Modeling of Load Carrying Capacity and Durability of Metal-polymer Spur Gears Made of Polyamide PA6 Based Compositions

Based on the author's method of calculation of metal-polymer (MP) gears, the study of load carrying capacity and durability of metal-polymer spur gears with a gear made of unfilled polyamides PA6, PA66, and polyamide PA6 based composites filled with glass (PA6 + 30GF) or carbon (PA6 + 30CF) fibers, molybdenum disulfide (PA6 + MoS2) oil-filled cast polyamide (PA6 + Oil) is performed. The conditions of the teeth engagement and the change of the radiuses of curvature of the involute profile during wear are taken into account. The calculation of the maximum contact pressures during the cycle of tooth interaction - from the entrance to the engagement to the exit from it is performed. The effect of tooth wear on their reduction is also studied. The linear wear of polymer teeth at the selected points of a profile and the characteristic point where the acceptable value is reached (at two-pair or one-pair engagement) is established. The gear's durability is calculated and their comparative estimation is given. Qualitative and quantitative regularities of change of the specified service characteristics of MP gears at the achievement of the acceptable wear of polymeric gear teeth are established.

Keywords: metal-polymer spur gears; polyamide PA6, PA66, and composites PA6+30GF, PA6+30CF, PA6+MoS2, PA6+Oil; calculation method; load-carrying capacity and durability; wear of polymer gear teeth

1. INTRODUCTION

Metal-polymer and polymer gears are becoming increasingly used in many areas. Many thermoplastic polymer materials and composites based on them are used for the manufacture of gears: traditional – polyamides PA, polyacetal POM, highly effective – polyester ketone PEEK, ethylene terephthalate PET and others, which have quite different mechanical and tribological properties (wear resistance, coefficient of friction). To reduce wear and increase the durability of MP gears, the filling of the base polymer (matrix) with particles/fibers of different types and structures (molybdenum disulfide, graphite, glass, and carbon fiber, bronze powder, polyethylene, etc.) with different volume content is used, which minimize the disadvantages of unmodified polymers.

Common polymeric materials for MP gears are polyamides, in particular PA6, with various types of fillers (molybdenum disulfide, glass and carbon fiber, oil). Its wide use is due to high mechanical, operational, and tribological characteristics. These are the following properties:

- good frictional properties at dry sliding friction;
- good, in some of its composites increased wear resistance;
- resistance to oils, fats, diesel and gasoline, alkalis;
- high mechanical strength, rigidity, hardness, and elasticity;
- high damping ability;
- good fatigue strength;
- ease of machining.

Therefore, it would be important and practically necessary to estimate the load-carrying capacity, wear and durability of MP gears made of polyamides PA6, PA66 and composites PA6+30GF, PA6+30CF, PA6+MoS2, PA6+Oil. However, despite the sufficiently wide and growing use of MP gears, effective calculation methods for evaluating these specific performance characteristics in the literature are missing. Only in the old work [1] Based on a simplified method [2], since Archard's law of abrasive wear was used, a partial study of MP spur gears with a gear made of polyamide filled with glass or carbon dispersed fibers was carried out. There are also no studies of this species by various numerical methods.

In [3-5], the results of the contact and bending strength calculation of MP spur gears are available. In [3, 4] the influence of load distribution between gears on the specified stresses in the MP gears with a gear made of polyamide PA66 paired with a steel (S355J2) pinion was studied. And in [5] the estimation of the maximum contact pressures and sliding speeds of the same gears is carried out. In [6], the influence of friction force on bending stresses in spur gears is investigated. The article [7] is devoted to the study of the friction and wear behavior of a reinforced polymer composite.
Regarding experimental studies of the quantitative wear resistance of various polymer composites of metal-polymer gears, there are limited data in the literature. Works [8-10] are devoted to the experimental study of the wear of model MP gears. The teeth volumetric wear in metal-polymer gears made of polyamide PA6-Mg, PA6-Na, PA66 + 30GF, polyacetal POM-C - steel S355 without abrasive and in its presence was studied in [8, 9]. Weight wear of a gear made of PA6 paired with a steel pinion was studied in [10] on different types of metal-polymer gears (spur, bevel, worm).

Instead, the tribological properties of different types of polyamides were studied on a pin-on-disc testing machine. The work [11] is devoted to the wear resistance study of the tribo-couple PA66 - steel SAE 1045 used in the clutch discs. In [12 - 14] the tribological behavior of polyamide PA6 at dry friction under different conditions was studied.

