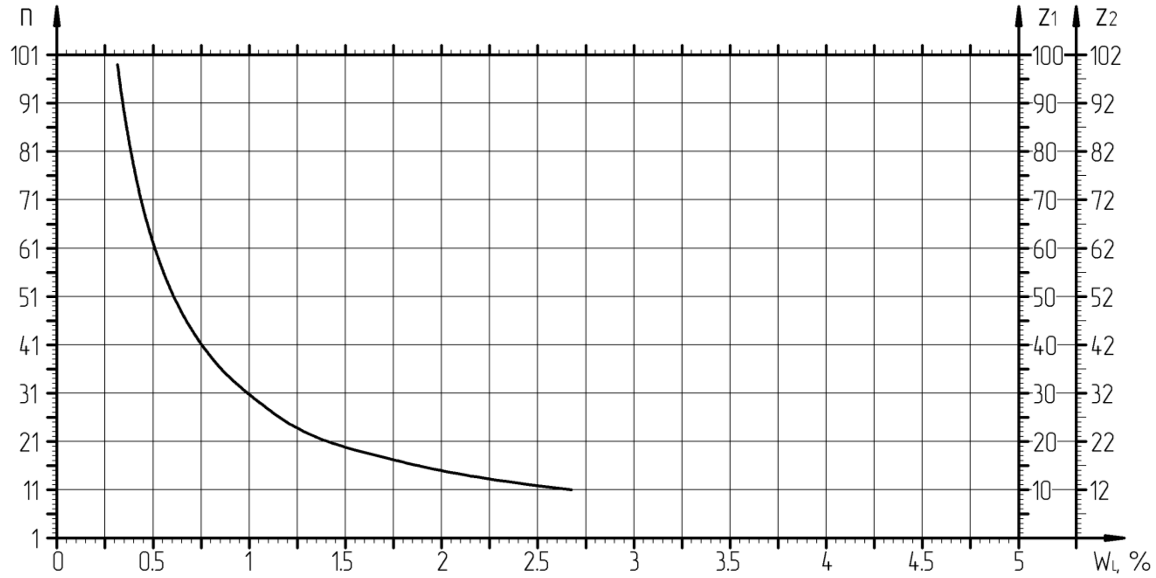

Fig. 3 Curve of dependency of average of slipping motion from periodicity of periodic race grooves and number of rolling elements

On fig. 4 difference of value of slipping motion in engagement rolling elements and wheels with lesser and greater periodicity of periodic race grooves is presented and determined by formula:

