Substantiation of Basic Parameters of Gear Teeth of Open Gears on Wear Resistance

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Abstract: In the article the method of calculation of speed of wear of open gear transmission by protrusions of roughness which are on surfaces of friction of teeth having rounded forms, in the presence of slipping between teeth of gears and pure rolling occurring in a zone of contact of initial circles of meshing gears without participation of abrasive particles, taking into account bending stress arising at transmission of circumferential force is resulted. As a result of the formation on the friction surfaces of the teeth of the teeth of the open gear transmission of the equilibrium roughness is accompanied by an increase in the actual contact area of the teeth. Therefore, the friction surfaces of gear teeth can operate without seizure at higher loads. Roughness of gear teeth surfaces, formed as a result of mechanical processing, under friction under the influence of circumferential force in meshing accompanied by rolling and slipping are subjected to plastic deformation.

1 INTRODUCTION

For open gears the most characteristic type of loading leading to tooth breakage is the bending stress arising from the circumferential force in the meshing. The radial component of the circumferential force deforming the roughness of the friction surfaces of gear teeth leads to the formation of equilibrium (operational) roughness, differing from the original (technological) form (Myshkin et al., 2007; Dubovik, 2015; Ishmuratov, 2019; Mamasalieva, 2024). Profilograms obtained from the friction surfaces of gear teeth after the formation of equilibrium roughness show that their protrusions and hollows of irregularities have relatively close dimensions in height and has a sufficiently large radius of volumetric curvature than at the technological roughness (Mamasalieva, 2024). As a result of the formation of equilibrium roughness on the friction surfaces of the teeth of the open gear transmission, the actual contact area of the teeth is increased. Therefore, the friction surfaces of gear teeth can operate without seizure at higher loads. According to the theory of fatigue wear, wear products from the friction surfaces of gear teeth are separated after a small number of repeated deformation protrusions of roughness having a rounded shape. As a result, the process of wear of gear teeth occurs in the presence between the teeth of rolling with slippage, occurring on the head and foot of the teeth, and wear of teeth in the pure rolling occurring in the contact zone of the initial circles. For these types of contact of gear teeth the rate of wear is determined.

2 MATERIALS AND METHODS

Taking into account the research results given in (Gorlenko et al., 2014; Ishmuratov, 2019) and in order to simplify the calculation of wear rate, the protrusions of roughnesses of tooth friction surfaces, participating in the process of wear are modeled in the form of spheres (Fig. 1) with the volume radius equal to (r).

At contact, in the process of wear of gear teeth, roughness protrusions are embedded to the rubbing surfaces, and under the influence of the radial component of the circumferential force transmitted by the gear mesh, these roughnesses can be partially deformed. When slippage occurs between the gear teeth, the embedded roughness protrusions plow a

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deepened path on the friction surfaces of the teeth (Myshkin et al., 2007).

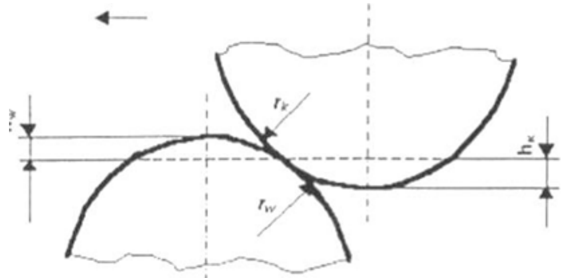

Figure 1: Schematic of contact of roughness protrusions during friction.

In the process of force interaction of meshing gear teeth, the volume of deformed material depends on the depth of embedding, the radius of volume curvature of roughness protrusions, the path of relative slippage of teeth, the number of roughness protrusions located along the length of the width of the contact area and which participate in the process of deformation of friction surfaces (Myshkin et al., 2007).

