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

M. Chernets
Professor
National Aviation University
Aerospace faculty
Ukraine

A. Kornienko
Associate Professor
National Aviation University
Aerospace faculty
Ukraine

A. Dukhota
Professor
National Aviation University
Aerospace faculty
Ukraine

Yu. Chernets
Ph.D. student
National Aviation University
Aerospace faculty
Ukraine

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 + MoS_2) 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 teeth 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+MoS_2, 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 other, 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+MoS_2, PA6+Oil. However, despite the sufficiently wide and growing use of MP gears, effective calculation methods for evaluating these specified 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.

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Correspondence to: Dr M. Chernets, National Aviation University, Aerospace faculty, 1, Liubomyra Huzara ave., Kyiv, Ukraine, 03058
Email: myron.czerniec@gmail.com
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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-disk 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 + MoS_2, 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:

$$\frac{1}{v}\frac{dh_1}{dt} = \Phi_1^{-1}(\tau),\ \frac{1}{v}\frac{dh_2}{dt} = \Phi_2^{-1}(\tau), \tag{1}$$

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 $\Phi_i(\tau_i)$:

$$\Phi_k(\tau) = C_k\left(\frac{\tau_S}{\tau}\right)^{m_k}, \tag{2}$$

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

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

$$\Phi_i(\tau_i) = L / h_i, \tag{3}$$

where h_i – experimental values of linear wear of material samples; L = v_t – 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

$$t_k = \frac{C_k \tau_S^{m_k}}{v}\int_0^{h_k^*} \tau^{-m_k} dh_k,\ \ m_k \rangle 0. \tag{4}$$

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

$$t_k = \frac{C_k}{v}\left(\frac{\tau_S}{\tau}\right)^{m_k} h_k. \tag{5}$$

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

$$h_k = \frac{vt_k}{C_k}\left(\frac{\tau}{\tau_S}\right)^{m_k}. \tag{6}$$

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'_{kj} at any point of their lateral surface during a single time of tribo-contact t'_j for single-pair engagement is expressed as follows

$$h'_{kj} = \frac{v_j t'_j \left(fp_{j\max}\right)^{m_k}}{C_k\left(0.5\sigma_B\right)^{m_k}} = \frac{v_j t'_j \left(\tau_{j\max}\right)^{m_k}}{C_k\left(\tau_S\right)^{m_k}} \tag{7}$$

where $v_j = v$ – sliding speed at the j^{th} points 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; $v_0 = \omega_1 r_1 \sin\alpha$ – the speed of movement of the point of contact along the contour of the tooth; ω_1– the pinion angular velocity; $r_1 = z_1 m$ – the pinion pitch radius; m – module; α = 20° – pressure angle; z_1 – 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:

$$h_{kj} = 60 n_k h'_{kj} t^* \quad (8)$$

where n_k – number of revolutions of gears.

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

$$t = h_{k*} / \bar{h}_{kj}, \; \bar{h}_{kj} = 60 n_k h'_{kj} . \quad (9)$$

The sliding speed is calculated by the formula

$$v_j = \omega_1 r_{b1} (tg\alpha_{1j} - tg\alpha_{2j}), \quad (10)$$

where $\alpha_2 = \arccos[(r_2/r_{2j})\cos\alpha]$.

The maximum contact pressures $p_{jmax} = -\sigma_H$ and the contact area width $2b_j$ at the j^{th} point of the profile are calculated by Hertz formulas

$$p_{j\max} = 0{,}564\sqrt{N'/\theta\rho_j}\,,\; 2b_j = 2{,}256\sqrt{\theta N'\rho_j}\,, \quad (11)$$

where σ_H – contact pressure; $N' = N/bw$; $N = T_{nom}K_g/r_1 \cos\alpha$ – force in the engagement; $T_{nom} = 9550\ P/n_1$ – rated torque; P – power on the driving shaft; n_1 – the pinion rotational speed; K_g – dynamism coefficient; $\theta = (1-v_1^2)/E_1 + (1-v_2^2)/E_2$; E, v – Young's modulus and Poisson's ratios of gears materials; b – gear width; w – number of pairs in the engagement; ρ_j – the reduced radius of curvature of the working teeth profiles.

