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METHODOLOGIES FOR CALCULATING ABRASIVE PARTICLE SIZE IN GEAR WEAR

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Abstract: The paper considers methodology for calculating the average size of abrasive particles involved in the wear process of gear teeth working in the oil of closed machinery units. Analytical dependences for calculating the friction coefficient between the abrasive particle and the gear tooth surface, the largest and smallest sizes of abrasive particles, the thickness of the oil film between the gear teeth, taking into account the values of elastic deformation of the contact surfaces are proposed.

1. Introduction

The maximum size of abrasive particles involved in the wear process of the gear teeth was determined by taking into account the friction coefficient between the abrasive particle and the tooth surfaces and the concentration of abrasive particles involved in the wear process of drive and idler gears. The higher the coefficient of friction, the greater the probability of larger abrasive particles entering the wedge gap. Under the influence of friction force of abrasive particles penetrating into the wedge-shaped gap of meshing gears' teeth, as they move towards the contact zone of gear teeth surfaces, the value of normal force acting on the abrasive particle, resulting from the circumferential force in the gearing, increases. As a result of that this particle penetrates deeper to friction surfaces, causing elastoplastic deformation of contacted surfaces, leading after some number of repeated loading cycles of gear teeth to their wear [1]. Increase of abrasive particles concentration in the oil of the aggregate, participating in the wear process leads to the proportional growth of wear rate of gear teeth.

The minimum abrasive particle size leading to abrasive wear was determined based on the thickness of the oil film between the gear teeth and the roughness of the running surfaces of the gears.

The purpose of this work is to develop a methodology for calculating the size of abrasive particles involved in the wear process of gear teeth, taking into account the thickness of the oil film, the friction coefficient between abrasive particles and gear tooth surfaces and the concentration of abrasive particles in the oil of the unit involved in the wear process.

2. Theoretical and experimental methods

Materials and calculation procedure. In the area of elastoplastic deformation of friction surfaces, there is a relationship between the elastic modulus of gear materials and the resistance to abrasive particles penetration [2],

$$\frac{2\pi\left(1-\nu^2\right)}{E}=\frac{10^{-6}\pi r_{kp}^2}{4P_{kp}} \quad (1)$$

From the expression for calculating the radius of curvature of the segment, we have:

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$r_{kp} = \sqrt{d_{cp} h_{kp}{}^{2}}$, м, (2)

Substituting the value of $r_{кр}$ from (2) into (1) and solving the expression with respect to $h_{кр}$, we obtain the critical depth of penetration of the abrasive particle into the friction surface:

$h_{kp} = 0{,}5\left(d_{cp} - \sqrt{d_{cp}^{2} - 32 P_{kp}\theta}\right)$, м, (3)

the elastic constant of the gear material is equal:

$\theta = \frac{1 - v^{2}}{E}$, 1/MPa

According to [3] at the elastoplastic deformation boundary, the critical depth of penetration of the abrasive particle into the friction surface is

$h_{kp} = \frac{3 d_{cp} (c\sigma_T \theta)^{2}}{2}$, м, (4)

By equating expressions (3), (4) and solving them with respect to $P_{кр}$, we obtain the resistance force for the abrasive particle to penetrate the friction surface at the elastoplastic deformation boundary:

$P_{kp} = 0{,}19 d_{cp}^{2} (c\sigma_T)^{2} \theta$, Н. (5)

Substituting the values of $P_{кр}$ from (5) into (3), we obtain

$h_{kp} = 0{,}5 d_{cp}\ \ 1 - \sqrt{1 - 6(c\sigma_T \theta)^{2}}$, м. (6)

Thus, the critical depth of penetration of abrasive particles into the friction surface in the elastoplastic deformation zone, depends on the size of the abrasive particles and the yield strength of the gear materials.

According to the diagram in Figure 1, from the geometry of the segment, the path of the abrasive particle at the elastic deformation boundary of the surfaces is equal:

$s_{y} = \sqrt{\rho h_{kp} - h_{kp}^{2}}$, м.