3 RESULTS AND DISCUSSION

Wear resistance, in the presence of slippage between gear teeth. According to the results of the study obtained in the works (Myshkin et al., 2007), (Ishmuratov, 2019), (Gorlenko et al., 2014), (Ishmuratov et al., 2023), (Irgashev, 2005), between the diameter of the contact spot introduced to the friction surfaces of the teeth roughness protrusions and the circumferential force (P) acting on these protrusions, at the contact of friction surfaces of gear teeth there is a relationship:

$$P = \frac{\pi \cdot a^2_{ш,к} \cdot c \cdot M_{об} \cdot H_{ш,к}}{4} \text{, Н} \qquad (1)$$

where аш,к - diameter of the contact spot of the introduced roughness protrusion to the friction surfaces of teeth, m; c - coefficient depending on the shape of protrusions and on the hardening of the material (Ishmuratov 2019); Моб - the total number of roughness protrusions located on the working area of contact of teeth; - hardness of the material of the driving (driven) gear, MPa.

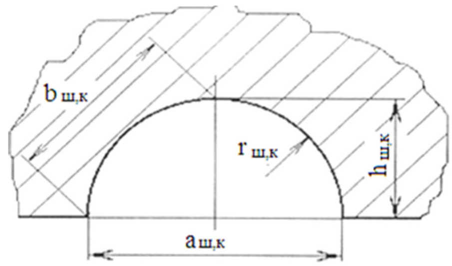

Figure 2: Scheme for determination of geometrical parameters of roughness protrusion introduction into the friction surface of gear teeth.

To calculate the tooth wear rate, it was conventionally assumed that the dimensions of the roughness protrusions in terms of height and volumetric radius of curvature are the same. Which are located sequentially along the length and height of the tooth. During friction, the roughness protrusions are partially embedded in the friction surfaces of the contacted tooth surfaces. According to the accepted conditions of arrangement of protrusions of roughness of friction surfaces, their number of located along the length of gear teeth L, is equal to:

$$M = \frac{L}{a_{ш,к}} .$$

Then the load carried by a single roughness protrusion:

$$\frac{P \cdot a_{ш,к}}{L} = \frac{\pi \cdot a^2_{ш,к} \cdot c \cdot H_{ш,к}}{4},$$

We assume that the circumferential force transmitted by the open gear working in dry friction are perceived by the protrusions of irregularities located on the contact surface of the teeth, their strength depends on the bending stress arising at the foot of the teeth, then the circumferential force perceived by all protrusions of irregularities located on the contact area of the teeth is equal:

$$P = 2 \cdot k \cdot m \cdot \sigma_{изш,к} \cdot M \cdot a_{ш,к} \qquad (2)$$

where is the maximum bending stress occurring on the plane of the circle of the troughs located on the tooth leg of the driving (driven) gear, MPa; is the coefficient of tooth height relative to the dividing circle of the gears; m is the meshing module, m.

For approximate calculations, equating expressions (1) and (2) and solving them with respect to the hardness of the material is obtained:

$$H_{ш,к} = \frac{8 \cdot k \cdot m \cdot \sigma_{изш,к}}{\pi \cdot a_{ш,к} \cdot c} .$$

From the obtained expression it is possible to obtain the ratio of material hardness and bending stress in the gear mesh, the optimal value of which is approximated as Нш,к = 1.88.σ из ш,к..

Due to the fact that the contact patches of the embedded roughness protrusion has an approximate circular shape then the contact patches of the friction surface of the teeth of the driving (driven) gears, is calculated by the expression (Irgashev, 2005),

$$a_{ш,к} = \frac{1{,}27 \cdot P}{L \cdot c \cdot H_{ш,к}} = \frac{2{,}54 \cdot m \cdot k \cdot \sigma_{изш,к}}{c \cdot H_{ш,к}} \text{, м.} \quad (3)$$

According to the scheme presented in Fig. 2 the depth of introduction of roughness protrusions to the friction surfaces at the contact of gear teeth is determined and the dependence is obtained,

$$h_{ш,к} = \frac{0{,}0019 \cdot a_{шк}}{c \cdot H_{ш,к}\theta_{ш,к}} \text{, м} \quad (4)$$

