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# DESIGN OF WORM GEARS WITH OPTIMAL GEOMETRIC PARAMETERS BASED ON MINIMIZATION OF LOSSES IN GEARING

*The mathematical model of definition of losses in gearing worm gears and their decrease on 10 ... 15 % is stated at definition on the basis of planning mathematical experiment of rational parameters of transfers.*

***Keywords:*** *mathematical model, worm gears, experiment, parameters.*

**Introduction**. Gearing worm gears have wide spreading in driving gear of modern cars because of evenness and quiet work and big reduction ratio in one degree. However, they have big losses on friction in gearing of worm gears [5]. That`s why it is connected with a solving of a problem of optimal designing of machine-building constructions [9].

There are a lot of reports about researching of the problem of definition of losses in gearing worm [e,g. 1, 2, 5, 15]. In these reports it was proved that geometry of working surfaces of worm gearing has great influence on a quantity of losses in engagement. Losses in engagement in these reported are determined by experiment [5] or by liquid friction [1, 2] between working surfaces. The last one isn`t confirmed experementally as far as it was installed in the report that there is a treatment of boundary friction in worm gearing.

This report is devoted to creation of mathematical model of definition of losses on friction in points of the field of gearing worm gears in a treatment of boundary friction. There is creation of a policy of optimization parameters of worm pair for the purpose of minimization of overall losses on friction of working surfaces of worm turns and worm wheel teeth on this basis.

**Objects and problems.**

**1. Mathematical Model of definition of losses on friction in gearing**

Look at worm gears, equalization of worm turn, which is connected with it in a coordinate system $X_1Y_1Z_1$ (the axis $O_1Z_1$ is directed along the axis of worm)

$$x_1 = \lfloor f_{1(\lambda)} - R \rfloor \cos\mu ,\; y_1 = \lfloor f_{1(\lambda)} - R \rfloor \sin\mu ;\; z_1 = f_2(\lambda) + P\mu . \quad (1)$$

Here, $\mu, \lambda$ — parameter; $f_1$, $f_2$ — optional functions, describing a profile of worm turns in axial section (in further $\lambda$ in the definition of function $f_1$, $f_2$ will be missed); $R$ — radius of worm dividing cylinder; $P$ — turn parameter.

Values of function $f_1$ and $f_2$ you can define using the references of the report [10] is depends on geometry of abrasive wheel, which used in last worm processing.

Capacity of frictional force in gearing worm with worm wheel equals

$$\Delta P^* = q_n f v^{12} d\ell . \quad (2)$$

where $q_n$ is a loading on the unit of length of contact lines in gearing, directed along a perpendicular of surface (1);

$f$ – a coefficient of friction between working surfaces, determinate with using references [4, 13]; $v^{12}$ – slip velocity in gearing worm gears [6, 14]; $d\ell$ – length of contact line of working surfaces [14].

We`ll concern that there is a linear contact of working surfaces, including possible reduction of burn-in worm pair even in that case when the first contact was localized in a boundary gearing field. We`ll also propose that in the result of running-in wear of working surfaces of contact voltages by Gerts take on a fixed value on the whole line of a momentary contact line. It is possible if constant load is in gearing.

Then, using Gerts formula we`ll determine contact voltages and have

$$q_n = \frac{T_1}{X_{np}\int \frac{\{[\bar{r}_1 \times \bar{e}_1]_{Z1} + f[\bar{r}_1 \times \bar{\tau}]_{Z1}\}d\ell}{X_{np}}}. \tag{3}$$

where $T_1$ — rotational moment of worm barrel; $X_{np}$ — reduced curve of contact surfaces of worm turn and worm wheel [14]; $[\bar{r}_1 \times \bar{e}_1]_{Z1}$ — moment projection of perpendicular crosscut (1) on axis of worm; $[\bar{r}_1 \times \bar{\tau}]_{Z1}$ — moment projection of perpendicular crosscut of surface (1), directed along relative velocity vector on worm axis; $\bar{r}_1$ — vector with coordinates (1); $\bar{e}_1, \bar{\tau}$ — vectors of perpendicular crosscut and tangent, in the direction of the vector of slip velocity to surface (1), look [6, 14].