Given that $\rho \gg h_{кр}$, the value of $h_{кр}^{2}$ can be neglected in further calculations. Then by substituting $h_{кр}$ from (6) into (7), we obtain

$s_{y} = \sqrt{0{,}5 \rho d_{cp}\ \ 1 - \sqrt{1 - 6(c\sigma_T \theta)^{2}}}$, м. (7)

At the beginning of contact of abrasive particles with friction surfaces their elastic deformation occurs, as in this case the contact pressure in the zone of contact does not exceed the yield strength of the material of gears. Given this condition, let us determine the number of abrasive particles involved in the elastic deformation of friction surfaces of gear teeth. Conditionally we assume that abrasive particles are evenly distributed throughout the volume of oil in the unit, the thickness of the layer of oil adhering to the friction surface of the gear teeth is equal to the average size of abrasive particles. Then the mass of abrasive particles in the oil adhering to the friction surface of the gears in one revolution of the gear:

$G_{a} = 2mLd_{cp}\varepsilon_{k}\gamma_{м}$, kg, (8)

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The mass of one egg-shaped abrasive particle is equal:

$$G_{a1} = \frac{\pi d_{cp}^3 \gamma_a k_v}{6} = \frac{d_{cp}^3 \gamma_a k_v}{2}, \quad \text{kg.} \qquad (9)$$

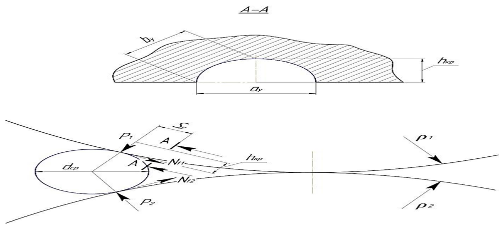

Figure 1. Schematic diagram for calculating the friction resistance coefficient between the abrasive particle and the friction surfaces

The total amount of abrasive particles adhered in one revolution of the gear in abrasive oil is determined from the ratio of expressions (8) to (9):

$$n_{об} = \frac{4mL\varepsilon_k \gamma_м}{d_{cp}^{\ 2} \gamma_a k_v} \qquad (10)$$

According to the assumed condition of abrasive particle distribution over the whole oil volume in the unit, we determine the number of abrasive particles located over the length equal to L| and over the height n_h of the gear tooth. This condition is expressed by the following system of equations:

$$n_1 n_h = n_{об} = \frac{4mL\varepsilon_k \gamma_м}{d^2_{cp} \gamma_a k_v},$$

$$\frac{n_1}{n_h} = \frac{L}{2m}.$$

Solving the system of equations (11) with respect to n_l and n_h , we obtain

$$n_1 = \frac{\sqrt{\frac{\varepsilon_k \gamma_м}{\gamma_a k_v}}}{d_{cp}}; \ (12) \quad n_h = \frac{2m\sqrt{\frac{\varepsilon_k \gamma_м}{\gamma_a k_v}}}{d_{cp}}. \qquad (13)$$

The abrasive particles, being in wedge-shaped clearance of gear teeth, before crushing can, penetrate to friction surfaces on depth 0,1 d_{cp} [3], corresponding to elastic deformation of friction surfaces. Then, the number of abrasive particles on the elastic deformation length is equal to:

The total number of abrasive particles in the elastic deformation zone is determined by expressions (12) and (13),

$$n_y = n_1 n_{sy} = \frac{L\varepsilon_k \sqrt{\rho \gamma_M}\left(1-\sqrt{\left(1-6(c\sigma_T \theta)^2\right)}\right)^{0,5}}{\sqrt{d_{cp}^3 \gamma_a k_v}}.$$

Total volume of elastic deformation of the gear material:

$v_y = v_{1y} n_1$, m^3 ,

the amount of elastic deformation of the gear material, by a single abrasive particle,

$v_{1y} = F_y s_y$, m^3 . (14)

The base area of the deformed gear tooth surface material in a perpendicular section along the path of the abrasive particle in the contact zone is equal:

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$$F_y = \frac{h_{kp} D_y}{15}, \text{ m}^2 \qquad (15)$$

According to the geometry of the segment denote:

$$D_y = 6a_y + 8b_y, \text{ м},$$

The chords of the segment formed as a result of the abrasive particle penetrating the gear tooth surface:

$$a_y = 2\sqrt{h_{kp} d_{cp} - h_{kp}^2}, \text{ м}.$$

After simplifying the expression for calculating D_y we get:

$$D_y = 28{,}2 d_{cp}\sqrt{1-\sqrt{1-6(c\sigma_T\theta)^2}}, \text{ м}. \qquad (16)$$

Substituting $h_{кр}$ from (6), D_v from (16) into (15) we obtain

$$F_y = 5{,}333 d_{cp}^2 \; 1-\sqrt{1-6(c\sigma_T\theta)^2}, \text{ m}^2. \qquad (17)$$

