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GEARING WITH INCREASED TEETH WEAR RESISTANCE

The most important criterions of gearing efficiency are teeth wear resistance. The order of synthesis of rack cutter instrument for manufacture of gears with increased teeth wear resistance is considered. It is based on the model of teeth wear of closed gearing that works in oil reservoir. The differential equation is derived; its solution determines geometrical parameters of original profile. The parameters depend on setpoint of comparative teeth wear of synthesizing gearing and involute gearing. The finding solution is used for creation of equation of working teeth surfaces in toothing.

Key words: *gear-drives, teeth, teeth wear, geometrical parameters, increased wear resistance.*

Introduction. In modern conditions the enterprises of different branches feel a need for high-quality reliable and durable gear-drives, which are one of the most critical part of modern machine. The capacity for work is mostly determined by indexes of driving gears. That`s why the perfection of gear-drives, incoming into the problem of multicriterion synthesis of machine-building structure, is actual.

During many decades the geometry-kinematical criterions such as relative velocity, summary velocity of roll working surfaces, reduced curvature, specific slips [2; 3; 4] are used for estimation of capacity for gear work. Besides these criterions the integrates criterions are used, too. They are contact stability, wear criterion, criterion of loses in toothing, criterion of thickness of oil film in the place of teeth contact, temperature criterion of jam, specific work of force friction [3; 4].

Last years the topic of synthesis became very actual. The range of reports is devoted to it. For example [4; 5]. Using the results, it is possible to produce gears with high meanings of every given criterions. The synthesis is made according to one of these criterions, and others are used for comparative analysis.

In work [4; 5] there are the results of synthesis of gears with meanings of geometry-kinematical criterions with sequential analysis of integrates criterions. However, it is possible the synthesis of original loop geometry directly with meanings of integrates criterions.

One of the most important criterions of gearing efficiency is teeth wear resistance. Especially if it is open gearings, where increased teeth wear is noticed [6]. However, teeth wear cannot be excluded in conditions of boundary friction, even they work in closed oil bath. That is why it is important to pay attention in solving task of estimation teeth wear closed gearings and development teeth geometry of gearings with increased wear resistance, characterizing of reduced wear of working teeth surfaces.

Objects and problems. The purpose of article is the determination of functional correlation between geometrical parameters of basic rack for spur gear and wear criterion.

The value of teeth wear for spur gears will be the meaning [4; 6].

$$h_u = \Omega_u f^{t_y} \eta \quad . \tag{1}$$

Where: f – the coefficient of sliding friction in toothing; η – the specific sliding; Ω_u – the parameter, which is not depended on geometry of working teeth surfaces; t_y – the parameter of curve friction weariness.

According to the equality (1), we can notice that we can reduce the value of teeth wear if we reduce the coefficient of sliding friction and specific sliding. Look at the task when we determine the geometry of working teeth surfaces of spurs with reduced value wear in comparing with involute teeth. According to (1) the value wear of involute teeth will equal to

$$h_{ue} = \Omega_u f_e^{t_y} \eta_e \quad . \tag{2}$$

Where: f_e – the value of coefficient of sliding friction in involute teeth; η_e – the specific sliding in toothing of involute teeth.

The ratios of wear values (1) and (2) which use the results of work (2) will equal

$$\overline{h}_u = \overline{x}^{0,3t_y+0,5} \varsigma^{0,3t_y-1,5} \sin^2 \alpha_e \quad , \tag{3}$$

where $\overline{x}$ – relative reduced curvature of working surfaces of gearing with increased unknown gearing in comparing with involute gearing. It equals to [5]

$$\overline{x} = \frac{\left(\varsigma - f_1 \varsigma'\right)^2}{\varsigma^3}, \tag{4}$$

$\varsigma = \sin\alpha$.

In these equalities: f_1 – the variable parameter; α – the profile angle of original profile in unknown gearing; α_e – the profile angle of original profile instrument of involute gearing; ς' – the derivative function s according ς to f_1.