Capacity on worm barrel, made by force $q_n$ from (3) equals

$$P_1^* = q_n \{[\bar{r}_1 \times \bar{e}_1]_{Z1} + f[\bar{r}_1 \times \bar{\tau}]_{Z1}\}\omega_1 d\ell . \tag{4}$$

It is known that losses on friction in gearing are characterized by a coefficient of losses. It equals thickness ratio of friction force to Available capacity [5]. Consequently, momentary coefficient of losses on friction for a point of contact line equals (reference $\Delta P^*$ to $P_1^*$)

$$\psi_{мгн} = \frac{fv^{12}}{\{[\bar{r}_1 \times \bar{e}_1]_{Z1} + f[\bar{r}_1 \times \bar{\tau}]_{Z1}\}\omega_1}. \tag{5}$$

It follows that a coefficient of losses on friction depends on geometrical parameters of a contact point and in whole from geometry of surface of turn worm. For momentary contact line capacity of force friction with using (2) equals

$$\Delta P = \int q_n f v^{12} d\ell, \tag{6}$$

Available capacity on a worm barrel for this contact line with using (4) will have value

$$P_1 = \int q_n \{\int [\bar{r}_1 \times \bar{e}_1]_{Z1} + f[\bar{r}_1 \times \bar{\tau}]_{Z1} \int \omega_1 d\ell .\} \tag{7}$$

A coefficient of losses on friction for momentary contact line has value (reference $\Delta P$ to $P_1$)

$$\psi_{кл} = \frac{\int q_n f v^{12} d\ell}{\int q_n \{[\bar{r}_1 \times \bar{e}_1]_{Z1} + f[\bar{r}_1 \times \bar{\tau}]_{Z1}\}\omega_1 d\ell}. \tag{8}$$

Consequently, in a boundary field of gearing worm gears a coefficient of losses on friction changes when it transitions from one contact line to another. It follows there is a important conclusion that in an process in gearing worm pair with a permanent whirling moment on the shaft of worm wheel whirling moment in a boundary step in gearing on worm barrel is variable quantity. It makes variable load on motional gear. This condition is right for all kinds of gears. We`ll notice that the boundary of integration for every contact line in (8) is determined in depending on its position in the gearing field.

Now we`ll determine the overall coefficient of losses on friction in gearing worm gears. Work of friction forces in the gearing field with using (6) equals

$$A_m = \frac{1}{\omega} \sum_{i=1}^{\kappa 1} \int q_n v^{12} d\ell \Delta\varphi , \tag{9}$$

where $\Delta\varphi = \frac{\varphi_{1\kappa} - \varphi_{1H}}{\kappa_1}$;

$\varphi_{1\kappa}, \varphi_{1H}$ – angles of worm twist, corresponded to the gearing field. They are determined according to references [2, 6]; $\kappa_1$ – number of momentary lines in the gearing field. The current meaning of angle of turning is determined according to formula

$$\varphi_{1ш} = \varphi_{1H} + i\Delta\varphi \quad (i = 1,2...k_1) , \tag{10}$$

$\omega_1$ – angular velocity of worm.

The meaning of payload with using (7) equals

$$A_n = \sum_{i=1}^{\kappa 1} \int q_n \{[\bar{r}_1 \times \bar{e}_1]_{Z1} + f[\bar{r}_1 \times \bar{\tau}]_{Z1}\} d\ell\Delta\varphi . \tag{11}$$

Common coefficient of losses on friction in worm gears will equal

$$\psi_3 = \frac{A_m}{A_n} = \frac{\sum_{i=1}^{\kappa} \int q_n f v^{12} d\ell \Delta\varphi}{\omega_1 \sum_{i=1}^{k} \int q_n \{[\bar{r}_1 \times \bar{e}_1]_{Z1} + f[\bar{r}_1 \times \bar{\tau}]_{Z1}\} d\ell\Delta\varphi} . \tag{12}$$