Volume of elastic deformation of a gear tooth by a single abrasive particle:

$$v_{1y} = 5{,}333 k_v \sqrt{d_{cp}^5}\sqrt{\rho}\; 1-\sqrt{1-6(c\sigma_T\theta)^2}^{\;3/2}, \text{ m}^3. \qquad (18)$$

Deformation volume of the elastic contact zone

$$v_y = 5{,}333 k_v \sqrt{d_{cp}^3}\sqrt{\rho} L \sqrt{\frac{\varepsilon_k \gamma_M}{\gamma_a}}\; 1-\sqrt{1-6(c\sigma_T\theta)^2}^{\;3/2}, \text{ m}^3. \qquad (19)$$

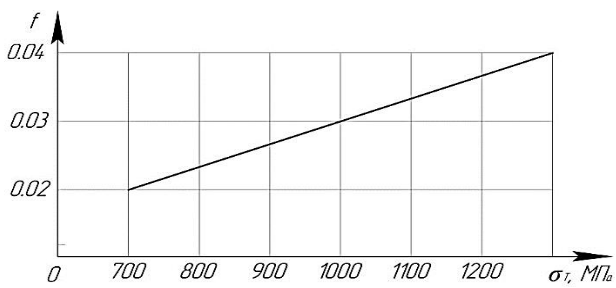
To determine the frictional relationship between the gear teeth and the abrasive particle, calculate the work expended on the elastic deformation of the gear material by the abrasive particles in the wedge-shaped clearance of the gear teeth:

$$A_y = 0{,}84\frac{k_v}{\theta}\sqrt{d_{cp}^3}\sqrt{\rho} L \sqrt{\frac{\varepsilon_k \gamma_M}{\gamma_a}}\; 1-\sqrt{\left(1-6(c\sigma_T\theta)^2\right)}^{\;3/2} \text{ Nm}. \qquad (20)$$

The work expended to overcome the friction force generated between the abrasive particle and the gear teeth in the elastic deformation zone:

$$A_T = 0{,}27 k_v \sqrt{d_{cp}^3}\sqrt{\rho}\theta L f (c\sigma_T)^2 \sqrt{\frac{\varepsilon_k \gamma_M}{\gamma_a}}\; 1-\sqrt{1-6(c\sigma_m\theta)^2}, \text{ Nm}. \qquad (21)$$

Let's suppose that work of elastic deformation is spent for compensation of work of force of friction, made at movement of abrasive particle on surface of gear tooth [4, 5]. By equating expressions (20) and (21) and solving them with respect to friction coefficient f we obtain:



$$f = \frac{3{,}11\sqrt{1-\sqrt{1-6(c\sigma_T\theta)^2}}}{(c\sigma_T\theta)^2}, \qquad (22)$$

Thus, the graph in Fig.2 obtained from expression (22) at θ=0.456* 10^{-6} 1/MPa and c=3, shows that with increasing yield strength of material, friction coefficient between friction surfaces and abrasive particle grows linearly. The abrasive particle penetrates

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into the wedge-shaped gap between the gear teeth when the friction force between the abrasive particle and the friction surfaces is greater than the forces preventing the abrasive particle from penetrating,

$N_f > P$. (23)

Figure 2. Friction coefficient between abrasive particles and friction surfaces of gear teeth, as a function of the yield strength of the material

Estimation of abrasive particle size involved in the wear process of gear teeth. The equal-action force that prevents the abrasive particle from penetrating into the wedge-shaped tooth gap is defined as the geometric sum of the tangential and normal components (fig. 3). These forces act on the abrasive particle through the contact surfaces. The equal-action force which prevents penetration of the abrasive particle tends to displace the abrasive particles from the wedge-shaped tooth gap, and acts against the action of the equal-action friction force (fig.3). The normal force acting on the abrasive particle tends to drive it into the friction surfaces of the gear teeth. The frictional force generated between the abrasive particle and the tooth surfaces pushes it into the wedge-shaped gap, towards the tooth contact zone.

Consider the contact options of an abrasive particle in a wedge-shaped gap with friction surfaces.