The reduced radius of curvature is expressed as follows

$$\rho_j = \frac{\rho_{1j}\rho_{2j}}{\rho_{1j} + \rho_{2j}},\; j = 0, 1, 2, 3,\ldots, s \quad (12)$$

where ρ_{1j}, ρ_{2j} – respectively, the curvature of the teeth lateral surfaces of the gear and the pinion; $j = 0$, $j = s$ correspond to the first and last point of the teeth engagement.

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

$$\rho_{1j} = r_{b1}\tan\alpha_{1j} \quad (13)$$

where $r_{b1} = r_1\cos\alpha$, $\alpha_{1j} = a\tan(\tan\alpha_{10} + j\Delta\varphi)$,

$\tan\alpha_{10} = (1+u)tg\alpha - \dfrac{u}{\cos\alpha}\sqrt{(r_{20}/r_2)^2 - \cos^2\alpha}$,

$r_2 = mz_2$, $r_{20} = r_{a2} - r$, $r_{a2} = r_2 + m$, $r = 0{,}2m$,

$\tan\alpha_{1s} = \sqrt{(r_{1s}/r_1)^2 - \cos^2\alpha}$, $r_{1s} = r_{a1} - r = r_{a1} - 0{,}2m$,

$r_{a1} = r_1 + m$.

$$\rho_{2j} = r_2\sqrt{(r_{2j}/r_2)^2 - \cos^2\alpha} \quad (14)$$

where $r_{2j} = \sqrt{a^2 + r_{1j}^2 - 2ar_{1j}\cos(\alpha - \alpha_{1j})}$,

$r_{1j} = r_1\cos\alpha/\cos\alpha_{1j}$, $\cos\alpha_{20} = \dfrac{r_2}{r_{20}}\cos\alpha$,

$\tan\alpha_{2s} = \left(1 + \dfrac{1}{u}\right)\tan\alpha - \dfrac{1}{u\cos\alpha}\sqrt{\left(\dfrac{r_{1s}}{r_1}\right)^2 - \cos^2\alpha}$,

$a = (z_1 + z_2)m/2$, r_1, r_2 respectively the radii of the pitch circles of the gear and the pinion; r_{b1}, r_{b2} – radii of the base circles; r_{a1}, r_{a2} – radii of the addendum circles; r - rounding radius of the addendum; u – gear ratio; $\Delta\varphi$ – the angle of the tooth rotation from the point of initial contact (p.0) to point 1, etc.; α_{10} – the angle corresponding to the 1st point of the line of contact in a fixed coordinate system yO_1x; α_{1s} – the angle that determines the position of the last point of engagement of the pinion tooth on the line of contact; α_{20}, α_{1s} – the angles that determine the position of the first and last point of engagement of the gear tooth on the line of contact; z_1, z_2 – numbers of teeth.

In spur gear, two - one - two - pairs engagement of teeth is realized resulting in a fundamental change of the conditions of contact interaction. The angles of transition from two-pair ($\Delta\varphi_{1F2}$) to one-pair and again to two-pair ($\Delta\varphi_{1F2}$) engagement are calculated as follows:

$$\Delta\varphi_{1F_2} = \varphi_{10} - \varphi_{1F_2},\; \Delta\varphi_{1F_1} = \varphi_{10} + \varphi_{1F_1}; \quad (15)$$

where $\varphi_{1F2} = \tan\alpha_{F2} - \tan\alpha$, $\varphi_{1F1} = \tan\alpha_{F1} - \tan\alpha$, $\varphi_{10} = \tan\alpha_{10} - \tan\alpha$;