The more meaning "$k_1$", the more coefficient of losses is bigger:

– The order of calculation of coefficient of losses is following: with using [6, 14]; we determine $\varphi_{1н}$ $u$ $\varphi_{1\kappa}$, boundaries of fields of gearing according to reference [4];

– Put the value "$k_1$";

– according to formula (10) we can determine $\varphi_{1i}$;

– for each $\varphi_{1i}$ we can determine a boundary of integration in (12) (we notice, that the line $\varphi_{1i} = const$ can consist of two filiations, as far as there is a recurrent contact of working surface of worm);

– By a formula (12) we determine of a coefficient of losses on friction in gearing of worm gear.

**2. Mathematical Model of optimization of geometrical parameters of worm gears**

By choosing a geometry of profile of worm turn we can improve the conditions of lubrication of working surfaces and reduce losses worm gear in gearing of worm pair. So, for example, using ZT – worms with concave profile of turns reduce losses in gearing on 14%, comparing to the gears, which have involute worm [12]. You can notice the same if it is the gear with convex worm in a case of choosing an acceptable coefficient of deposition. For example, if we use deposition with a coefficient of $X < -1$ for worm gears with involute worm (if number of turns $z_1^1 > 1$), we can receive the most favorable arrangement of contact lines relative to vector of sliding [11].

Loading ability and quality of worm gears are determined by Nimann, Nimann-Davis and Block criterions, except the losses on friction in gearing [2]. Contact soundness of teeth of worm wheel and heat generation in a contact zone of working surfaces are characterized by these criterions. Consequently, optimization of parameters of worm gears should be made by providing acceptable values of specified criterions and quantity of losses on friction in gearing means that it needs to use multi-objective optimization. However, we can reduce the number of criterions of optimization. It should use the planning of experiment when we decay a raw of functions. These functions determine specified criterions and quantity of losses on friction in gearing. Then we can use a method of correlation analysis. So, it is shown in work [8] that the main of criterion of optimization of parameters of worm gears is one criterion - quantity of losses on friction in gearing. The function (12) should be put in a raw, using the plan of mathematical experiment. For example, with matrix of central composition orotrabasic uniform of planning (CCOUP) the second order [7]. This raw has a view:

$$\psi_3 = b_0 + \sum_{u=1}^{k} b_u Z_u + \sum_{u=1}^{k} \sum_{l=1}^{k} b_{ul} Z_u Z_l + \sum_{u=1}^{k} b_{uu} Z_{uu}^2, \qquad (13)$$

where $b_0, b_u, b_{ul}, b_l$ – coefficients, which determined by definition on the basis of plan of mathematical experiment; $k$ – number of plan factors; $Z_u$, $Z_l$ - normalized values of factors.

We can accept four factors, they will be independent parameters: $X$ - a coefficient of deposition; q - a number of modules in dividing diameters of worms; $U$ - a reduction rate; α - an angle of turn profile on dividing diameters of worms.

Normalized factor will be denoted $Z_u$ and we`ll determine it according by formula

$$Z_u = \frac{X_u - X_{u0}}{\Delta X_u} \quad (u = 1,2,3,4), \qquad (14)$$

where $X_{u0} = \dfrac{X_{u\max} + X_{u\min}}{2}$ – an average value of factor;

$\Delta X = \dfrac{X_{u\max} - X_{u\min}}{2}$ – an interval of factor variation;

$X_{u\max}$, $X_{u\min}$ – levels of factor values.

This implies the value of the factor $X_u$ through its normalized value $Z_u$

$$X_u = \Delta X_u Z_u + X_{u0}. \qquad (15)$$

If $X_1 = X$, $X_2 = q$, $X_3 = u$, $X_4 = \alpha$ and for example, we can see warm gears with axle base $a_w = 80$ mm with involute worm and worm ZT with concave and convex turn profile, determining a coefficient of losses on friction in gearing for parameters of gears. We`ll determine a coefficient of raw (13) according to references of work [7] for chosen plan of mathematic experiment.