1. The abrasive particle has a force interaction with friction surfaces having the same radii of curvature $\rho = \rho_{wk}$ and $(\alpha = \alpha)$.$_{12}$

The equal force of friction between the abrasive particle and the friction surfaces is equal:

$$N_f = N_f^2 + N_{f2}^2 - 2N_{f1}N_{f2}\cos(90^\circ + \alpha_1),\text{H}, \qquad (24)$$

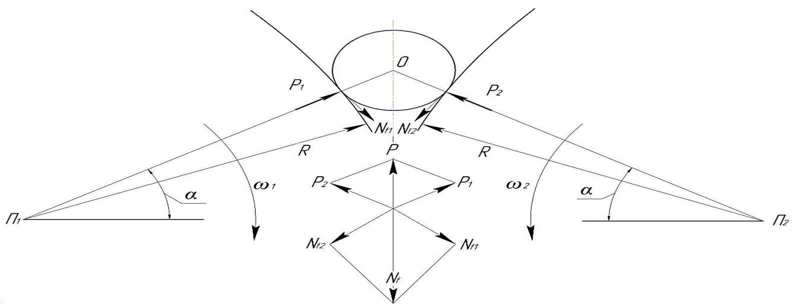

Figure 3. Diagrams for calculating the largest abrasive size

particles penetrating into the wedge-shaped gap of the gear teeth

Assume that the gears are made from different grades of steel with different mechanical properties and surface roughness.

Then, with $f_1 \neq f_2$:

$N_f = P\ f_1{}^2{}_1{}^2 + P\ f_2{}^2{}_2{}^2 + 2P\ P\ f_{1212}\ f \sin\alpha_1$, H,

Dynamic equilibrium of the friction surfaces of the gear teeth and the abrasive particle

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occurs under the conditions P $=P_{12}$ and $N_{f1} = N_{f2}$. Then the total frictional force between the friction surfaces and the abrasive particle

$$N_f = P_1 f_1^2 + f_2^2 + 2 f_1 f_2 \sin\alpha_1 \text{, H.} \quad (25)$$

The total force that prevents the abrasive particle from penetrating the wedge-shaped tooth gap,

$$P = \sqrt{P_1^2 - 2P_1P_2\cos\alpha_1} = P_1\sqrt{2(1-\cos\alpha_1)}\text{, H.} \quad (26)$$

Given expression (15) and taking into account (25) and (26), we have

$$f_1^2 + f_2^2 + 2 f_1 f_2 \sin\alpha_1 = 2(1-\cos\alpha_1). \quad (27)$$

From Δ P_1 O_1 figure 3 the distance from the point of contact of the abrasive particle with the friction surfaces to the point of contact of the friction surfaces is equal:

$$OO_1 = \sqrt{\Pi_1 O_1^2 - \Pi_1 O^2} = \frac{\sqrt{4\rho d_{\max} - d_{\max}{}^2}}{2},$$

By replacing the values:

$$\sin\alpha_1 = \frac{\sqrt{4\rho d_{\max} + d_{\max}{}^2}}{2\rho + d_{\max}};$$

$$\cos\alpha_1 = \frac{2\rho}{2\rho + d_{\max}},$$

from (3.58) given in [6] we obtain

$$\frac{f_1^2 + f_2^2 + f_1 f_2\sqrt{4\rho d_{\max} - d_{\max}{}^2}}{2\rho + d_{\max}} = 2 - \frac{4\rho}{2\rho + d_{\max}}. \quad (28)$$

After appropriate transformations, equality (28) takes the form:

$$d_{\max}{}^2\left(2f_1^2 f_2^2 - 4 + 4f_1^2 + 4f_2^2 - f_1^4 - f_2^4\right) + d_{\max}\left(f_1^2 f_2^2\rho + 8f_1^2\rho + 8f_2^2\rho - 4\rho f_1^4 - 4\rho f_2^4\right) - \\ -\left(4\rho^2 f_1^4 + 8\rho f_1^2 f_2^2 + 4\rho^2 f_2^4\right) = 0$$

(29)

The coefficients of friction between the friction surfaces and the abrasive particles f_1 , f_2 are close in value and in equation (29) can be approximated:

$2f\, f_1{}^2{}_2{}^2 = f_1{}^4 + f_2{}^4$.