There is a relationship between the volume radius of curvature rш,к and the depth of introduction of the roughness protrusion of the gear teeth hш,к (Ishmuratov and Irgashev, 2020):

$$r = \frac{h_{ш,к}}{3 \cdot c \cdot H_{ш,к}\theta_{ш,к}} \text{, м} \quad (5)$$

The length of the chord of the segment (bш,к), formed as a result of embedding a rounded roughness protrusion, is determined according to the scheme shown in Fig. 2, in which the calculated values of the contact spot diameter aш,к , embedding depth hш,к and volume radius of curvature (rш,к) of roughness protrusions are taken into account,

$$b_{w,k} = \sqrt{2 \cdot h_{ш,к} \cdot r_{ш,к}} = \frac{0{,}816 \cdot h_{ш,к}}{(c \cdot H_{ш,к} \cdot \theta_{ш,к})^{0,5}} \text{, м.} \quad (6)$$

We calculate the cross-sectional area of the deformed metal volume as a result of the introduction of a single roughness protrusion located on the friction surface of the teeth of the driving (driven) gear (Ishmuradov et al., 2024):

$$F_{ш,к} = \frac{h_{шк} \cdot (6a_{ш,к} + 8b_{ш,к})}{15} = \frac{2{,}448 \cdot k \cdot h_{ш,к} \cdot m \cdot \sigma_{из}}{c \cdot H_{ш,к}} \text{, m}^2. \quad (7)$$

The number of roughnesses located on the tooth length of the driving (driven) gear (Irgashev, 2005) taking into account the expression (3), we obtain:

$$M = \frac{0{,}39 \cdot L \cdot c \cdot H_{ш,к}}{k \cdot m \cdot \sigma_{из}} \quad (8)$$

Considering the value of the cross-sectional area of the deformed volume of metal la as a result of introduction of one roughness protrusion from (7), paths of relative slip of roughness protrusions located on the surface of contacted teeth (Ishmuradov et al., 2024) and the number of roughnesses located on the tooth length of gears M calculated by the expression (8) after some simplification we obtain the expression for calculation of the volume of deformation of the material of the driving (driven) gear, by all protrusions of roughnesses located on the contact surface, in the presence of slip between the teeth of gears is equal to:

$$v_{ш,к} = F_{ш,к} s_{ш,к} M_{ш,к} = \frac{3}{z_{ш,к}} \cdot h_{ш,к} \cdot L \cdot m \cdot (i+1) \cdot \psi \text{, m}^3 \quad (9)$$

Here, the slip coefficient of the contacted gear teeth is denoted by , which is defined through the reduced number of teeth,

$$\psi = \sqrt{z_{np}^2 \sin^2 \alpha + 4k_ш \pm 4k^2} - z_{np} \sin \alpha \text{ ,}$$

where is the reduced number of gear teeth, ; here zш , zк are the numbers of teeth of the driving and driven gears, respectively.

In the expression, the plus sign in front of is placed when the tooth head slip is calculated, the minus sign is used to calculate the tooth foot slip.

In the process of friction roughness protrusions of teeth of one gear for each cycle of loading contact with different protrusions of roughness of another gear. And the roughness of the teeth of these gears differ from each other in density of arrangement on the tooth surface and in size. In addition, during contact, the roughness protrusions themselves may partially deform and change their original shape. The same roughness protrusion and deformed counter body surface can meet each other after a certain number of loading cycles of gears, which is taken as the probability of re-deformation, to calculate the value, its obtained dependence (Ishmuradov et al., 2024):

$$\eta_{w,k} = \frac{1}{z_{ш,к} \cdot M} = \frac{k \cdot m \cdot \sigma_{из}}{0{,}39 \cdot z_{ш,к} \cdot L \cdot c \cdot H_{ш,к}} = \frac{2{,}56 \cdot k \cdot m \cdot \sigma_{из}}{z_{ш,к} \cdot L \cdot c \cdot H_{ш,к}} \quad (10)$$

Failure of deformed surfaces of gear teeth occurs according to the fatigue theory of wear, after a certain number of repeated loadings.