Because of this, an urgent scientific and technical problem is the creation of effective and reliable methods for the estimated assessment of wear and service life of gears made of polymer composite materials. In [15 - 17] the author's method of calculation of MP gears is based on the phenomenological methodology of research of materials wear resistance at sliding friction [16, 19, 21, 24] and frictional - fatigue wear mechanism is given. It is based on methods of calculating different types of metal gears (spur, bevel, worm) [18 - 24]. In this article, the study of contact strength, wear and service life of gears made of polyamides PA6, PA66 and composites PA6 + 30GF, PA6 + 30CF, PA6 + MoS2, PA6 + Oil is performed.

2. METHODS, MATHEMATICAL MODEL OF WEAR

A mathematical model of the tribo-process [16, 19, 21, 24] described by a system of linear differential equations are used to study the tooth wear kinetics in the engagement of the gears at combined rolling and sliding friction:

\[
\frac{1}{v} \frac{dh_i}{dt} = \Phi_i^{-1}(\tau), \quad \frac{1}{v} \frac{dh_2}{dt} = \Phi_2^{-1}(\tau),
\]

where \(v\) – sliding speed; \(h\) – linear wear; \(t\) – wear time; \(\Phi(t)\) – a characteristic function of wear resistance (wear resistance indicator) of materials in the accepted friction pair under the specified friction conditions used as the basic integral parameter of the model; \(\tau = f_p\) - the specific force of friction according to Coulomb's law; \(k = 1; 2\) – numbering of tribosystem elements; \(f\) – sliding friction coefficient; \(p\) – contact pressure determined by the methods of the theory of elasticity.

The developed mathematical model of wear relies on the frictional - fatigue wear mechanism, in which the wear rate functionally depends on the value of Coulomb friction forces arising in tribo-contact under the influence of external load and mutual displacement of tribosystem elements.

Since polymeric materials and composites are based on different strengths the following ratio is used to approximate the experimental values of the wear resistance function \(\Phi_i(\tau)\):

\[
\Phi_k(\tau) = C_k \left( \frac{\tau}{\tau_0} \right)^{m_k},
\]

where \(C_k\), \(m_k\) – wear resistance characteristics of tribo-couple materials for the accepted conditions; \(\tau = \sigma_y / 2\) – shear strength of the polymeric material (according to the Tresca-Saint-Venant hypothesis); \(\sigma_y\)– tensile strength (compression).

The experimental values of the wear resistance indicator (function) \(\Phi_i(\tau)\) based on the results of tribological studies are calculated as follows:

\[
\Phi_i(\tau) = L / h_i,
\]

where \(h_i\) – experimental values of linear wear of material samples; \(L = v_1\) – the path of friction; \(i\) – selected values of contact pressures.

After separating the variables and integrating the system (1) of tribo-kinetic wear equations, the wear time function is obtained

\[
t_k = \frac{C_k \tau_{0}^{m_k}}{\tau} \int_{0}^{\tau_{0}} dh_k, \quad m_k > 0.
\]

Under simplified contact conditions, the pressure \(p\) and, accordingly, the specific friction forces \(\tau\) are considered to be constant during wear of the plastic gear, and then expression (4) takes the form

\[
t_k = \frac{C_k \tau_{0}^{m_k}}{\tau} h_k.
\]

Hence, respectively, linear teeth wear \(h\) is calculated by the formula

\[
h_k = \frac{v_k \tau_{0}^{m_k}}{C_k \tau_{S}^{m_k}}.
\]

Taking into account the operating conditions and the peculiarities of the teeth contact in the process of their interaction, the function of teeth linear wear \(h_j\) at any point of their lateral surface during a single time of tribo-contact \(t_j\) for single-pair engagement is expressed as follows

\[
h_j = \frac{v_j t_j \left( \frac{\tau_{p_{j_{max}}}^{m_j}}{C_k \left( 0.5 \sigma_y \right)^{m_j}} \right)}{\frac{C_k \left( \tau_{S} \right)^{m_j}}{\tau_{j_{max}}^{m_j}}}
\]

where \(v_j = v \) – sliding speed at the \(j\)th point of the teeth lateral surfaces; \(t_j = 2b_j / v_0\) – time of teeth tribo-contact during movement of the \(j\)th point of contact along the tooth contour to the width of the contact area; \(v_0 = \omega_i \sin \alpha\) – the speed of movement of the point of contact along the contour of the tooth; \(\omega_i\) – pinion angular velocity; \(r_i = z_i m\) – the pinion pitch radius; \(m\) – module; \(\alpha = 20^\circ\) – pressure angle; \(z_i\) – the number of teeth of the pinion; \(p_{j_{max}}^{(2)} = p_{j_{max}} / \sqrt{2}\) – for two-pair engagement.