$\tan\alpha_{F_2} = \dfrac{r_1\sin\alpha - (p - e_1)}{r_1\cos\alpha} = \tan\alpha - \dfrac{p - e_1}{r_1\cos\alpha}$,

$\tan\alpha_{F_1} = \dfrac{r_1\sin\alpha - (p - e_2)}{r_1\cos\alpha} = \tan\alpha - \dfrac{p - e_2}{r_1\cos\alpha}$; $p = \pi m$

$\cos\alpha$ – gear step; $e_1 = \sqrt{r_{1s}^2 - r_{b1}^2} - r_1\sin\alpha$,

$e_2 = \sqrt{r_{20}^2 - r_{b2}^2} - r_2\sin\alpha$.

Exit angle $\Delta\varphi_{1E}$ a pair of teeth from the engagement is determined according to the formula

$$\Delta\varphi_{1E} = \varphi_{10} + \varphi_{1E}; \quad (16)$$

where $\varphi_{1E} = \tan\alpha_E - \tan\alpha$, $\alpha_E = \mathrm{acos}(r_{b1}/r_{1s})$.

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'_{kj} 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_{2jn} of the polymer gear at j^{th} points of the profile will be reached after n_{2s} revolutions

$$h_{2jn} = \sum_{1}^{n_{2s}} h'_{2j}\,, \quad (17)$$

where h'_{2j} = var in each subsequent gear turn.

In the second case, the resulting linear teeth wear h_{2jn} 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 $n_{2B} < n_{2s}$. Then

$$h_{2jn} = \sum_{1}^{B} h_{2jB}\ , \tag{18}$$

where $h_{2jB} = \sum h'_{2j} = const$ in each block of interactions, however h_{2jB} = 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'_{2j}, h_{2jB}, ρ_{1j}, ρ_{2j}, ρ_j, p_{jmax}, $2b$, t'_j. 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

$$t = n_{1s} / 60n_1 = n_{2s} / 60n_2\ , \tag{19}$$

where $n_2 = n_1 / u$ – the gear rotational speed.

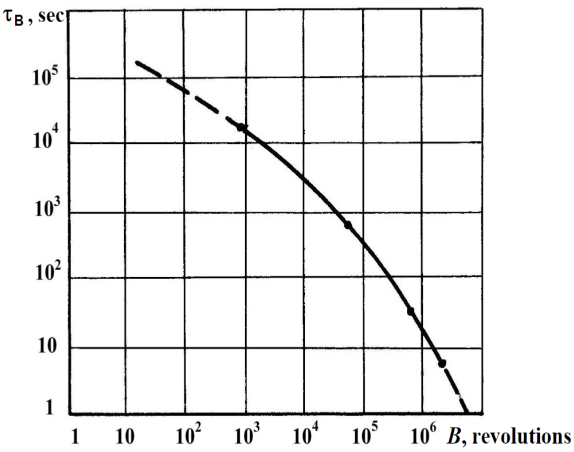

Figure 1. Influence of the teeth interactions block size on the calculation duration

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 $\tau_{B2100000}$ relative to τ_{B700} 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 τ_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 triboexperimental 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_{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 …10000 m; diameter of the pin sample d = 3 mm. The working chamber was provided with forced air cooling to T = 23±1°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 τ 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.