The number of loading cycles leading to the destruction of the deformed tooth surface of the driving (driven) gear is equal to:

$$n_{p(ш,к)} = \psi_{ш,к}^{t} \quad (11)$$

where is the coefficient of relative elongation of the leading (driven) gear material; t is the coefficient of frictional fatigue of the gear material, for gears made of steel, t=1.3.

Then, in general, the wear rate of gear teeth of open gears, in the presence of slippage between gear teeth is determined by:

$$\gamma_{\partial(ш,к)} = \frac{v_{ш,к} \cdot n_{ш,к} \cdot \eta_{ш,к}}{F \cdot n_{p(ш,к)}} \text{ , m/h} \qquad (12)$$

where пш,к is the speed of rotation of the driving (driven) gear; F is the area of contact of meshing pairs of gear teeth F=BL.

Widths of contact of gear teeth of the zone of initial circles is calculated by the expression (Ishmuradov et al., 2024):

$$B = \frac{3{,}04 \cdot \sqrt{P \cdot \rho_{пр}} \cdot (1-\mu^2)}{\sqrt{L \cdot E_{пр}}} \text{ , м,} \qquad (13)$$

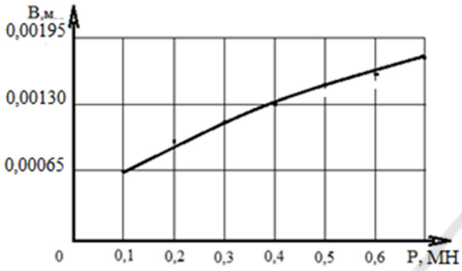

Figure 3: Variation of tooth contact width from circumferential force in a gear mesh.

where μ is the Poisson's ratio; is the reduced radius of curvature of the rolling zone gear teeth.

The graph of change in the contact width of the teeth, depending on the circumferential force presented in Fig. 3 is obtained by expression (12) with the following initial data: =9,78; =0,03; Епр = 215000 MPa shows that increasing the contact width of the gear teeth leads to an increase in the circumferential force transmitted by the gear.

Substituting the values from expression (9) , η ,wk from (10) and (11) into (12) taking into account, and after some simplification, we obtain an expression for calculating the wear rate of the driving (driven) gear, in the presence of slippage between the teeth,

$$\gamma_{\partial(w,k)} = \frac{910{,}8 \cdot h_{ш,к} \cdot m \cdot (i+1) \cdot \psi \cdot n_{ш,к} \cdot k \cdot \sigma_{из} \cdot P^{0,5}}{E_{пр}^{0,5} \cdot \rho_{пр} \cdot L \cdot (1-\mu^2) \cdot z_{ш,к}^2 \cdot c \cdot n_{рш,к} \cdot H_{ш,к}} \text{,m/h} \qquad (14)$$

In Table 1 as an example of the results of calculation of the amount of wear of the teeth of the pinion gears of the driving gear depending on the meshing module in the presence of slippage between the teeth of the gears ψ = 1.035, the coefficient of tooth height from the initial circumference of the driving gear k = 1; the speed of rotation of the driving gear nш = 2.92 r / s; maximum bending stress at the foot of the teeth σиз = 153.7 MPa; reduced modulus of elasticity Епр =215000MPa; gear ratio i=0,125 (gear accelerating); number of teeth of the driving pinion zш =88; deformation factor c = 3; hardness of the material of the driving pinion Нш =282 MPa; the number of deformation cycles leading to the destruction of the deformed surface of the driving pinion at =6% is nрш = 10,273. The results of calculating the resource of the driving pinion are shown in the table. For calculation it is accepted that according to the recommendations proposed in the "Encyclopedia of Mechanical Engineering XXL" limit wear of gear teeth is 20% of the tooth thickness.