When we determine optimal geometrical parameters of worm gears we should pay attention to these circumstances:

– an overlap coefficient of worm pair should be bigger than one unit;
– there isn`t any tagging and undercutting of teeth in worm wheel;
– the worm flexure should be in acceptable boundaries.

When we chose the geometrical parameters the first circumstance is always fulfill. The second circumstance is provided by corresponding choice of worm deposition value and using worm gears with unequal axle step, if it is necessary. When we chose the value $q$ you should pay attention to worm flexure and soundness.

The analysis of the quantity of worm flexure with using [5] shows that acceptable worm flexure for worm gears with axle base $a_w = 80$ mm is provided with $d_{f1} \cong (22...24)$ mm ($d_{f1}$ - a diameter of worm turn cavity). The firmness of worms is provided for spreading materials and thermal treatment, which used when they were produced [3].

**3. Examples of optimization of parameters of worm gears with minimization of losses on friction in gearing.**

**3.1. Worm gears with involute worms.**

In this chapter we are going to pay attention to worm gears with gear-ratio $U = 31$. The axle base equals $a_w = 80mm$. We`ll determine geometrical dimensions according to references [5], boundaries of field engagement and contact lines with constant angular displacement will be determined according to references [2, 6].

The levels of factors, planned by mathematical experiment (for the whole factor experiment PFE), equal:

$$-0{,}75 \le X_1 = X \le 0{,}75;\ 7 \le X_2 = q \le 11;\ 29 \le X_3 = U \le 33;\ 16^\circ \le X_4 = \alpha \le 24^\circ.$$

In the result of data processing of mathematical experiment with using in given earlier models we`ve got the following coefficients of regression equation (13):

$$b_0 = 0{,}1710;\ b_1 = -0{,}0203;\ b_2 = 0{,}0264;\ b_3 = 0{,}0026;\ b_4 = -0{,}0221;\ b_{12} = +0{,}0009;$$
$$b_{13} = -0{,}0009;\ b_{14} = 0{,}0087;\ b_{23} = 0{,}0004;\ b_{24} = -0{,}0070;\ b_{34} = -0{,}0019;\ b_{11} = 0{,}0075;$$
$$b_{22} = -0{,}0005;\ b_{33} = 0;\ b_{44} = +0{,}0054.$$

These coefficients were determined rotating velocity of worm $n_1 = 1500$ t/m, $T_1 = 8$ Hm, $HB_2 = 100$ (firmness of teeth in worm wheel), $R_{a1} = 0{,}32$ (surface roughness of turn worm), $\nu = 50$ cSt (oil body), $E_{np} = 1{,}1 \cdot 10^5$ MPa (coerced coefficient of elasticity), material of worm wheel Бр010Ф1.

The most preferred worms with losses on friction in gearing are worm gears with parameters, indicated in Table 1.

Table 1.