Linear teeth wear during the given gears resource \(t\) under the simplified conditions of contact interaction is expressed as follows:
\[ h_{ij} = 60 n_k h_{ij}'^* \]  

where \( n_k \) – number of revolutions of gears.

The gears durability (resource) at the given acceptable teeth wear \( h_{ij} \) is

\[ t = h_{ij} / h_{ij}'^* = 60 n_k h_{ij}'^* . \]  

(9)

The sliding speed is calculated by the formula

\[ v_j = a_\alpha r_{h_0} (tg \alpha_{a,j} - tg \alpha_{a,j}) , \]

where \( a_\alpha = \arccos[ (r_j / r_k) \cos \alpha] \).

The maximum contact pressures \( p_{j_{\text{max}}} = \sigma_{j_{\text{H}}} \) and the contact area width \( 2b_j \) at the \( j^{th} \) point of the profile are calculated by Hertz formulas

\[ p_{j_{\text{max}}} = 0.564 \sqrt{N' / \rho_j} , \]

\[ 2b_j = 2.256 \sqrt{N' / \rho_j} , \]  

(11)

where \( \sigma_{j_{\text{H}}} \) – contact pressure; \( N' = N / bw \); \( N = T_{\text{nom}} K_r \); \( \cos \alpha \) – force in the engagement; \( T_{\text{nom}} = 9550 \ P \text{m} \) – rated torque; \( \rho_j \) – power on the driving shaft; \( m_j \) – the pinion rotational speed; \( K_r \) – dynamism coefficient; \( \theta = (1-\alpha_j)(E_i + (1-\alpha_j^2)E_2) \); \( E_i \), \( E_2 \) – Young's modulus and Poisson's ratios of gears materials; \( b \) – gear width; \( w \) – number of pairs in the engagement; \( \rho_j \) – the reduced radius of curvature of the working teeth profiles.

The reduced radius of curvature is expressed as follows

\[ \rho_j = \rho_{j_1} \rho_{j_2} \rho_{j_3} / \rho_{j_1} + \rho_{j_2} + \rho_{j_3} , \]  

where \( \rho_{j_1}, \rho_{j_2}, \rho_{j_3} \) – respectively, the curvature of the teeth lateral surfaces of the gear and the pinion; \( j = 0, 1, 2, 3, \ldots, s \).

(12)

Accordingly, the ratios for the radii of curvature at the \( j^{th} \) point of the engagement is expressed as follows

\[ \rho_{j_1} = r_{h_1} \tan \alpha_{a,j} \]

where \( r_{h_1} = r_1 \cos \alpha \), \( \alpha_{a,j} = \tan (\tan \alpha_{a,j} + j \Delta \phi) \).

\[ \tan \alpha_{a,j} = (1 + u)(tg \alpha - u \cos \alpha) \sqrt{r_{20} / r_2} \cos^2 \alpha , \]

\[ r_2 = m_{z_2} , \]

\[ r_{20} = r_{u_2} - r , \]

\[ r_{u_2} = r_{z_2} + m , \]  

(13)

\[ \rho_{j_2} = r_j \sqrt{r_{j_2} / r_2} \cos^2 \alpha , \]

\[ r_{j_2} = r_j \cos \alpha / \cos \alpha_{a,j} , \]

\[ r_0 = \sqrt{r_{1} / r_{a,j}} \cos \alpha - \alpha_{a,j} \cos \alpha_{a,j} \cos \alpha_0 \]

\[ \tan \alpha_{a,j} = (1 + u)(tg \alpha - u \cos \alpha) \sqrt{r_{20} / r_2} \cos^2 \alpha , \]

\[ a = (z_1 + z_2)m / 2 , \]

(16)

During the operation of the MP gears, the teeth of the polymer gear will wear out, which leads to an increase in the radii of curvature of their profile and, accordingly, to a decrease in the contact pressures in the gear. In [18, 20, 21] a method for determining the contact pressures taking into account the linear teeth wear \( h_{ij}'^* \) of the polymer gear was developed:

- in each separate interaction (procedure of the most accurate calculation);
- during a certain cycle of interactions (revolutions in one minute, one hour, 10 hours, 100 hours, etc.).