Wear resistance of gear teeth, when rolling. In the rolling zone of gear teeth, the process of wear as noted above occurs as a result of deformation of localized volumes of metal friction surfaces. When between the friction surfaces of gear teeth are absent slippage from the introduction of protrusions of roughness in their contact zone formed crater-shaped wells, with wear products are formed after a certain number of repeated deformation of the friction surface of gear teeth protrusions of roughness rounded shape. In this case, the rate of wear of gear teeth in the contact zone of the initial circles, when rolling in general form is determined by the expression:

$$\gamma_{\partial(w,k)} = \frac{v_{1н(w,k)} \cdot M_{об} \cdot n_{w,k} \cdot \eta}{F_{пk} \cdot n_{p(w,k)}} \text{,m/hour} \qquad (15)$$

where Моб is the total number of roughness protrusions located on the contact area of gear teeth.

To calculate the deformed volume of metal of contact surfaces of gear teeth with one roughness protrusion of spherical shape, taking into account the diameter of the contact spot aw,к and hardness of the gear material H ,w,к when the roughness protrusion of the tooth surface has a rounded shape, when rolling the zone of the initial circles of the contacted gear teeth, the dependence [7] is obtained:

$$v_{1H(wk)} = 5{,}75 \cdot \frac{\theta_{ш,к}^2 \cdot k^3 \cdot m^3 \cdot \sigma_{изг}^3}{9 \cdot c \cdot H_{шк}} \text{, m}^3 \qquad (16)$$

In the contact zone of the initial circles of the meshing gears - the value of the gear tooth height coefficient in the zone of the initial circle k can be represented by the ratio of the tooth contact width to the meshing modulus,

$$v_{1H(wk)} = 0{,}639 \cdot \frac{\theta_{ш,к}^2 \cdot B^3 \cdot \sigma_{изг}^3}{c \cdot H_{шк}} \text{ ,}$$

then expression (16) has the form,

The contact areas of the friction surfaces of the rolling zone of the gear teeth are equal:

$$F_{пк} = L \cdot B \text{ , m}^2 \qquad (17)$$

The amount of deformation of friction surfaces depends on the number of roughness protrusions . To calculate the number of roughness protrusions located along the contact width of gear teeth, the dependence is obtained:

$$M_b = \frac{1{,}69 \cdot \sqrt{\rho_{шк}} \cdot (1-\mu^2) \cdot c \cdot H_{ш,к}}{\sqrt{E \cdot B \cdot \sigma_{из}}} \quad (18)$$

The number of consecutive roughnesses located on the tooth length of the driving (driven) gear is determined by expression (8).

Table 1: Basic parameters of calculation on wear resistance of teeth of the driving pinion of open gear transmission.

Hitching modulus m, m	Shes-tern tooth length L, m	The width of the con-tine tact B, m	Tooth contact area $F_к$, m^2	Hitching modulus m, m	Tooth base area $F_о$, m^2	Bending stress, at the tooth foot σ, MPa	Hitching modulus m, m	Depth of penetration of the roughness protrusion	Radius of curvature of the pinion contact point	Drive gear tooth wear rate m/hour $\gamma_ш$	Permissible wear of gear tooth, m	Resource leading gears, hour
1	2	3	4	5	6	7	8	9	10	11	12	13
0,001	0,025	0,00012	0,00000696	0,014	0,00009106	153,7	0,001	0,00014	0,00167	0,0000000662	0,000314	4743
0,002	0.029	0,00024	0,00001392	0,028	0,00018212	153,7	0,002	0,00028	0,00334	0,0000001622	0,000628	3872
0,004	0,037	0,00048	0,00002784	0,056	0,00036424	153,7	0,004	0,00056	0,00668	0,0000003596	0,001256	3492
0,006	0,045	0,00072	0,00004176	0,084	0,00054636	153,7	0,006	0,00084	0,01002	0,0000005432	0,001884	3468
0,008	0,053	0,00096	0,00005568	0,112	0,00072848	153,7	0,008	0,00112	0,01336	0,0000007100	0,002512	3538
0,010	0,061	0,00120	0,00006960	0,140	0,00091060	153,7	0,010	0,00140	0,01670	0,0000008621	0,003140	3642