The equality (4) together with (3) has given $\overline{h}_u$. It is a differential equation. The answer is determined the current profile angle of original profile $\overline{h}_u = const$ (the meaning $\overline{h}_u \langle 1$ shows how many times the wear value is less than the wear value of involute gearing ($\varsigma = \varsigma(f_1)$):

$$\varsigma = \frac{f_1}{\left(\sqrt{c^{1+\beta}} + \sqrt{\chi_o f_1^{1+\beta}}\right)^{\frac{2}{1+\beta}}}, \quad c = \frac{f_{10}\left(1 - \sqrt{\chi_o \varsigma_o^{1+\beta}}\right)^{\frac{2}{1+\beta}}}{\varsigma_o},$$

$$\chi_o = \left(\overline{h}_u / \sin^2 \alpha_e\right)^{\frac{1}{\alpha_1}}, \tag{5}$$

$$\beta = \frac{-0,3t_y + 1,5}{0,3t_y + 0,5}, \quad \alpha_1 = 0,3t_y + 0,5, \quad \varsigma_o = \sin\alpha_o$$

α_o – the profile angle of original profile in unknown gearing if $f_1 = f_{10}$.

The equation of original profile in initial gearing will be offered in the form of series (axis $O_p f_2$ (picture 1) has the direction along the initial straight line)

$$f_2 = f_{20} + f_{20}'(f_1 - f_{10}) + \frac{1}{2} f_2''(f_1 - f_{10})^2 + \frac{1}{6} f_2'''(f_1 - f_{10}) + \ldots \tag{6}$$

Where f_{20} – the value of function f_2 and $f_1 = f_{10}$; f_{20}', f_{20}'', f_{20}''' – the values of the first three derivations when $f_1 = f_{10}$.

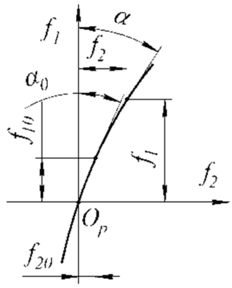

Fig. 1. The scheme of original profile

The values of derivations will equal

$$f_{20}' = \frac{\varsigma_o}{\sqrt{1-\varsigma_o^2}}, \quad f_{20}'' = \frac{\varsigma_o'}{\left(1-\varsigma_o^2\right)^{3/2}}. \tag{7}$$

$$f_{20}''' = \frac{\varsigma_o''\left(1-\varsigma_o^2\right) + 3\varsigma_o\left(\varsigma_o'\right)^2}{\left(1-\varsigma_o^2\right)^{2,5}}$$

According to equalities (3) and (4) it follows that

$$\frac{\left(\varsigma - f_1\varsigma'\right)^2}{\varsigma^{3+\beta}} = \chi_o \tag{8}$$

From here

$$\varsigma_o' = \frac{\varsigma_o - \sqrt{\chi_o\varsigma_o^{3+\beta}}}{f_{10}}, \quad \varsigma_o'' = -\frac{(3+\beta)\varsigma_o'\sqrt{\chi_o\varsigma_o^{1+\beta}}}{2f_{10}} \tag{9}$$

If we determine f_2 from (6) it is necessary to assign ς_o in proportions (5), $f_1 = f_{10}$. Then the equation of curve will be determined when the given value of $\overline{h}_u$ and t_y from (6). The original profile is drawn a line around by the equation of curve.

The equalities of teeth surfaces of engaged wheels, which connected with them in the coordinates $X_1O_1Y_1$ and $X_2O_2Y_2$, will be [4]

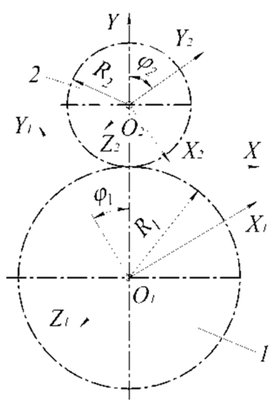

Fig. 2. The scheme of toothing

– the equation of the surfaces for toothed wheel 1

$$x_1 = (f_1 + R_1)\cos\varphi_1 + \Omega_1 \sin\varphi_1,$$
$$y_1 = (f_1 + R_1)\sin\varphi_1 - \Omega_1 \cos\varphi_1, \qquad (10)$$