$$\Delta W_L = \frac{W_{L2} - W_{L1}}{W_{L2}} \cdot 100\% ; \qquad (6)$$

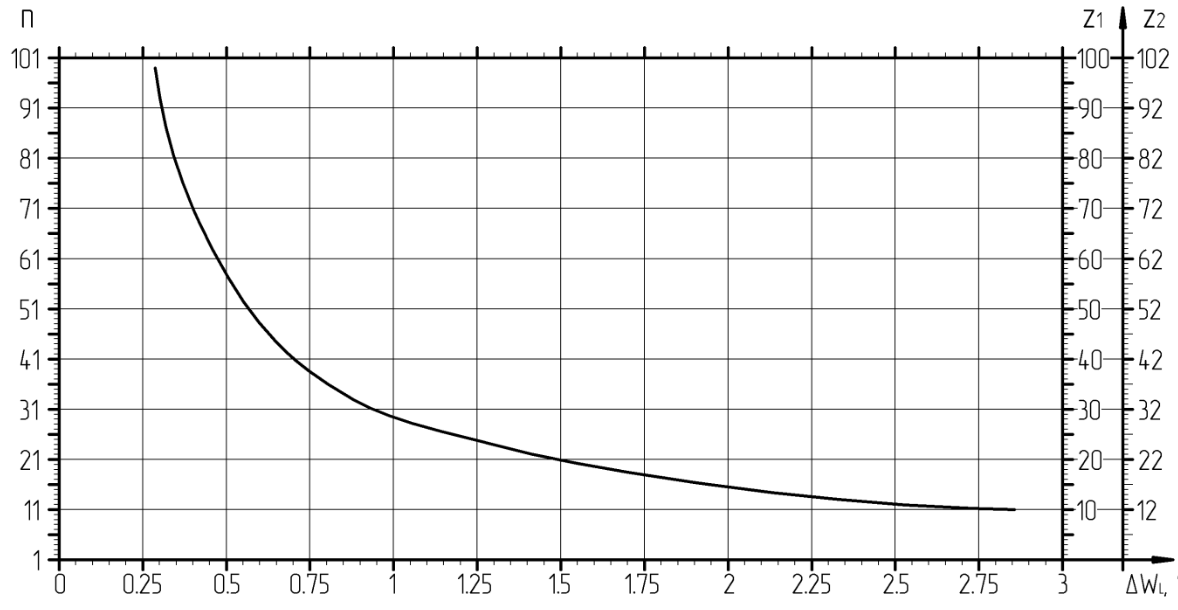

Fig. 4 Curve of difference of value of slipping motion in engagement rolling elements and periodic race grooves

Therefore in spite of difference of path length periodic race grooves conjugate wheels and length path of motion of center of the rolling element interaction of elements will occur only when rolling friction with light slipping motion is present (0.25–2.7%).

To create technique of calculation of contact strength of active surfaces periodic race grooves we consider only those factors that are presented in Hertz formula, e.g. load, quality characteristics of the material and relative curvature.

Permissible contact stress that appear while mechanism is working can be calculated with formula given in [4] considering additional coefficient that accounts the presence of slipping motion during rolling motion.

Permissible contact stress σ_{HP}, MPa:

$$\sigma_{HP} = 2800 \cdot K_T \cdot K_{HL} \cdot K_W \,, \tag{7}$$

where K_T – coefficient that depends on hardness of contacting surfaces [4] (fig. 5);
K_{HL} – durability coefficient;
K_W – coefficient of wear rate that depends on slipping velocity.

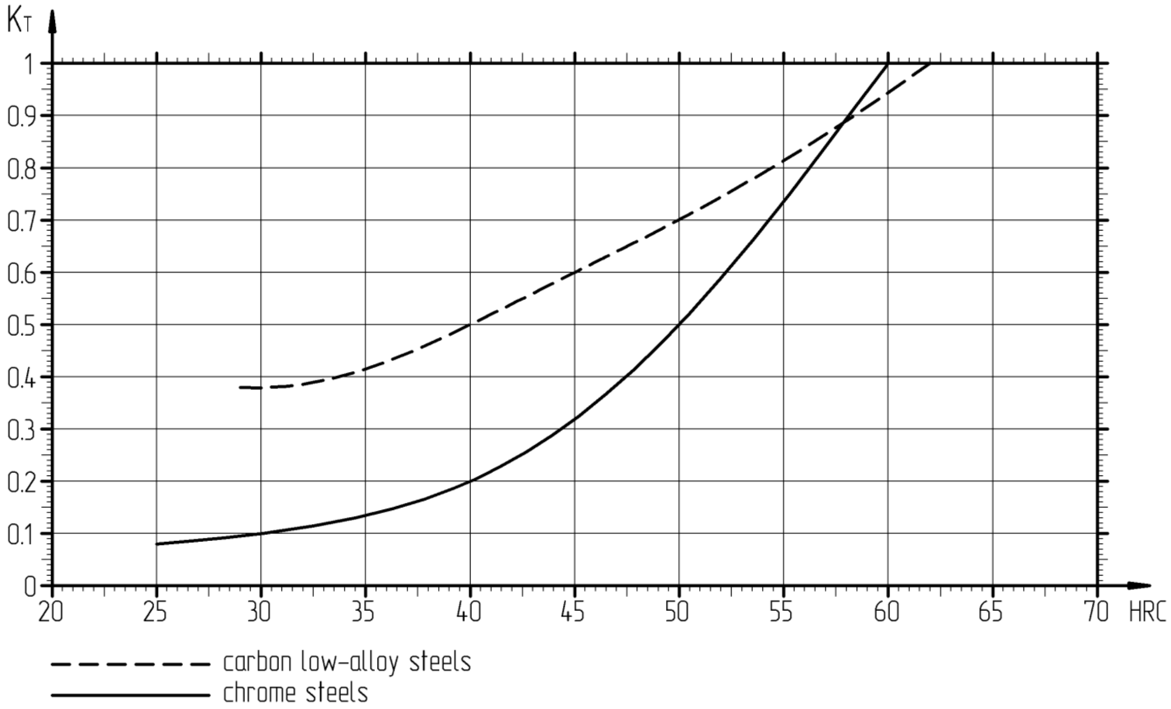

Fig. 5 Value of coefficient K_T depending on hardness of contacting surfaces of harmonic precession drives with rolling elements master links