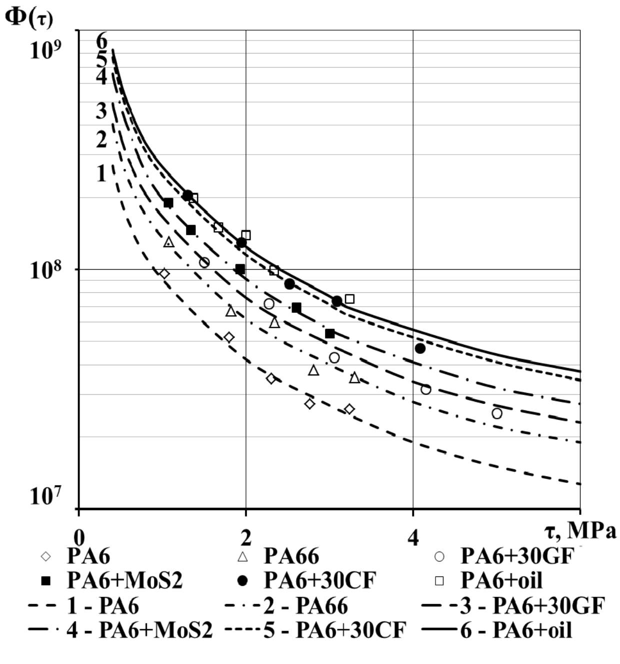

Figure 2. Wear resistance diagrams of polyamides

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.

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 $\tau = f_{p0}$ 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 τ. 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 Φ_i for each level of specific friction force $\tau_i = f_{pi}$. The wear resistance indicators Φ_i at the same contact pressure are located on the axis of specific friction force differently depending on the coefficient of sliding friction f in tribo-couple. As a result of their approximation, the wear resistance diagrams are plotted as graphical indicators, according to which it is easy to compare the wear resistance of polymers at different specific friction forces. Unfilled polyamide PA6 has the lowest wear resistance, and PA6 + oil has the highest. The wear resistance diagrams indicate the nonlinear nature of the dependence of wear on the specific friction force.

By approximation of the wear resistance indicators Φ_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

Wear resistance characteristics	Thermoplastic polymeric materials					
	PA6	PA66	PA6+ 30GF	PA6+ MoS_2	PA6+ 30CF	PA6+ Oil
$C\cdot 10^6$	1,34	1,98	1,88	3,08	3,67	4,20
m	1,15	1,15	1,15	1,15	1,15	1,15
τ_S , MPa	40	40	50	38	40	38

Note: PA6+30GF – fiberglass filler (30% by volume) with small fibers, PA6+30CF – carbon fibers (small fibers - 30%), PA6+MoS_2 – molybdenum disulfide, 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 τ = 4 MPa.

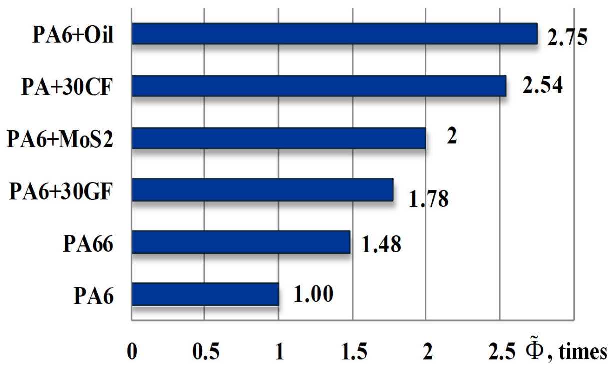

Figure 3. The polyamides relative wear resistance in relation to PA6

The relative wear resistance $\tilde{\Phi}$ 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 grinded carbon steel (0.45 %C), E = 2,1·10^5 MPa, ν = 0,3; C = 10^9, m = 2;
- polyamides and composites – table 2.

Table. 2. Characteristics of polyamides and composites

Characteristics of polyamides	Thermoplastic polymeric materials					
	PA6	PA66	PA6+ 30GF	PA6+ MoS_2	PA6+ 30CF	PA6+ Oil
Young's module E, MPa	2000	2300	2700	1660	3300	1960
Poisson's ratio ν	0.4	0.4	0.41	0.4	0.41	0.4
Coefficient of friction f	0.23	0.23	0.31	0.23	0.25	0.25

Initial data for calculation: T_{nom} = 4000 N·mm, n_1 = 700 rpm; B = 420000 revolutions (10 hours of work); K_g = 1,2; m = 4 mm, u = 3, z_1 = 20, z_2 = 60, b = 50 mm, h_{2*} = 0,5 mm, ε_α = 1,372 – end overlap coefficient.