The advantage of the second technique of calculation is a significant reduction in the duration of calculations compared to the procedure of the accurate calculation.

In the first case, the resulting linear teeth wear \( h_{2,ja} \) of the polymer gear at \( j^{th} \) points of the profile will be reached after \( n_{2,a} \) revolutions

\[ h_{2,ja} = \sum_{j=1}^{n} h_{2,j} , \]

(17)

where \( h_{2,j} \) = var in each subsequent gear turn.

In the second case, the resulting linear teeth wear \( h_{2,ja} \) of the polymer gear will be achieved after a...
certain number of interaction blocks $B$ with the received number of teeth interaction cycles $C$ with constant contact parameters. Accordingly, in this case, the total number of revolutions $n_{2B} < n_{2s}$.

\[ h_{2jn} = \sum_{j=1}^{B} h_{2jB} , \quad (18) \]

where $h_{2jB} = \sum_{h=1}^{h'} h_{2j} = \text{const}$ in each block of interactions, however $h_{2B} = \text{var}$ in each subsequent block.

Therefore, after each interaction ($n = 1$ revolution) or block of tooth interactions ($n > 1$ revolution) the following parameters will change: $h_{2jB}, h_{2B}, \rho_{1j}, \rho_{2j}, \rho_{j}, p_{\text{max}}$, $B$, $\tau'$. In each block $B$ of teeth interactions, the change of parameters is calculated as the product of the block size on their value in a single cycle at invariable initial conditions of contact. Then these accumulated changes of the specified parameters are considered as starting points in the next block of calculations. The procedure is repeated cyclically until the given acceptable linear wear is reached at one of the characteristic points of the gear tooth profile (at the entrance to the two-pair, at the entrance or exit of the one-pair, at the exit of the second phase of two-pair engagement).

Gears durability $t$ at rotational speed $n_{2s}$ or $n_{2B}$ of polymer gears, when the acceptable teeth wear $h_{2jn}$ is reached, is expressed as follows

\[ t = n_{1} / 60n_{1} = n_{2s} / 60n_{2} , \quad (19) \]

where $n_{2} = n_{1} / u$ – the gear rotational speed.

The acceptable (limit) teeth wear in gears depending on operating conditions can be reached within several hundred million turns of a pinion. Therefore, it is important to determine the spread of results of the durability calculation by the block calculation procedure, which significantly reduces the duration of the calculations, concerning the accurate solution. The results of such studies of the influence of the teeth interactions block size on the spur gears durability are given in [25]. The following sizes of teeth interactions blocks with constant conditions were investigated: $B = 700, 42000, 84000, 420000, 2100000$ revolutions of the pinion. It is established that the durability of the gear with the size of the block $B = 2100000$ revolutions relatively to $B = 700$ revolutions is minimally higher (up to 0.3%). Instead, computational time $\tau_{B=2100000}$ relative to $\tau_{B=700}$ is less about 2720 times. The duration of calculations at $B = 1$ revolution would be about 110 hours, and at $B = 2100000$ revolutions is about 5.5 seconds. That is, the calculation duration is reduced by more than 72,000 times.

Dependence of calculation duration $\tau_{B}$ on teeth interactions block size $B$ is given in Fig. 1 (solid line). The dashed line shows the extrapolated values for smaller and larger block sizes.

3. MATERIALS, TRIBO-EXPERIMENTAL RESEARCH

According to the pin-on-disk scheme in the conditions of dry friction tribo-experimental researches of unfilled polyamides PA6, PA66 and composites PA6+30GF, PA6+30CF, PA6+MoS$_2$, PA6+Oil in pair with carbon steel (0.45% C) were carried out. Research program: $T_{\text{nom}} = 4.0$ Nm; sliding speed $v = 0.4$ m/s; contact pressures $p = 2, 4, 6, 8, 12$ MPa; path of friction $L = 5000 \ldots 10000$ m; diameter of the pin sample $d = 3$ mm. The working chamber was provided with forced air cooling to $T = 23\pm1^\circ$C at relative humidity 50\%±5\% (standard ISO 7148-2).