**Optimal parameters of worm gears**

| Number of gears | $z_1'$ | $U$ | $q$ | $X$ | $m$, mm | $\alpha^\circ$ | $\psi_3$ |
|---|---|---|---|---|---|---|---|
| Gears with involute worms | | | | | | | |
| 1 | 1 | 33 | 7 | 0,75 | 3,86 | 24 | 0,1315 |
| 2 | 1 | 30 | 7 | 0,75 | 4,16 | 24 | 0,1283 |
| 3 | 1 | 31 | 5 | 4,44 | 0 | 20 | 0,1188 |
| 4 | 1 | 30 | 8 | 4 | 1 | 20 | 0,1450 |
| 5 | 1 | 31 | 9 | 4 | 0 | 20 | 0,1707 |
| Gears with ZT-type worm with concave profile of turns | | | | | | | |
| Number of gears | $z_1'$ | $U$ | $q$ | $X$ | $m$, mm | $\alpha^\circ$ | $\psi_3$ |
| 1 | 1 | 30 | 9 | 0,9 | 3,92 | 19 | 0,1194 |
| 2 | 1 | 30 | 7 | 0,9 | 4,12 | 19 | 0,1101 |
| 3 | 1 | 31 | 8 | 1,0 | 3,90 | 21 | 0,1197 |
| 4 | 1 | 31 | 6 | 0,8 | 4,15 | 21 | 0,1038 |
| 5 | 1 | 31 | 8 | 0,8 | 3,94 | 21 | 0,1201 |
| Gears with ZT-type worm with convex profile of turns | | | | | | | |
| Number of gears | $z_1'$ | $U$ | $q$ | $X$ | $m$, mm | $\alpha^\circ$ | $\psi_3$ |
| 1 | 1 | 33 | 8 | -1,25 | 4,16 | 23,5 | 0,1172 |
| 2 | 1 | 33 | 8 | -1,25 | 4,16 | 18,5 | 0,1195 |
| 3 | 1 | 31 | 10 | -2,00 | 4,32 | 21 | 0,0888 |
| 4 | 1 | 31 | 6 | -1,50 | 4,71 | 21 | 0,0812 |
| 5 | 1 | 33 | 8 | -1,75 | 4,27 | 23,5 | 0,0960 |

You can see the changing diagrams of coefficient of losses in gearing at fig. 1. It depends on $q,X,\alpha$ Gear-ratio is $u = 31$. Quantity of modules in dividing diameter of worm gearing has the main influence on friction in gearing in dividing diameter of worm. You can notice it from analysis of diagram. When quantity of modules changes in boundaries $5 \le q \le 13$ losses on friction change on 45….100%, the least values of losses with minimal value $q$ have different values $\alpha$ and $X$.

According to geometrical calculation (look [5]) the gear-ratio of worm pair is $U = 30$ with $d_{f1} = 23{,}2$, $m = 4$ and $q = 8$. For given value $q$ and number of teeth in wheel $Z_2 = 30$ we have $X = 1$. This gear has losses in gearing $\psi_3 = 0{,}145$ $\alpha = 20^\circ$ and $\psi_3 = 0{,}14$, $\alpha_n = 24^\circ$. According to OCT –2H21 – 4 – 84 worm of this gear is made by $q = 9$, $m = 4$ mm and coefficient of displacement $X = 0$. Losses in gearing of this worm gear is $\psi_3 = 0{,}17$ (look at picture 1). It means that it has 17% of losses in gearing bigger. So, using above mention regression equation we can provide the choice of parameters of worm gear according to OCT – 2H21 – 4 – 84. Parameters of worm gear $q = 8$, $X = 8$, $\alpha = 20...24^\circ$ of $u = 30$ are called optimal. They provide reduction of losses in gearing with preservation of worm firmness and inflexibility.