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

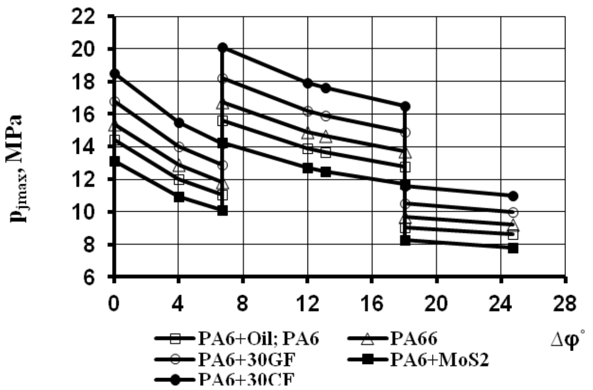

Figure 4. Maximum contact pressures

On the left is the area of the first phase of two-pair engagement (Δφ = 0…6,69°). The central area is a one-pair engagement (Δφ = 6,69°…18°). On the right is the area of the second phase of two-pair engagement (Δφ = 18°…24,7°). In the area of one-pair engagement, the pitch point with the angular coordinate Δφ = 13,08° 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_{jmax} changes confirms the decisive influence of Young's modulus on their value. The harder the polymeric material, i.e., the larger it's 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_{jmax} will decrease slightly at the entrance to the two-pair and one-pair engagement (on average on ≈ 5,4%), as shown in Fig. 5.

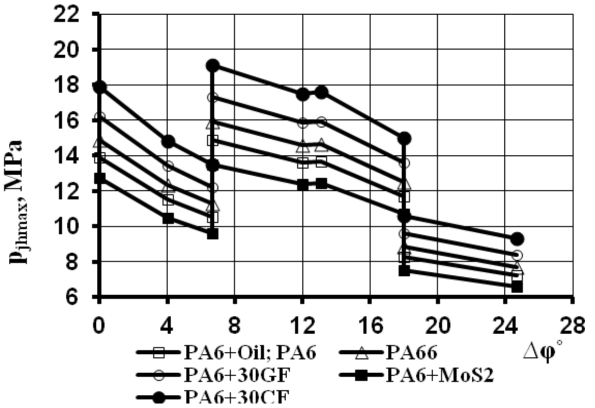

Figure 5. Wear-contact maximum pressures

Teeth wear has a greater effect on reducing the pressure p_{jmax} 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.

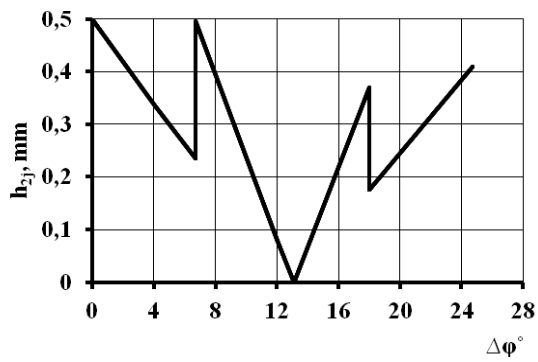

Figure 6. Linear wear of the teeth profile of polymer wheels

The gear teeth acceptable wear h_{2*} = 0.5 mm is reached at the entrance to the first phase of two-pair engagement. Although the pressure $p_{j\max}$ 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.