The author’s method of model research of materials wear [16, 19, 21, 24] at sliding friction provides several levels of loading in tribo-couple and definition of samples wear (weight or linear) and specific friction force $\tau$ by Coulomb’s law. This approach extends the requirements of ISO 7148-2 and ASTM G99, which provide to conduct experimental studies of wear at a constant value of contact pressure.

The results of tribo-experimental studies of these polymeric thermoplastic materials are presented in Fig. 2 and 3. In particular, Fig. 2 presents diagrams of their wear resistance.

![Figure 1. Influence of the teeth interactions block size on the calculation duration](image)

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![Figure 2. Wear resistance diagrams of polyamides](image)
Based on the results of model tribo-experimental studies, the wear resistance characteristics \( C \), \( m \) of the tribo-couple materials in the selected range of specific friction forces \( \tau = f_{\mu} \) are established. In the methodology of studying the wear kinetics during sliding friction, as described above, it is assumed that the wear rate (intensity) of the tribosystem materials depends on the level of specific friction force \( \tau \). Thus, using the characteristics of wear resistance of materials \( C \), \( m \) according to the above calculation method, the assessment of durability and wear of gears is performed.

Here, markers show the experimental wear resistance indicators \( \Phi \), for each level of specific friction force \( \tau = f_{\mu} \). The wear resistance indicators \( \Phi \), at the same contact pressure are located on the axis of specific friction force differently depending on the coefficient of sliding friction \( f \). As a result of their approximation, the wear resistance indicators diagrams are plotted as graphical indicators, according to which it is easy to compare the wear resistance diagrams indicate the nonlinearity of the dependence of wear on the specific friction force.

By approximation of the wear resistance indicators \( \Phi_i \) of the studied polymers according to Eq. (2), their wear resistance characteristics \( C \), \( m \) were established, which are given in Table 1.

### Table 1. Polyamides and composites based on them

<table>
<thead>
<tr>
<th>Wear resistance characteristics</th>
<th>Thermoplastic polymeric materials</th>
</tr>
</thead>
<tbody>
<tr>
<td>( C \times 10^5 )</td>
<td>PA6</td>
</tr>
<tr>
<td>( m )</td>
<td>1.34</td>
</tr>
<tr>
<td>( \tau_\mu \cdot \text{MPa} )</td>
<td>40</td>
</tr>
</tbody>
</table>

Note: PA6+30GF – fiberglass filler (30% by volume) with small fibers, PA6+30CF – carbon fibers (small fibers - 30%), PA6+MoS2 – molybdenum disulphide, PA6+Oil - oil-filled cast polyamide.

Accordingly, Fig. 3 shows a comparison of wear resistance (ranking by wear resistance) of the studied polyamides relative to the base unfilled polyamide PA6 at \( \tau = 4 \) MPa.

### Figure 3. The polyamides relative wear resistance in relation to PA6

The relative wear resistance \( \Phi \) of polyamides PA6 and PA66 and PA6 based composites is different depending on the type of filler.

### 4. CALCULATION RESULTS, DISCUSSION

MP spur gears with involute teeth consisting of a steel pinion and a polymer gear are considered. Gears materials:
- normalized ground carbon steel (0.45 \%C), \( E = 2.1 \times 10^5 \) MPa, \( v = 0.3 \); \( C = 10^2 \); \( m = 2 \);
- polyamides and composites – table 2.

<table>
<thead>
<tr>
<th>Characteristics of polyamides</th>
<th>Thermoplastic polymeric materials</th>
</tr>
</thead>
<tbody>
<tr>
<td>Young’s module ( E ), ( \text{MPa} )</td>
<td>2000</td>
</tr>
<tr>
<td>Poisson’s ratio ( v )</td>
<td>0.4</td>
</tr>
<tr>
<td>Coefficient of friction ( f )</td>
<td>0.23</td>
</tr>
</tbody>
</table>

Initial data for calculation: \( T_{\text{nom}} = 4000 \text{ N·mm}, n_1 = 700 \text{ rpm}, B = 420000 \text{ revolutions (10 hours of work)}; K_c = 1.2; m = 4 \text{ mm}, u = 3; z_1 = 20; z_2 = 60; b = 50 \text{ mm}, h_2* = 0.5 \text{ mm}, \varepsilon_0 = 1.372; \) end overlap coefficient.