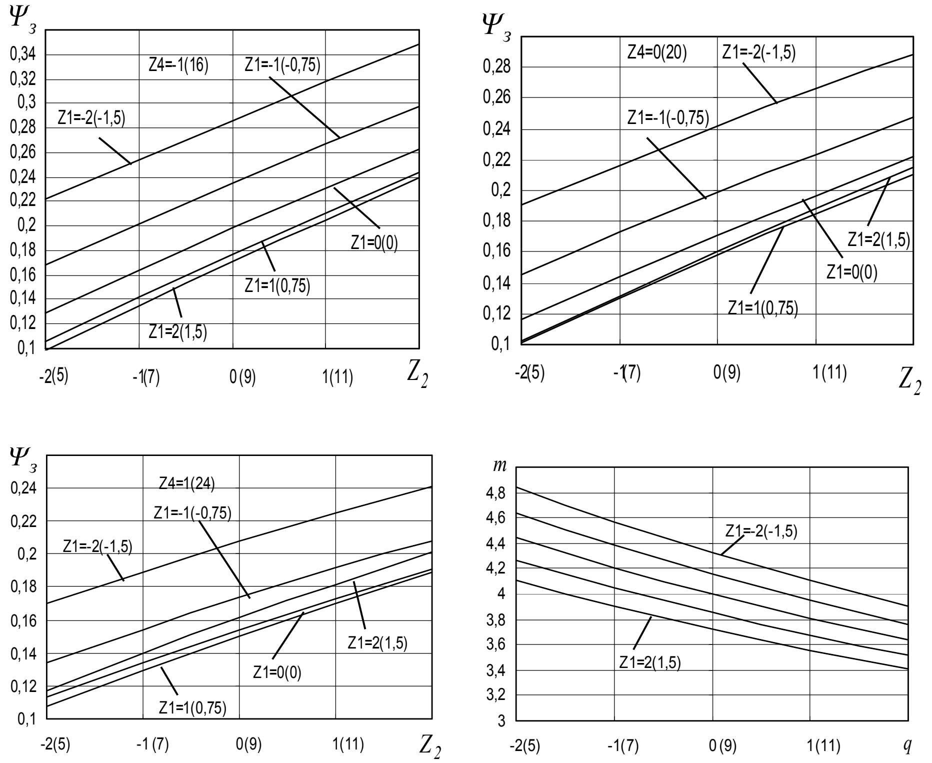


Fig. 1. Meanings of coefficient of losses on friction in gearing
(*U*=31, involute worm, there are unfixed values of factors in the brackets, m- module of engagement in mm)

**3.2. Worm gears with ZT-type worm with concave profile of turns**

We`ll look at worm gears with average gear-ratio $u = 31$ and axel base $a_w = 80$ mm. Number of worm turns is $z_1^{'} = 1$, the diameter of abrasive disc is $D_{\kappa p} = 80m$, the radius of profile arc of abrasive disc is $r_{\kappa p} = 5m$ (m is a module of gear), a coefficient of altitude of turn capping is $h_{a1}^{*} = 1$, an angle of incidence of abrasive disc on its dividing cylinder equals an hill climbing ability when a worm is grinded. According to geometrical calculation and definition of field boundaries in gearing the references [2, 6] were used.

The values of factor levels were accepted (for PFE)

$$0{,}7 \le X_1 = X \le 0{,}9\,;\; 7 \le X_2 = q \le 9\,;\; 30 \le X_3 = U \le 32\,;\; 23^{\circ} \le X_4 = \alpha \le 19^{\circ}.$$

Coefficients of regression equation (13) for definition of Coefficient of losses on friction equal:

$$b_0 = 0{,}1201;\; b_1 = -0{,}0126\,;\; b_2 = 0{,}0138;\; b_3 = 0{,}0058;\; b_4 = -0{,}0039\,;\; b_{12} = -0{,}0048\,;$$
$$b_{13} = 0{,}0030\,;\; b_{14} = 0{,}0078\,;\; b_{23} = 0{,}0051;\; b_{24} = -0{,}0029\,;\; b_{34} = -0{,}0027\,;\; b_{11} = 0{,}0103\,;$$
$$b_{22} = 0{,}0040;\; b_{33} = 0{,}0008\,;\; b_{44} = 0{,}0061\,.$$

According to diagram and regression equation it was installed that the most preferable gears on losses on friction in gearing are worm gears with parameters in table 1.

Looking through data analysis (table 1) you can notice that the area of optimal value of worm gears with ZT- worms with concave profile of turns when $u = 31$ is determined by values:

$$6 \le q \le 9\,;\ 0{,}8 \le X \le 1\,;\ 19^\circ \le \alpha \le 21^\circ .$$

Losses in gearing of these gears are less in 1,42…1,64 times than losses in gearing of worm gear with involute worm according to OCT 2H21-4-84. From condition of worm firmness and inflexibility the most optimal are gears 3 and 5. These gears have coefficients of losses in gearing. They have 83% of coefficients of losses in gear with involute worm with the same parameters (table 1, gear 4, it coincides with information of work [12] (According to this work, losses in gearing of worm gear with ZT-worm consist of 86% of losses in gearing with involute worm).

