

Comparison of Load-Carrying Capacity, Wear, and Resource of Metal-Polymer Corrected Spur Gears with a Gear Made of Polyamides PA6 or PA6+30CF

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Based on the developed calculation research method of metal-polymer gears the analysis of the influence of height and angular correction of engagement of spur gears on load-carrying capacity, linear teeth wear, and resources is carried out. Gears with a pinion made of carbon steel 0.45%C and a gear made of basic polyamide PA6 or carbon fibre composite PA6 + 30CF were investigated. Quantitative and qualitative patterns of influence of both types of correction, changes in the initial teeth profile due to wear, mating conditions on the load carrying capacity, teeth linear wear, and service life of investigated metal-polymer gears have been established. Profile correction causes a decrease in the contact pressure in the engagement by different amounts during the meshing cycle. They are most significantly reduced in the input area of two-pair engagement. It was found that when using PA6 + 30CF, the maximum contact pressures are about 1.29 times higher than with PA6 at all values of the shift coefficients, regardless of its type. The tooth linear wear has different values during the meshing cycle depending on the shift coefficients and parity of engagement. The resource first increases with increasing the shift coefficients, reaching their optimum, and then decreases again. Regularities of this influence for each type of correction of engagement are established. Metal-polymer gears with a gear made of PA6 + 30CF will have a significantly higher service life than with a gear made of PA6.

Keywords: metal-polymer spur gear, height and angular correction of engagement, polyamides PA6 and PA6 + 30CF, maximum contact pressures, linear teeth wear, resource

Highlights

- Calculation of contact pressures, wear, and resources of corrected MP spur gears is performed.
- The influence of height and angular teeth correction on these service characteristics of gears is investigated.
- Quantitative and qualitative patterns of the effect of correction on these characteristics are given.
- The optimal values of the shift coefficients providing maximum gears resource are determined.
- A comparative evaluation of service characteristics of gears made of PA6 and PA6 + 30CF was performed.

0 INTRODUCTION

Metal-polymer (MP) gears are common in various fields of human activity, in a variety of technical devices, mechanisms, and machines. As with metal gears in MP gears for the purpose of increasing their loading carrying capacity, resources, and wear resistance, profile shifting is used. However, effective methods for calculating tribotechnical parameters, i.e., the tooth wear and resource of both uncorrected and corrected MP have not been developed. There are standardized calculation methods [1] and [2] and numerical methods [3] to [5] for studying uncorrected gears with metal-toothed wheels. In [3] to [5], the modelling of the tooth wear of spur and screw gears by numerical methods with Archard's linear law of abrasion wear was carried out. They were used to assess the load-carrying capacity of MP gears in [6] to [10], where the contact and bending teeth strength was studied. In [6] and [7], the MP gears with a gear made of polyamide PA6 were investigated for these stresses. In [8], the load capacity of spur gears with PA66

was studied. In contrast, virtually no computational methods exist for studying the wear and resources of MP gears in the literature. [9] presents simplified and [10] presents improved methods for calculating the wear resistance of an MP spur gears with a gear made of PA6 + 30GF, PA6 + 30CF with dispersed glass and carbon fibres. Dry friction using Archard's law of abrasive wear is considered, by which the wear is directly proportional to the contact pressure and depends linearly on the path of sliding friction. This wear mechanism is not inherent, much less dominant for MP gears.

Regarding the study of the effect of correction (height or angle) on the load-carrying capacity, wear, and resources of MP gears, such methods and relevant calculations are not available in the literature. There are some results in the literature of studies of metal gears with corrected teeth [11] to [16] for their load-carrying capacity and efficiency. Accordingly, in [11], on the basis of the wear model of screw gears, the influence of different types of profile shifting on wear according to Archard's law was investigated; in [12],

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the influence of the angular correction of spur gears on the contact and bending stresses was investigated; in [13] and [14], the analysis of influence of geometrical parameters of gears, including teeth correction to the specified stresses was given; in [15], the optimization algorithm for spur gears with height correction of engagement is presented; in [16], the influence of height correction on normal Mises stresses, bending stresses, and teeth deformation was studied.

Also, [17] provides an overview of various types of metal spur and helical gears with non-involute teeth, found in various applications. The comparison of involute and non-involute gears was carried out on the basis of the following criteria: Hertzian pressure, oil film thickness, bending stress at the root of the tooth, contact temperature, and gear noise.

Given the urgency of these scientific and technical problems, a modified method of calculating MP gears with correction of engagement was developed [18] and [19] based on the author's multicriteria generalized method of studying metal gears, including correction of engagement [21] to [25]. These calculation methods are based on the phenomenological methodology of research of wear resistance at sliding friction [20] and [21] by friction-fatigue mechanism. This approach is fundamentally different from the wear mechanism of metal-polymer tribocouples according to the adhesive-abrasive wear Archard's law.

The solution of the wear-contact problem is given in the article and the tribological behaviour of MP spur gears with a steel pinion and gears made of two polyamides: unreinforced PA6 or reinforced with short carbon fibres PA6 + 30CF composite is investigated. The study considered such real factors as the conditions of tooth engagement, height and angular correction of engagement, and the impact of tooth wear on contact pressures and resource.

1 CALCULATION METHODS

The calculation method aspects for the study of uncorrected MP spur gears are thoroughly presented in authors' previous works [18] and [19]. Therefore, the following are the basic design relationships for gears with height and angular correction of engagement.