According to [8], in the contact zone of the initial circles of the meshing gears, only rolling occurs, without slippage of the teeth. For this case:

radius of curvature of the tooth profile of the drive gear,

$$\rho_ш = 0{,}5 \cdot m \cdot z_ш \cdot \sin\alpha \text{, м;}$$

radius of curvature of the tooth profile of the driven gear,

$$\rho_k = 0{,}5 \cdot m \cdot z_ш \cdot i \cdot \sin\alpha \text{, м.}$$

The total number of roughness protrusions located on the tooth contact area, taking into account expressions (8) and (18) is equal:

$$M_{об} = M_b \cdot M = \frac{0{,}34 \cdot \sqrt{\rho_{шк}} \cdot (1-\mu^2) \cdot \theta \cdot L \cdot c^2 \cdot H^3_{ш,к}}{\sqrt{E \cdot (B \cdot \sigma_{из})^3}} \quad (19)$$

The calculated value of the probability of repeated deformation , by the roughness protrusion of the same deformed surface is determined by the dependence (10) [9]:

Substituting the values of η from (10), from (11), v1н(w,k) from (12), Fпк from (16), Моб(ш,к) , from (19), into (14) finally obtains:

$$\gamma_{\partial(w,k)} = \frac{5950\theta^2_{щк} \cdot B^{5/2} \cdot \sigma_{из}^{5/2} \cdot n_{щк}}{i \cdot L^{1/2} \cdot \psi^t_{щк} \cdot P^{1/2}} \text{,м/ч.} \quad (20)$$

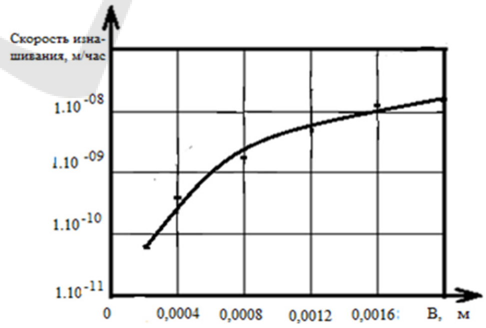

Figure 4: Variation of the wear rate of the driving gear tooth depending on the contact width of the gear teeth.

The resulting expression show that the rate of wear of gear teeth in the contact zone of the initial circles of the initial circles of the meshing gears, rolling zone depends on the tooth length, gear ratio and frictional fatigue of the material, the width of contact of gear teeth, bending stress arising at the foot.

Dependence of the change in wear rate of the rolling zone of the teeth of the driving gear presented in Fig. 4 is obtained from expression 20 at the following initial data: θ = 4,23*10-6 1/MPa; nш = 2,92 rpm/s; p = 0,14 MN; i = 0,125; L = m0,058; σиз = 153,7 MPa; ψш = 6 %; i = 2.

4 CONCLUSIONS

The rate of wear of gear teeth in the presence of slippage between the teeth of the gears increases with increasing modulus of meshing, speed of rotation of the driving (driven) pinion, decreases with increasing number of teeth of the driving (driven) pinion and friction fatigue of the material of the driving (driven) pinion.

In the contact zone of the initial circles of the leading and driven gears to increase the rate of wear leads to an increase in the contact width of the teeth, bending stress and the speed of rotation of the leading (driven) gear, the speed of rotation of the leading (driven) gear, to decrease, increase the gear ratio, tooth length of the gears, friction fatigue and circumferential force transmitted by the gear mesh.

With an increase in the width of contact up to 0.0008 m wear rate of teeth in the rolling zone of the initial circles of the gears involved in the meshing grows more intensively, further increase in the width of contact of teeth up to 0.0020 m leads to an increase in the rate of wear of teeth less intensively.

Established the relationship between the bending stress arising at the foot of the tooth and the hardness of the material gears, to increase the wear resistance of gear teeth is most effective when the ratio of hardness to bending stress arising at the foot of the tooth is 1.88 times.

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