– the equation of the surfaces for toothed wheel 2

$$x_2 = (f_1 - R_2)\cos\varphi_2 - \Omega_1 \sin\varphi_2,$$
$$y_2 = -(f_1 + R_2)\sin\varphi_2 - \Omega_1 \cos\varphi_2, \qquad (11)$$

The designations are used in the equations (9) and (11) such as

R_1, R_2 – radii of the nominal pitch circle for wheels 1 and 2 correspondently;

φ_1, φ_2 – angular turn of wheels 1 and 2 correspondently.

$$\Omega_1 = \frac{f_1\sqrt{1-\varsigma^2}}{\varsigma}, \quad \varphi_1 = \frac{1}{R_1}(\Omega_1 + f_2), \quad \varphi_2 = \frac{\varphi_1}{u}, \quad u = \frac{R_2}{R_1}$$

The boundaries of the field of toothing will be determined by proportions

$$R_{a1} = \sqrt{(f_{1a1} + R_1)^2 + \Omega_{1a1}^2},$$
$$R_{a2} = \sqrt{(f_{1a2} - R_2)^2 + \Omega_{1a2}^2} \qquad (12)$$

Where R_{a1}, R_{a2} – radii of addendum circles 1 and 2 correspondently; f_{1a1}, f_{1a2}, Ω_{1a1}, Ω_{1a2} – values corresponding to R_{a1} and R_{a2}.

Geometry-kinetic criterions of such gearing determine the complex criterions of capacity for work [4]. They will equal:

– the sliding velocity if $\omega_1 = 1\frac{1}{c}$ (ω_1 - the angular velocity of the toothed wheel 1)

$$V^{12} = \left(\sqrt{c^{1+\beta}} + \sqrt{\chi_o f_1^{1+\beta}}\right)^{\frac{2}{1+\beta}} \qquad (13)$$

– the circular velocity of teeth surfaces $\omega_1 = \frac{1}{c}$; $\omega_2 = \frac{1}{c}$ (ω_2 the angular velocity of the wheel 2 is $\omega_2 = \frac{\omega_2}{u}$)

$$V_1 = \frac{R_1\varsigma^3 + f_1(\varsigma - f_1\varsigma')}{\varsigma(\varsigma - f_1\varsigma')}, \quad V_2 = \frac{R_2\varsigma^3 - f_1(\varsigma - f_1\varsigma')}{\varsigma(\varsigma - f_1\varsigma')}; \qquad (14)$$

– the total circular velocity of the working teeth surfaces

$$V_\Sigma = \frac{2R_1\varsigma^3 + f_1(\varsigma - f_1\varsigma')\left(1 - \frac{1}{u}\right)}{\varsigma(\varsigma - f_1\varsigma')} \qquad (15)$$

– the reduced curvature of working teeth surfaces

$$\chi = \frac{(R_1 + R_2)(\varsigma - f_1\varsigma')^2}{\varsigma^3\left[R_1 + \frac{f_1(\varsigma - f_1\varsigma')}{\varsigma^3}\right]\left[R_2 - \frac{f_1(\varsigma - f_1\varsigma')}{\varsigma^3}\right]} \qquad (16)$$

– the specific sliding

$$\eta_i = \pm\frac{u+1}{u}\cdot\frac{f_1(\varsigma - f_1\varsigma')}{\left[R_i \pm \frac{f_1(\varsigma - f_1\varsigma')}{\varsigma^3}\right]\varsigma^3} \tag{17}$$

where the upper sign is $i = 1$ – for the wheel 1, and lower sign $i = 2$ – for the wheel 2.

It is necessary to continue for involute gearing in proportions (13) … (17) $\varsigma = \sin\alpha_e$, $(\alpha_e = const)$, $\varsigma' = 0$.

According to the qualities (3) and (4) we'll have

$$\varsigma - f_1\varsigma' = \left(\chi_o \cdot \varsigma^{3+\beta}\right)^{0,5} \tag{18}$$

the value will be used with determining of geometry-kinetic criterions (13) … (17).

The values of complex criterions of working capacity will equal (2).