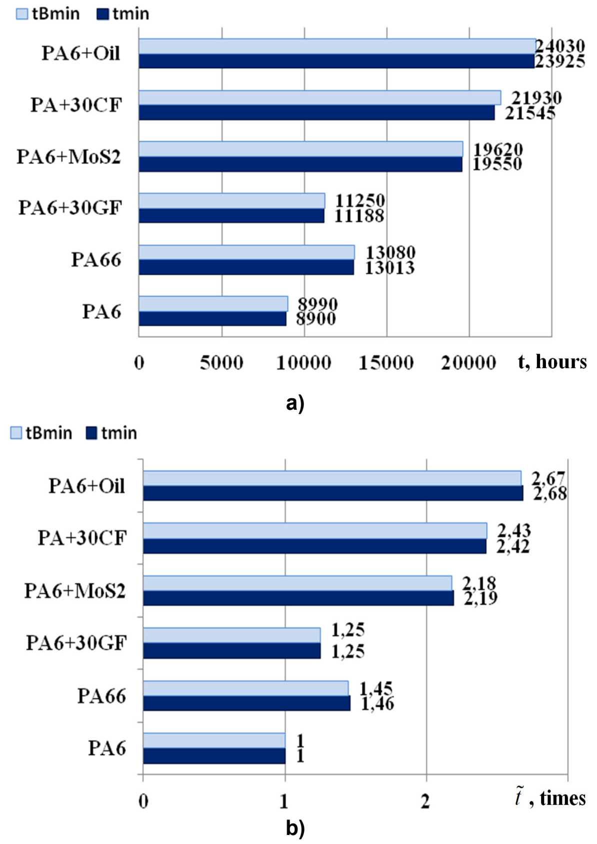

Figure 7. The minimum durability of MP gears: a) absolute, b) relative

It is accepted as the durability of the minimum gear the durability 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+MoS_2 (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 tribo-experimental studies of polyamides PA6, PA66, and composites PA6 + 30GF, PA6 + 30CF, PA6 + MoS_2, 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 oil-filled cast polyamide PA6 + Oil, and the lowest - base polyamide PA6.

3. Quantitative and qualitative regularities of change of the studied characteristics depending on a kind of polymeric materials are established. The highest maximum contact pressures occur in the engagement when using PA6 + 30CF, and slightly lower - when PA6 + 30GF. The pressure level depends on Young's modulus, which is the highest in these composites. The lowest is the maximum pressure when using the composite PA6+MoS_2.

4. The linear gear teeth wear is differentiated and depends on the sliding speed at different points of engagement (maximum $v_{j=0}$ = 0,84 m/s at the entrance to the first phase of two-pair engagement and slightly less $v_{j=0}$ = 0,745 m/s at the exit of the second phase of two-pair engagement). At the pitch point ($\Delta\varphi$ = 13,08°) there is a pure rolling and, accordingly, the sliding speed is almost zero. 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 + MoS_2, 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 ИЗДРЖЉИВОСТИ МЕТАЛ-ПОЛИМЕРСКИХ ЗУПЧАНИКА НА БАЗИ ПОЛИАМИД ПА6 КОМПОЗИТА

М. Чернец, А. Корнијенко, А. Духота, Ј. Чернец

На основу ауторске методе прорачуна метал-полимер (МП) зупчаника је извршено проучавање носивости и издржљивости метал-полимер зуп–чаника од непуњених полиамида ПА6, ПА66 и поли–амида ПА6 композита пуњених стаклом (ПА6). + 30ГФ) или угљеничних (ПА6 + 30ЦФ) влакана, односно ливеног полиамида пуњеног мо–либден дисулфидом (ПА6 + МоС2) (ПА6 + Оил). Узети су у обзир услови захватања зубаца и промена радијуса закривљености еволвентног профила током хабања. Извршава се прорачун максималних кон–тактних притисака током циклуса интеракције зубаца - од улаза до захвата до изласка из њега. Проучава се и утицај хабања зуба на њихово смањење. Утврђује се линеарно трошење поли–мерних зубаца на изабраним тачкама профила и карактеристична тачка на којој се постиже прих–ватљива вредност (при двопарном или једнопарном

захвату). Израчуната је издржљивост зупчаника и дата је њихова упоредна процена. Утврђене су квалитативне и квантитативне законитости промене наведених експлоатационих карактеристика МП зупчаника при постизању прихватљивог хабања полимерних зубаца зупчаника.