Then equation (29) has the form:

$$d_{\max}{}^2\left(f_1^2 + f_2^2 - 1\right) + 8\rho d_{\max}\left(f_1^2 + f_2^2\right) - 4\rho^2\left(f_1^4 + 2f_1^2 f_2^2 + f_2^4\right) = 0 \quad (30)$$

or

$$4d_{\max}{}^2\left(1 - f_1^2 - f_2^2\right) - 8\rho d_{\max}\left(f_1^2 + f_2^2\right) + 4\rho^2\left(f_1^4 + 2f_1^2 f_2^2 + f_2^4\right) = 0.$$

Solving equation (30) with respect to d_{max} , we obtain the largest possible abrasive particle size penetrating the wedge-shaped tooth gap:

$$d_{\max} = \frac{\rho\left(f_1^2 + f_2^2\right)}{1 + f_1^2 + f_2^2}\text{, м.} \quad (31)$$

If the friction surfaces are made of identical materials with the same mechanical properties and roughness, then

$$f = f_1 = f_2,$$

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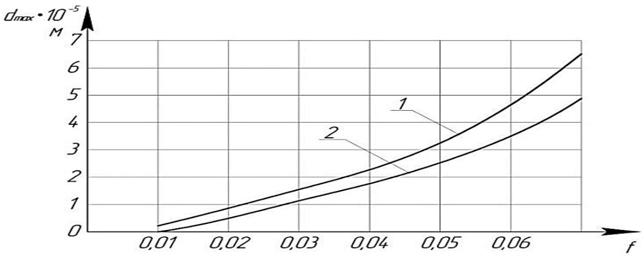
$$d_{\max} = \frac{2\rho f^2}{1+1,41f}, \text{ м.} \qquad (32)$$



The relationships shown in Figure 4 are derived from equation (32). 4 are obtained from equation (32).

Figure 4. Variation of greatest size of abrasive particles penetrating a wedge-shaped gap of equal radius of curvature as a function of friction coefficient between abrasive particles and friction surfaces: k=0.5, a=20°: 1 - t = 0.025 m, z_w =12;2- t = 0.01 m, z =30 $_w$

It is established that with increasing the coefficient of friction between the friction surfaces and the abrasive particle and the meshing module, the largest size of abrasive particles penetrating into the wedge-shaped gap of gear teeth, increases (Fig. 4).This is due, first, that increasing the friction coefficient leads to an increase in the pushing force of abrasive particles into the wedge-shaped gap in the direction of the contact zone of the surfaces. Secondly, an increase in the meshing modulus leads to an increase in the radius of curvature of the tooth profile, which leads to an increase in the maximum size of abrasive particles penetrating the wedge-shaped gap (see expression 32).

2. The radii of curvature of the friction surfaces are not equal, which corresponds to the contact profiles of the teeth of real gears. From the scheme shown in Figure 3, from ΔTFS the equilibrium friction force between the abrasive particle and the gear tooth surfaces, at $\alpha_1 \quad \alpha_2$:

$$N_f = P_1\sqrt{f_1^2 + f_2^2 - 2f_1 f_2 \cos(180°(\alpha_1+\alpha_2))}, \text{ H,} \qquad (33)$$

where α_1 , α_2 are the angles of action of force P on the abrasive particle, deg.

The equal force P that prevents the abrasive particle from penetrating the wedge-shaped gap of the gear teeth is determined from ΔTMR,

$$P = P_1\sqrt{2(1-\cos(\alpha_1+\alpha_2)}, \text{ H.} \qquad (34)$$

From $\Delta O_1 P_1 P_2$

$$\cos(180°(\alpha_1+\alpha_2) = 1 - \frac{8\rho_w\rho_k}{(2\rho_w + d_{\max})(2\rho_k + d_{\max})}, \qquad (35)$$

$$\cos(\alpha_1+\alpha_2) = \frac{8\rho_w\rho_k}{(2\rho_w + d_{\max})(2\rho_k + d_{\max})} - 1. \qquad (36)$$

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Substituting expressions (35) and (36) into (33) and (34), for the case where dry friction occurs between the friction surfaces and the abrasive particle, for the case $N_f = P$, we have

$$\frac{d_{\max c}^{2} + 2d_{\max c}(\rho_w + \rho_k) - 4\rho_w\rho_k(f_1 + f_2)^2}{4 - (f_1 - f_2)^2} = 0, \qquad (37)$$

where d_{maxc} is the largest size of abrasive particles penetrating into the wedge-shaped gap of the gear teeth during dry friction, m.

Solving equation (37) with respect to d_{maxc} , we obtain

$$d_{\max c} = \sqrt{(\rho_w + \rho_k)^2 + 4\rho_w\rho_k(f_1 + f_2)^2} - (\rho_w + \rho_k), \qquad (38)$$

If the friction coefficients between the abrasive particles and the friction surfaces of the gear teeth are equal, expression (38) takes the form:

$$d_{\max c} = \sqrt{(\rho_w + \rho_k)^2 - 4\rho_w\rho_k f^2} - (\rho_w + \rho_k), \qquad (39)$$

The values of ρ_w and $\rho_к$ are calculated from expressions (3.6) and (3.8) in [6].