– the criterion of wear of the working teeth surfaces

$$h_u = \Omega_u q_a f^{ty}\eta_i$$

– the criterion of losses in toothing

$$\Delta P = q_a f V^{12} \tag{19}$$

– the criterion of thickness of oil layer between working teeth surfaces

$$h_o = \Omega_o \frac{V_\Sigma^{0,75}}{\chi^{0,4} q_a^{0,15}} \tag{20}$$

– the temperature criterion of jam in working teeth surfaces

$$K_j = \Omega_j \frac{f q_a V^{12}\chi^{0,25}}{\sqrt{V_1 + \sqrt{V_2}}} \tag{21}$$

– the specific work of frictional forces in toothing

$$dA_{ffi} = q_a f \eta_i \tag{22}$$

The designations are introduced in ratios (19) ... (22): Ω_u, Ω_o, Ω_j – the values, which are not depended on the teeth geometry; q_a – the axial force, which has an effect per unit teeth length (the single axial force).

The values of coefficient of sliding friction are shown in the form [6]

$$f = \Omega_f q_a^{0,1}\left[10 + \lg\frac{HBR_a\chi}{E_{re}}\right]\chi^{0,25}V_\Sigma^{-0,1}\left(V^{1,2}\right)^{-0,35} \tag{23}$$

Where HB – the hardness of the less hard tooth that is in contact (kg/cm); R_a – the roughness of the hardest tooth that is in contact (cm); E_{re} – the reduced module of material elasticity of catching wheels (kg/cm).

The single axial force equals

$$q_a = \frac{q_t}{\sqrt{1-\varsigma^2}},$$

where q_t – circular force, which has an effect per unit teeth length.

Conclusions.

1. The recommendations for determination of geometrical parameters of spur gears profile with increased wear resistance were worked out.

2. The recommendations for determination of working capacity criterions for spur gears with increased wear resistance were considered.

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ЗУБЧАСТІ ПЕРЕДАЧІ ЗІ ЗБІЛЬШЕНОЮ ЗНОСОСТІЙКІСТЮ ЗУБЦІВ

Найважливішими критеріями ефективності зубчастих передач є зносостійкість зубців. Проте знос зубців не може бути виключений в умовах граничного тертя, навіть якщо вони працюють у закритій масляній ванні. Ось чому, при вирішенні задачі оцінювання зносу зубців замкнутих передач та розвитку геометрії зубців зубчастих передач з підвищеною зносостійкістю, важливо звернути увагу, на характеристики зменшення зносу робочих поверхонь зубів. Метою статті є визначення функціональної кореляції між геометричними параметрами базової стійки для зубчастих передач та критеріями зносу. Розглянуто порядок синтезу стрічкового інструменту для виробництва зубчастих передач з підвищеною зносостійкістю зубців. Він заснований на моделі зносу зубів замкнутої передачі, яка працює в масляному резервуарі. Ми можемо помітити, що можна знизити величину зносу зубців, якщо зменшити коефіцієнт тертя ковзання та питомого ковзання. Згідно із завданням, коли ми визначимо геометрію робочих поверхонь зубців зі зниженим значенням зносу порівняно з евольвентними зубцями. Отримано диференційне рівняння розв'язок якого визначає геометричні параметри оригінального профілю. Параметри залежать від заданої величини порівняльного зносу синтизованих зубців зубчастих передач двигунів. Геометрично-кінетичні критерії такої передачі визначають складні критерії працездатності: швидкість ковзання, кругова швидкість поверхонь зубців, загальна кругова швидкість робочих поверхонь зубців, зменшена кривизна робочих поверхонь зубців, питоме ковзання. Рішення для виявлення використовується для створення рівняння робочих поверхонь зубців: критерій зносу робочих зубців, критерій втрат між зубцями, критерій товщини шару оливи між робочими поверхнями зубців, температурний критерій заїдання у робочих поверхнях зубів, питома робота сил тертя у зубцях. Також представлені значення коефіцієнта тертя ковзання. Розроблено рекомендації щодо визначення геометричних параметрів профілю зубців зубчастих коліс з підвищеною зносостійкістю. Розглянуто рекомендації щодо визначення критеріїв працездатності зубців зубчастих коліс з підвищеною зносостійкістю.

Ключові слова: зубчасті передачі, зубці, знос зубців, геометричні параметри, підвищена зносостійкість.

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