Durability coefficient is determined by formula [7]:

$$K_{HL} = \sqrt[m]{\frac{N_{HD}}{N_{HE}}}, \tag{8}$$

where N_{HD} – basic cycle index, , correspond to continuous durability period, KHL=$11 \cdot 10^7$ [4];
N_{HE} – equivalent cycle index;
m – coefficient that accounts the hardness of contacting surfaces: m = 6 when ≤ HB 350, m = 9 when > HB 350.

$$N_{HE} = \frac{60 \cdot n_{re} \cdot n \cdot t}{u}, \tag{9}$$

where n_{re} – number of rolling elements;
n – input shaft speed, rpm;
t – durability, h;

u – reduction ratio.

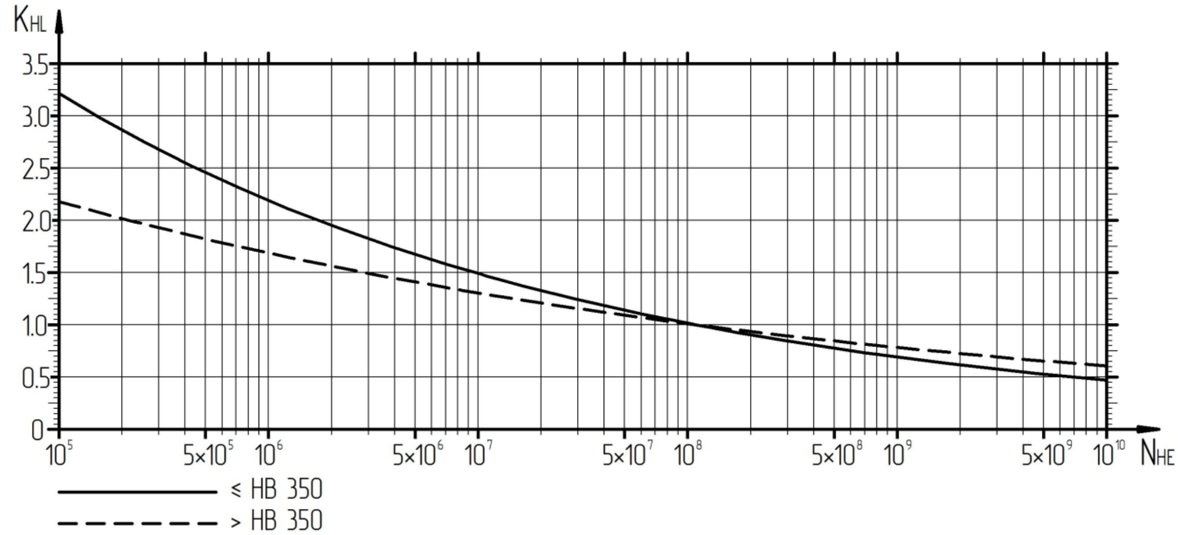

Fig. 6 Value of coefficient K_{HL} depending on equivalent cycle index

Coefficient of wear rate is determined by formula [5]:

$$K_W = 0.28 + 0.72 \cdot e^{-\frac{V_{CK}}{30.5}}, \tag{10}$$

where V_{CK} – slipping velocity, mps.