As the engagement modulus and friction coefficient between the friction surfaces and the abrasive particles increases, the probability of larger abrasive particles entering the wedge-shaped gap between the gear teeth increases.

In the presence of an oil film between the abrasive particles and the friction surfaces of the gear teeth, the greatest size of the abrasive particles penetrating the wedge-shaped gap by the friction surfaces decreases due to the reduced friction coefficient [2]:

$$d_{\max м} = d_{\max c} - 2h_a, \text{ м}, \qquad (40)$$

The coefficient of friction between the abrasive particle and the friction surfaces of the gear teeth in the presence of an oil film is defined by expression (40) and by solving the equation concerning the coefficient of friction in the presence of an oil film $f_{мэп}$ we obtain

$$f_{мэп} = f_{сап}^{2} - \frac{(\rho_w + \rho_k)^2 + \rho_w\rho_k f_{сап}^{2} - h_a^2}{\rho_w\rho_k}; \qquad (41)$$

In dry friction and in the presence of an oil film, the coefficient of friction between the gear tooth surfaces and the abrasive particle increases as the radius of curvature and the yield strength of the gear material increase.

The thickness of the oil film between the abrasive particle and the tooth friction surface, is [7]:

$$h_a = 2{,}4\mu_0 e^{\alpha p} \quad \rho_{w,k} + \frac{d_{\max}}{2} \quad (v_{1,2} + v_a)\frac{a}{P_1}, \text{ м},$$

From the condition of anchoring of abrasive particle to the friction surfaces of gear teeth [3], $v = v_1\,\alpha + v_2\,\beta$, m/s, here α, β are probabilities of anchoring of abrasive particle to friction surfaces of teeth of driving and driven gears [8, 9].

The force acting on a single abrasive particle in the deformation zone of the friction

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surfaces is equal:

$P_1 = ah_{кр}$ HB, N

The diameter of the abrasive particle's contact patch with the friction surface:

$$a = 2r_{kp} = 2 \quad d_{\max} h_{kp} - h_{kp}^{2}, \text{ м.} \qquad (42)$$

Substituting the value of $h_{кр}$ from (4) into (42), we obtain

$$a = 2d_{\max m}\sqrt{1{,}5(c\theta\sigma_T)^2} = 2{,}45 d_{\max m} c\theta\sigma_T, \text{ м.} \qquad (43)$$

Tension between the abrasive particle and the friction surface:

$$p = \frac{4p_1}{\pi a^2} = 0{,}78 c\theta\sigma_T HB, \text{ MPa.} \qquad (44)$$

Then, given expressions (42), (43) and (44), the thickness of the oil film between the friction surfaces and the abrasive particle will be

$$h_a = \frac{4{,}6\mu_0 e^{ap}\rho_{w,k}\left(v_{1,2} + v_a\right)}{d_{\max m} c\theta\sigma_T}, \text{ м.} \qquad (45)$$

Substituting the value of h_a from (45) into (41), we finally obtain the coefficient of friction between the surfaces and the abrasive particle $f_{мэп}$, in the presence of an oil film:

$$f_{мэп} = \sqrt{f_{can}^{2} - \frac{(\rho_w + \rho_k)^2 + \rho_w\rho_k f_{can}^{2}}{\rho_w\rho_k} - \frac{4{,}67\mu_0 e^{ap}\left(v_{1,2} + v_a\right)}{d_{\max m} c\theta\sigma_T}^{2}}\ . \qquad (46)$$

Table 1

Change in the greatest size of abrasive particles penetrating the wedge-shaped clearance between the gear teeth, depending on the modulus of engagement

m, m	z_w	ρ_w , м	ρ_k , м	f_{can}	$f_{мэп}$	d_{maxc}, m	d_{maxm}, m
For pinion head and idler foot							
0,001	300	0,05287	0,10143	0,04	0,0299	0,0000278	0,0000156
0,005	60	0,05892	0,09589	0,04	0,0346	0,0000290	0,0000217
0,010	30	0,06458	0,08972	0,04	0,0378	0,0000301	0,0000268
0,015	20	0,07037	0,08394	0,04	0,0394	0,0000306	0,0000297
0,020	15	0,07850	0,07850	0,04	0,0399	0,0000308	0,0000308
0,025	12	0,08094	0,07336	0,04	0,0398	0,0000308	0,0000305
For pinion stem and idler head							
0,001	300	0,05287	0,10143	0,04	0,0299	0,0000278	0,0000156
0,005	60	0,05822	0,09509	0,04	0,0345	0,0000290	0,0000216
0,010	30	0,06419	0,08011	0,04	0,0376	0,0000300	0,0000266