The results of the numerical solution for the calculation of contact and tribotechnical parameters of MP gears are presented in Fig. 4 - 7. Thus, Fig. 4 shows the change in the initial maximum contact pressures\( p_{\text{max}} \) at selected points of contact during the cycle of teeth engagement.

### Figure 4. Maximum contact pressures

On the left is the area of the first phase of two-pair engagement (\( \Delta \phi = 0...6.69^\circ \)). The central area is a one-pair engagement (\( \Delta \phi = 6.69^\circ...18^\circ \)). On the right is the area of the second phase of two-pair engagement (\( \Delta \phi = 18^\circ...24.7^\circ \)). In the area of one-pair engagement, the pitch point with the angular coordinate \( \Delta \phi = 13.08^\circ \) is also marked. The highest maximum contact pressures occur at the teeth entrance in a one-pair engagement, and slightly lower (8,7...9,0\%) they will be at the teeth entrance in the first phase of two-pair engagement. It should be noted that the pressure at the pitch point, which according to standard methods are determined at the pitch point when calculating the contact strength of gears, will be slightly lower not only than the pressure at the entrance to the two-pair engagement (5,2...5,7\%) but also much lower than the pressure at the entrance to the one-pair engagement (14,3...15,0\%).

The use of different types and structures of fillers in the base polyamide PA6 allows varying the level of...
maximum contact pressures in the engagement on 41%. The analysis of the kinetics of contact pressure $p_{max}$ changes confirms the decisive influence of Young's modulus on their value. The harder the polymeric material, i.e., the larger its Young's modulus (Table 2), the higher the level of contact pressures. Young's moduli of PA6 and PA6+Oil are close and therefore the contact pressures also almost coincide.

Due to the polymer teeth wear the pressure $p_{max}$ will decrease slightly at the entrance to the two-pair and one-pair engagement (on average on $\approx 5.4\%$), as shown in Fig. 5.

![Figure 5. Wear-contact maximum pressures](image)

Teeth wear has a greater effect on reducing the pressure $p_{max}$ at the exit of single-pair engagement and in the second phase of two-pair engagement. When the teeth of the polymer gear wear out, there is no significant change in the initial maximum contact pressures since Young's moduli of steel (pinion teeth) and polyamide polymers (gear teeth) differ by 64...128 times. Contact pressures in the pitch point remain unchanged, because there the sliding speed is almost zero, and therefore, accordingly, there is no teeth wear.

The course of linear gear teeth wear $h_{2j}$ at selected points of engagement is shown in Fig. 6.

![Figure 6. Linear wear of the teeth profile of polymer wheels](image)

The gear teeth acceptable wear $h_{2s} = 0.5$ mm is reached at the entrance to the first phase of two-pair engagement. Although the pressure $p_{max}$ at the teeth entrance in two-pair engagement is slightly higher than at the entrance to their one-pair engagement, for sliding speeds the trend is reversed. Therefore, the teeth' wear at the entrance to the one-pair engagement is close to acceptable (Eq. (7)). That is, at these two characteristic points the teeth' wear is equal. At the teeth exit from the one-pair engagement and the second phase of two-pair engagement, they are smaller. Presented in Fig. 6 patterns of teeth linear wear are the same for all studied polyamides. The durability of gears made of different types of polyamides will be significantly different, as shown below. It should be noted that the pinion steel teeth wear five orders of magnitude less than the polymer gear teeth.

Minimum estimated durabilities $t_{min}$ and $t_{Bmin}$ of MP gears are shown in Fig. 7.

![Figure 7. The minimum durability of MP gears: a) absolute, b) relative](image)

It is accepted as the durability of the minimum gear the durabiliy at a point of a tooth profile where the acceptable (maximum) wear is reached. In the studied gears, it is the point of teeth entry into the first phase of two-pair engagement ($\Delta \varphi = 0$). The gears with an unfilled polyamide PA6 have the lowest durability. The gear made of PA6+30GF has 1.25 times higher durability than the gear made of PA6, although the wear resistance of these materials differs by 1.78 times (Fig. 3). The ratio of the gear's durability with gears made of PA66 and PA6 is 1.46 times, as approximately the ratio of their wear resistances (1.48 times). Although the wear resistance of PA6+30GF is 1.20 times higher than PA66 (Fig. 2, 3) however, the durability of this gears is 1.16 times lower than PA66. This is due to the 35% higher coefficient of friction in the pair steel – PA6+30GF (table 2), than steel – PA66. Gears made of polyamide composites have significantly higher durability: PA6+MoS2 (2.19 times), PA6+30CF (2.43 times), PA6+Oil (2.67 times).