$$V_{CK} = \frac{n \cdot L \cdot W_L}{u \cdot 100000}. \tag{11}$$

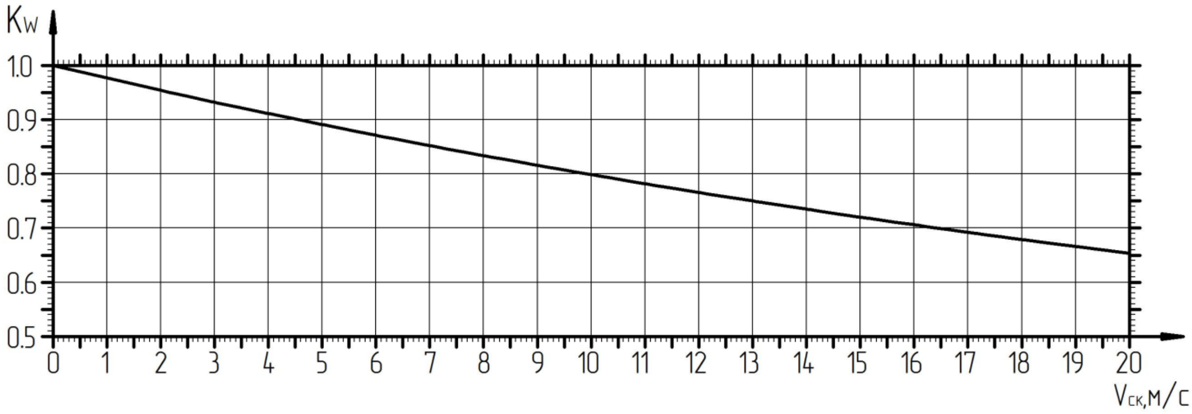

Fig. 7 Value of coefficient K_W depending on slipping velocity

According to the proposed calculation method the industrial sample of the double-reduction gear mechanism (Fig. 8) was designed and produced on the manufacturing capacity of Ltd Magma, Ukraine, Mariupol (www.magma.net.ua).

The research of the basic parameters of the industrial sample was made using methods of observation, fixation and analysis of the research results in the bench conditions of the gear working.

For evaluation of the mechanism durability and adequacy of the proposed method of detecting of the permissible contact stress, the tests with nominal power and designed number of loading cycle — 10^7 , that corresponds to 120 hours of continuous work with the frequency of in-put shaft rotation 1500 r/m, was made.

In the testing process after each 30 working hours the mechanism was disassembled, and its main parts were washed and carefully inspected to detect defects.

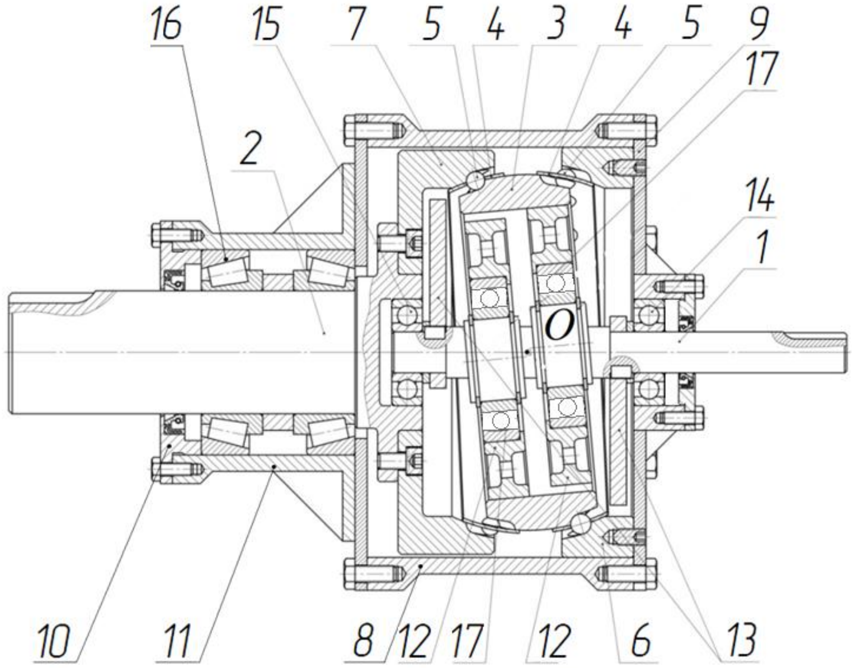

Fig 8 Industrial sample of the double-reduction gear mechanism: 1 – input shaft, 2 – output shaft, 3 – precessional wheel, 4 – separator, 5 – balls, 6 – idler wheel, 7 – idler wheel, 8 – housing, 9 – cover, 10 – cover, 11 – cartridge, 12 – disks, 13 – disks balancers, 14-15 – bearings

After 120 hours of work the mechanism kept its workability, seen defects and signs of deterioration were not detected. And there was made the decision to continue the testing until appearance of the sings of deterioration on the contact surfaces of the basic parts and until the beginning of the destruction. The testing hasn't been finished up to now.

4. CONCLUSIONS

In addition to the analytical way, the developed method is also represented in graphical way, where the target values can be detected according to the represented graphs. It lets to re-duce the time of calculation of the device not less than 50 times with the remaining of the exact-ness of the calculation in limits of 1% and to minimize possible mistakes (because the analytical calculation includes cumbersome and difficult formulas).

The made experimental researches have confirmed the adequacy of theoretical calcula-tions and possibility of its usage while designing of these mechanisms.

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