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0,015	20	0,06956	0,08474	0,04	0,0392	0,0000306	0,0000294
0,020	15	0,07447	0,07984	0,04	0,0399	0,0000308	0,0000307
0,025	12	0,07890	0,07531	0,04	0,0399	0,0000308	0,0000308

By substituting the value of $f_{МЭП}$ in (41), it is possible to calculate the largest abrasive particle size that would penetrate the wedge-shaped gap in the presence of an oil film:

$$d_{\max m} = (\rho_w + \rho_k)^2 + \rho_w \rho_k f_{can}{}^2 - (\rho_w + \rho_k); \text{ м.} \qquad (47)$$

Calculation results produced by expressions (22), (29), (46) and (47) given in Table 1, were performed at: A=0.45 m; i=2; σ_T =1100 MPa; μ_o =0.54*10^{-8} MPa s; α=8.58*10^{-3} 1/MPa; θ= 0.456*10^{-5} 1/MPa; k=0.5.

Thus, in the presence of an oil film between the gear teeth, an increase in the engagement modulus contributes to the penetration of larger abrasive particles into the wedge-shaped gap of the gear teeth. Abrasive particles of at least the thickness of the oil film can participate in the wear process of the gear teeth, i.e.

$$d_{\min} = h_{\min} + s_c, \text{ м,}$$

then the minimum thickness of the oil film between the friction surfaces [9],

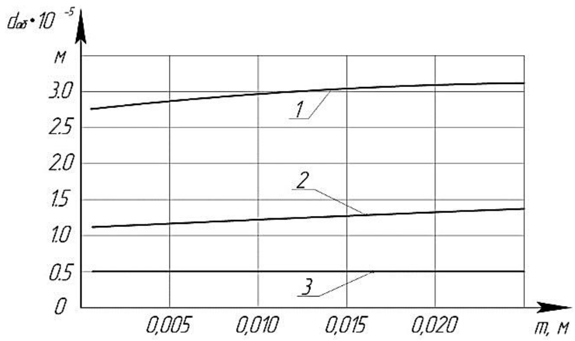
$$h_{\min} = \frac{1{,}12\mu_0 e^{ap} A \sin\alpha (v_1 + v_2) L}{P_{\max}}, \text{ м,}$$

From the equation of elastic deformation of gear teeth of heavily loaded gears [10], by double integration we obtain [7] the amount of elastic deformation of the contact surfaces,

$$s_c = \frac{2{,}36 P_{\max}(1 - v^2)}{\pi L E}, \text{ м.}$$

The logarithm of the average size of the abrasive particles involved in the wear process is

$$\ln d_{cp} = \frac{\ln d_{\max} + \ln d_{\min}}{2},$$



after conversion

$$d_{cp} = \sqrt{d_{\max} \; d_{\min}}, \text{м} \qquad (48)$$

Fig.5. Change in abrasive particle size entering the wedge-shaped tooth gap as a function of engagement modulus: 1 - maximum size; 2 - average size; 3 - minimum size.

Plots of changes of abrasive particles size, penetrating into wedge-shaped clearance of gear teeth, depending on meshing modulus, calculated from expressions (47) and (48) are shown on fig.5.

Conclusion. Thus, the obtained calculated dependences show that the change in gearing modulus does not affect the minimum size of abrasive particles penetrating into the wedge-shaped gap between the gear teeth, as the thickness of the oil film in the contact zone

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of the gear teeth does not depend on the gearing modulus. Increasing the gearing modulus leads to an increase in the maximum and average size of abrasive particles that penetrate into the wedge-shaped clearance of the gears. The maximum size increases at a higher rate.