It follows that when we project worm gears with TZ-type with concave profile of turns, it is necessary to pay attention to these points:

- minimal from condition of worm firmness and inflexibility value $q$;
- positive value of coefficient of deposion in boundary $0{,}8 \le X \le 1$;
- the value of profile angle of instrument $19^\circ \le \alpha \le 21^\circ$.

**3.3. Worm gears with ZT- worm with convex profile of turns**

We`ll consider at worm gears with average gear-ratio $u = 31$ and axel base $a_W = 80$ mm. Number of worm turns is $z_1' = 1$, the diameter of abrasive disc is $D_{кр} = 80m$, the radius of profile arc of abrasive disc is $r_{кр} = 3{,}75m$, a coefficient of altitude of turn capping is $h_{a1}^* = 1$, geometrical sizes of worm and worm wheel will be determined according to reference for gears with involute worm [5]. The axis of grinding coincides with axis of gearing of worm pair [6].

The values of factor levels with mathematical experiment equal (for PFE)

$$-1{,}75 \le X_1 = X \le -1{,}25\,;\ 8 \le X_2 = q \le 12\,;\ 29 \le X_3 = U \le 33\,;\ 18{,}5^\circ \le X_4 = \alpha \le 23{,}5^\circ .$$

The values of regression equation (13) for definition of losses on friction in gearing is equal:

$$b_0 = 0{,}1172\,;\ b_1 = 0{,}0143\,;\ b_2 = 0{,}0121\,;\ b_3 = 0{,}0016\,;\ b_4 = -0{,}0026\,;\ b_{12} = 0{,}0043\,;$$
$$b_{13} = -0{,}0009\,;\ b_{14} = -0{,}0010\,;\ b_{23} = 0{,}0002\,;\ b_{24} = 0{,}0002\,;\ b_{34} = 0{,}0012\,;\ b_{11} = 0{,}0011\,;$$
$$b_{22} = -0{,}0006\,;\ b_{33} = -0{,}0008\,;\ b_{44} = 0{,}0006 .$$

According to regression equation it was installed that the most preferable gears on losses on friction in gearing are gear parameters, which you can see in table 1.

Looking through data analysis (table 1) you can notice that the area of optimal value of worm gear are determined by values: $6 \le q \le 10$; $-2{,}00 \le X \le -1{,}25$; $18^\circ \le \alpha \le 21^\circ$. Losses in gearing of these gears are less in 1,42-2,09 times than losses in gearing of worm gear with involute worm according to OCT 2H21-4-84.

**Conclusions.**

1. The mathematical model of definition of losses on friction in worm gears was developed. This model considers the values in different points of contact patch.

2. The mathematical model of function expansion was developed. It determines losses on friction in gearing of worm gears according to planning of mathematical experiment.

3. Optimal parameters of worm gears were determined when $u = 31$ with involute worms. Quantity of losses on friction in gearing is less in 1.17 times than gears according to OCT.

4. Optimal parameters of worm gears were determined when $u = 31$ with ZT-worms. Quantity of losses on friction in gearing is less in 1,42-2,09 times than gears according to OCT.

5. The findings can be used when projection of worm gears with optimal geometrical parameters with minimization of losses on friction in gearing is made.

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## ПРОЕКТУВАННЯ ЧЕРВ'ЯЧНИХ ПЕРЕДАЧ З ОПТИМАЛЬНИМИ ГЕОМЕТРИЧНИМИ ПАРАМЕТРАМИ НА ОСНОВІ МІНІМІЗАЦІЇ ВТРАТ В ЗАЧЕПЛЕННІ

Геометрія робочих поверхонь черв'ячних передач має великий вплив на величину втрат у зчепленні. Представлену роботу присвячено створенню математичної моделі визначення втрат при терті в точках зачеплення черв'ячних передач на основі граничного тертя. На основі планування математичного експерименту та підбору раціональних параметрів передач представлена математична модель визначення втрат у зубчастих черв'ячних передачах обгрунтовує можливість їх зниження на 10 ... 15%.

**Ключові слова:** математична модель, черв'ячні передачі, експеримент, параметри.

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