Therefore, by modifying (filling) the base polyamide with different components, it is possible to increase the
MP gear's durability by 2.68 times. The results show that in the investigated MP gears the durabilities calculated by the simplified method according to (9) and by the block method according to (19) are close. This is since the change in the maximum contact pressures due to wear is also relatively small (Fig. 5).

5. CONCLUSION

1. According to the author's method of gears studying the purpose of the work is realized: the calculation of maximum contact pressures during the engagement cycle, their changes due to tooth wear, the linear wear of polymer gear tooth along with the profile, and the minimum gears durability.

The presented analytical method expands the possibilities of research of MP gears in comparison with the calculation methods presented in the standards ISO 6336 - 2 [26] and DIN 3900 [27], VDI 2736 [29]. In particular, it is possible:

a) calculate the contact pressure, sliding speed, radii of profile curvature, teeth wear and transmission durability at any point of the engagement;

b) to establish points (angles) of change in two-one-two-pair teeth engagement, a point of their exit from engagement; accordingly, this is also done for three-two-three-pair engagement;

c) assess the impact of tooth wear on the change of their radii of curvature, and, accordingly, on the contact pressures and transmission durability;

d) take into account the effect of gear correction on the contact and tribo-technical characteristics of the MP gears.

2. By the model trib-o-experimental studies of polyamides PA6, PA66, and composites PA6 + 30GF, PA6 + 30CF, PA6 + MoS2, PA6 + Oil at dry friction according to the pin-on-disk friction scheme (ISO 7148-2) their wear resistance indicators were determined. The least-squares method establishes their wear resistance characteristics as the basic parameters of the mathematical model for studying wear kinetics. The wear resistance diagrams of the specified polymeric materials are plotted. The highest wear resistance has PA6 + 30CF, PA6 + MoS2, PA6 + Oil at dry friction and, Accordingly, PA6 + Oil is much higher than PA6 + 30CF, and PA6 + MoS2. The highest wear resistance of PA6 + Oil and PA6 + MoS2 is an essentially zero value. The product of the sliding speed and the specific friction force determines the amount of linear tooth wear at different points of the profile.

5. The quantitative influence of different types of base polyamide PA6 fillers has been studied to improve the performance of metal-polymer gears. It was found that the durability of gears with composites PA6 + MoS2, PA6 + 30CF, PA6 + Oil is much higher compared to the base polyamide PA6.

6. Using the given calculation method effectively solves important engineering practice issues of estimation the teeth contact strength at the most loaded points of contact and the prediction of gears durability, as well as a very important task of optimal choice of polymeric materials for MP gears according to these criteria.

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МОДЕЛИРАЊЕ НОСИВОСТИ И ИЗДРЖЉВИОСТИ МЕТАЛ-ПОЛИМЕРСКИХ ЗУПЧАНИКА НА БАЗИ ПОЛИЛАМИД ПА6 КОМПОЗИТА

М. Чернец, А. Корниjenко, А. Духота, Ї. Чернец

На основу ауторске методе прорачуна метал-полимер (МП) зупчанка је извршено процување носивости и издржљивости метал-полимер зупчаника од непуњених полиамида ПА6, ПА66 и полиамид ПА6 композита пуњених стаклом (ПА66 + 30ГФ) или угљеничних (ПА6 + 30ЦФ) влакана, односно лијевог полиамида пуњеног мољебен дисульфидом (ПА6 + МоС2) (ПА6 + Оил). Узети су у обзир услови захвата зубаца и промена радиуса закривљености еволвентног профила током хода. Извршава се прорачун максималних контактних притисака током циклusa интеракције зубача, од узгаја до захвата до изласка из њега. Проучава се и утицај хода зуба на њихово смањење. Утврђује се линеарно трошење поли- мерних зубаца на изабраним тачкама профила и карактеристична тачка на којој се постиже прихваљива вредност (при двопарном или једнопарном
Израчуната је издржљивост зупчаника и дата је њихова упоредна процена. Утврђене су квалитативне и квантитативне законитости промене наведених експлоатационих карактеристика МП зупчаника при постизану прихватљивог хабања полимерних зубаца зупчаника.