3. Conclusion

E - elastic modulus of the gear material, MPa; $P_{кр}$ - resistance force to penetration of an abrasive particle into the friction surface on the border of elastoplastic deformation, N; $d_{кр}$ - radius of the penetrated abrasive particle to the friction surfaces on the border of elastoplastic deformation, m; v - Poisson's ratio; h_{kp} -depth of penetration of abrasive particle into the friction surface, at the boundary of elastoplastic deformation, m; d_{cp} - average size of abrasive particles, m; θ-elastic material constant, 1/MPa;s - strain factor; $\sigma_т$ - yield strength of gear material, MPa; m - gear module, m; L - gear tooth length, m; i-transmission ratio of gear transmission; ε_k - concentration of abrasive particles involved in the wear process; $\gamma_м$ - oil density, kg/m ;^{3} k_v - factor that takes into account the abrasive particle shape; γ_a - abrasive particle density, kg/m ;^{3} n_1 - number of abrasive particles located along the gear tooth length; n_h - number of abrasive particles located along the gear tooth width n_{sy} - number of abrasive particles located across the width of the elastic deformation zone of the gear tooth; a_v , b_y - chords of the segment formed as a result of penetration of an abrasive particle into the gear tooth surface, m; N_f - the equivalent friction force between the friction surfaces and the abrasive particle, N; P - the equivalent force preventing the abrasive particle from penetrating into the wedge-shaped gap of the gear teeth, N; N_{f1} , N_{f2} - the friction force between the abrasive particle and the friction surfaces of the gear teeth, N; α_1 - angle between the vectors of forces acting on the friction pair and the normal force acting on the abrasive particle, deg;P_1 , P_2 - normal forces acting on the abrasive particle located in the wedge-shaped gap between the gear teeth, N; ρ - tooth profile curvature radius, m; f_1 and f_2 - friction coefficients between the surfaces and the abrasive particle; h_a - thickness of oil film between the friction surfaces of the gear teeth and the abrasive particle, m; f_{can} - dry friction coefficient; μ_0 - oil viscosity, MPa s; α - piezo viscosity coefficient, 1/MPa; p - stress between abrasive particle and friction surfaces, MPa; $v_{1,2}$ - rolling speed of friction surfaces, m/s; v_a - speed of abrasive particle in wedge gap, m/s; a - contact spot diameter, embedded in friction surface, abrasive particle, m; P_1 - force acting on one abrasive particle, N; HB - material hardness, MPa; s_c - value of elastic deformation; gear teeth in friction, m; A - axial distance of gear, m; P_{Max} - circumferential force transmitted by meshing, N; $d_{\max}$ - maximum abrasive particle size; $d_{\min}$ - minimum abrasive particle size; R_z - roughness height of friction surfaces, m

4. References

1. Ikramov U A, Irgashev A. Calculation of wear of teeth of open toothed gears.//- Theses of international scientific-practical conference "Problematic issues of mechanics and mechanical engineering". Tashkent, 1993. C. 280- 281.
2. Kh Ishmuratov, R K Hamroev, B B Kurbonov, N N Mirzaev. Method for modeling the process of wear of gear teeth. Journal of Physics: Conference Series. 2022, **2176(1)**, 012096

eISSN: 2349-5715 pISSN: 2349-5707

Volume: 11, Issue 03, March-2024 SJIF 2019: 4.702 2020: 4.737 2021: 5.071 2022: 4.919 2023: 6.980 2024: 7,662

3. Kragelsky I V et al. Fundamentals of Calculations of Friction, Wear. - Moscow: Mashinostroenie. 1977. - 526 c.
4. Khachaturian S V, Negmatov S S, Irgashev A. Results of experimental research of entropic balance of abrasive wear process. Vestnik (Herald) of Tashkent State Technical University. - Tashkent 2004, N 3.
5. N N Mirzaev, G Sh Zhuraeva, R K Hamroev. The use of soft wall containers for transporting vegetables. Energy Systems Research 2021, E3S Web of Conferences **289**, 07029 (2021).
6. Irgashev A. Methodological bases of increasing wear resistance of gears of heavy loaded gearings of machine units. - D. in Technical Sciences. - Tashkent, 2005, 244 p.
7. Ishmuratov H K. Theoretical justification of gear life of cotton harvesting machines according to wear criterion. - Ph. in Technical Sciences (PhD). - Tashkent, 2019, 156 p.
8. Ergasheva Z K, Rakhmanova T T, Abhijit Tarawade, Sultonova S A, Safarov J E. Investigation of the drying process of moist materials under convective heat input conditions. // IOP Conf. Series: Earth and Environmental Science (AEGIS-2022). **1076 (2022)** 012058. P.1-6.
9. Zulpanov Sh U, Samandarov D I, Dadayev G T, Sultonova S A, Safarov J E. Research of the influence of mulberry silkworm cocoon structure on drying kinetics. // IOP Conf. Series: Earth and Environmental Science (AEGIS-2022). **1076 (2022)** 012059. P.1-6.