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GEARING WITH INCREASED TEETH WEAR

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Summary. The most important criterions of gearing efficiency is teeth wear. The order of synthesis of original profile geometry of cutting instrument for cutting teeth with increased teeth wear is set out in this report. It is based on the model of teeth wear of closed gearing that work in oil reservoir. The differential equation is received; its solution determines geometrical parameters of original profile. The parameters depend on set point of comparative teeth wear of synthesizing gearing and evolvent gearing with toothing. 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.

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. 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, characterizing of reduced wear of working teeth surfaces.

OBJECTS AND PROBLEMS

The purpose of our article is the definition of functional correlation between geometrical parameters of original loop of cylindrical straight gear and wear criterion.

The value of teeth wear of cylindrical wheels will be the meaning [4,6]:

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

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 evolvent teeth. According to (1) the value wear of evolvent teeth will equal to:

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

Where: f_e – the meaning of coefficient of sliding friction in evolvent teeth,

η_e – the specific sliding in toothing of evolvent teeth.

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

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

where $\overline{x}$ – relative reduced curvature of working surfaces of gearing with increased unknown gearing in comparing with evolvent 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 evolvent 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 evolvent gearing ($\varsigma = \varsigma(f_1)$):

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

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

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

$$\beta = \frac{-0{,}3t_y + 1{,}5}{0{,}3t_y + 0{,}5},$$

$$\alpha_1 = 0{,}3t_y + 0{,}5,$$

$$\varsigma_o = \sin\alpha_o,$$

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 (axle $O_p f_2$ (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 meaning of function f_2 and $f_1 = f_{10}$;

f_{20}', f_{20}'', f_{20}''' – the meanings of the first three derivations when $f_1 = f_{10}$.

The meanings of derivations will equal

$$f_{20}' = \frac{\varsigma_o}{\sqrt{1 - \varsigma_o^2}},$$

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

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

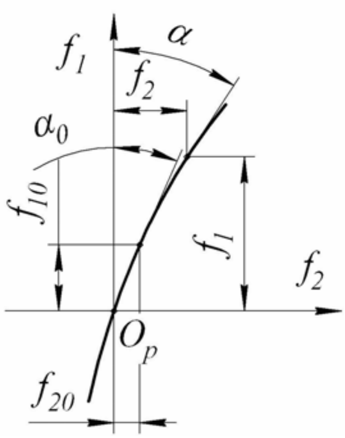

Fig. 1. The scheme of original profile

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

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

From here:

$$\varsigma_o' = \frac{\varsigma_o - \sqrt{\chi_o \varsigma_o^{3+\beta}}}{f_{10}},$$

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

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 catching wheels, which connected with them in the co-ordinates $X_1O_1Y_1$ and $X_2O_2Y_2$, will be [4].

– the equation of the surfaces of teeth wheel 1:

$$x_1 = (f_1 + R_1)\cos\varphi_1 + \Omega_1 \sin\varphi_1,$$

$$y_1 = (f_1 + R_1)\sin\varphi_1 - \Omega_1 \cos\varphi_1, \tag{10}$$

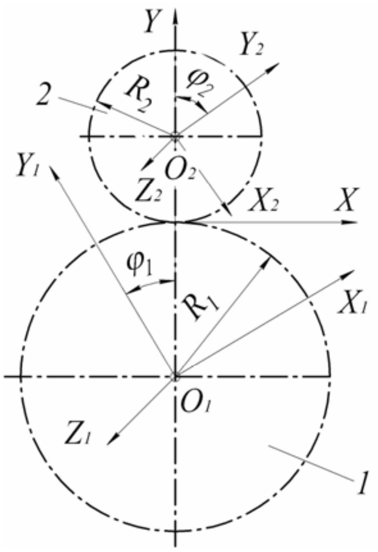

Fig. 2. The scheme of catching wheels

– the equation of the surfaces of teeth wheel 2:

$$x_2 = (f_1 - R_2)\cos\varphi_2 - \Omega_1 \sin\varphi_2,$$

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

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

R_1, R_2 – initial circle radius of wheels,

φ_1, φ_2 – angular displacement of wheels.

$$\Omega_1 = \frac{f_1\sqrt{1-\varsigma^2}}{\varsigma},$$

$$\varphi_1 = \frac{1}{R_1}(\Omega_1 + f_2),$$

$$\varphi_2 = \frac{\varphi_1}{u},$$

$$u = \frac{R_2}{R_1}.$$

The boundaries of the field of catching teeth wheels will be determined by proportions

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

where: R_{a1}, R_{a2} – radiuses of apical teeth of the wheel;

f_{1a1}, f_{1a2}, Ω_{1a1}, Ω_{1a2} – the meanings of values, which are corresponded to apical teeth of the wheels.

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

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

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

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

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

– the total velocity of roll of the working teeth surfaces:

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

– the reduced curvature of working teeth surfaces:

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

– the specific sliding:

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

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

The values of complex criterions of capacity for work will equal (2):

– the criterion of wear of the working teeth surfaces:

$$h_u = \Omega_u q_a f^{t_y} \eta_i,$$

– 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 meanings of coefficient of sliding friction are seen 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 most hard 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.

CONCLUSION

1. The recommendations on definition of geometrical parameters of ordinal profile of cylindrical gears with increased wear were worked out.

2. There were the recommendations on definition of geometrical parameters and criterions of capacity for work of cylindrical gears with increased wear with use of geometrical parameters of ordinal profile.

REFERENCES

1. Kindatskiy B, Sulim G.// Modern condition and problems of multicriterion synthesis of engineering construction. - Lvov, Engineering industry, 2002, N10(64). – p. 26 – 40.
2. Korostylyov L. V. Kinematic activities of bering ability of spaial gears. Publisher: . Engenireing industry, 1964. – № 10. – p. 5 – 15.
3. Kudryavtsev B.N. The componenets of machines- L: Engineering industry. Leningrad department, 1980. – p.464.
4. Shishov V.P., Nosko P.L., Revyakina O.A. Cylindrical gears with arched teeth . Monograph. Lugansk. Publisher EUNU V.Dahl, 2004. – p. 336.
5. Shishov V.P., Nosko P.L., Fil P. V. The theoretical foundations of synthesis in toothing gears. – Lugansk. Publisher EUNU V.Dahl, 2006. – p. 408 .
6. Reference book in two books. The editors are I.V.Kragel`skiy and V.V.Alisina. Book 2, M:Engineering industry, 1979. – p. 358.

ЗУБЧАТЫЕ ПЕРЕДАЧИ С ПОВЫШЕННОЙ ИЗНОСОСТОЙКОСТЬЮ ЗУБЬЕВ

Носко П., Шишов В., Ткач П., Скляр Ю.

Аннотация. Одним из важных критериев работоспособности зубчатых передач является износостойкость их зубьев. Это в большей мере относится к открытым зубчатым передачам, где наблюдается повышенный износ зубьев. Однако в условиях граничного трения не исключается износ зубьев, работающих с наличием закрытой масляной ванны. Поэтому заслуживает внимание решение задачи по оценке износа зубьев закрытых передач и по разработке геометрии зубьев зубчатых передач с повышенной износостойкостью, характеризуемой уменьшенным износом рабочих поверхностей зубьев.

Ключевые слова. зубчатый механизм, зуб, зубчатый износ, геометрические параметры, повышенный