The teeth contact strength of the MP gears is determined using the known Hertz formula for different gear materials. Respectively

$$p_{j\max} = 0.564 \sqrt{N' / (\theta \rho_j)}, \quad (1)$$

where $p_{j\max}$ is initial maximum contact pressures at the j^{th} point, which occur under the action of nominal

torque; $j=0, 1, 2, \dots, s$ the points of contact on the teeth profiles; $N'=N/(bw)$; $N=T_{\text{nom}}K_g/(r_1 \cos \alpha)$ force in engagement; $T_{\text{nom}}=9550P/n_1$ rated torque; P power on the driving shaft; n_1 pinion rotational speed; K_g dynamic coefficient; $r_1=z_1m$ pitch circle diameter of the pinion; m module; $\alpha=20^\circ$ pressure angle; z_1, z_2 numbers of teeth; $\theta = (1-v_1^2)/E_1 + (1-v_2^2)/E_2$; E, ν Young's modules and Poisson's ratios; b face width; w the number of teeth pairs in engagement; ρ_j reduced radius of curvature of the tooth profiles at the j^{th} point.

Respectively,

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

where ρ_{1j}, ρ_{2j} are the radii of curvature of the teeth profiles of the spur pinion and gear at j^{th} point [19] and [25], $j=0$ and $j=s$ correspond to the first and last point of the teeth engagement.

The linear teeth wear h'_{2j} of the polymer gear 2 at any point j of their profiles during a single time of their contact t'_j in a one-pair engagement at maximum contact pressure $p_{j\max}$ is calculated as follows [18], [21], [22] and [24].

$$h'_{kj} = \frac{v_j t'_j (f p_{j\max})^{m_k}}{C_k (0.5R_m)^{m_k}} = \frac{v_j t'_j (\tau_{j\max})^{m_k}}{C_k (\tau_S)^{m_k}}, \quad (3)$$

where $v_j = \omega_1 r_{b1} (t g \alpha_{1j} - t g \alpha_{2j})$ [21] and [22] sliding speed in the engagement; $t'_j = 2b_j / v_0$ time of teeth contact at movement of j^{th} contact point on a tooth profile to hertz contact patch width $2b_j = 2.256 \sqrt{\theta N' \rho_j}$; $v_0 = \omega_1 r_1 \sin \alpha$ the speed of contact point movement on the tooth profile; ω_1 pinion angular velocity; f sliding friction coefficient; $\tau_{j\max} = f p_{j\max}$ specific force of friction according to Coulomb's law; R_m tensile strength of polymeric material under tension (compression); τ_{S2} the shear strength of the gear polymer material; C_k, m_k wear resistance characteristics of gear materials; $k = 1, 2$, toothed wheels numbering (1 – pinion, 2 – gear).

In spur gears two - one - two-pair teeth engagement is realized. The angle $\Delta\varphi_{1F_2}$ of exit from the two-pair and at the same time the entrance to the one-pair engagement and the angle $\Delta\varphi_{1F_1}$ of exit from it is determined according to the author's method

During the gears operation due to the teeth slippage in the engagement, their wear inevitably occurs. Its variable value during each subsequent act of their tribocontact is calculated as follows [19], [21], [22], and [25]:

$$h'_{kin} = \frac{v_j t'_{jh} (fp_{jh\max})^{m_k}}{C_k (0.5R_m)^{m_k}}, \quad (4)$$

where $t'_{jh} = 2b_{jh}/v_0 = \text{var}$ time of teeth wear during movement of j that point of their contact along a tooth contour to a variable due to wear width of a contact patch $2b_{jh}$; $p_{jh\max}$ variable maximum contact pressure at the j^{th} point of teeth interaction during wear.

As a result of tooth profiles wear their initial radii of curvature increase ρ_{1j} , ρ_{2j} . Consequently, the maximum contact pressures $p_{j\max}$ are reduced and the width of the contact patch $2b_j$ at j^{th} points is increased. Accordingly, their current values of $p_{jh\max}$ and $2b_{jh}$ are calculated according to modified Hertz formulas [19] and [25]:

$$p_{jh\max} = 0.564\sqrt{N' / (\theta\rho_{jh})}, \quad 2b_{jh} = 2.256\sqrt{\theta N' \rho_{jh}}, \quad (5)$$

where $\rho_{jh} = (\rho_{1jh}\rho_{12h}) / (\rho_{1jh} + \rho_{12h})$ is the reduced radius of curvature of the gear profile, changeable in the result of wear, ρ_{1jh} , ρ_{2jh} changeable curvature radii of pinion and wheel tooth profiles.

The variable radii of curvature ρ_{kjh} are calculated according to the method given in [18], [19], and [25].

For the numerical solution, a block calculation procedure is used, in which, during a randomly selected number of teeth interactions (blocks of interactions B), the single wear is calculated according to Eq. (2) followed by linear summation [18] and [25]. In each subsequent block, the single wear of the teeth is calculated by Eq. (4).

For the number of revolutions of the pinion n_{1s} and the gear n_{2s} corresponding to a certain number of the accepted size of the interaction blocks B with constant contact conditions, their total wear h_{1jn} and h_{2jn} at j^{th} points of contact is determined as follows:

$$h_{1jn} = \sum_1^{m_s} h_{1jB}, \quad h_{2jn} = \sum_1^{n_{2s}} h_{2jB}, \quad (6)$$

where $n_{2s} = n_{1s}/u$; $h_{kjb} = \Sigma h'_{kj}$ teeth wear in each block of their interaction.

Then, considering the change in initial contact pressures $p_{j\max}$ due to tooth wear, the gears resource t_B at the calculated number of revolutions n_{1s} or n_{2s} is expressed as follows:

$$t_B = \frac{n_{1s}}{60n_1} = \frac{n_{2s}}{60n_2}. \quad (7)$$

The above method of calculating the uncorrected MP gears is also used [22], [24] and [26]. For gears with height and angular correction (Fig. 1).

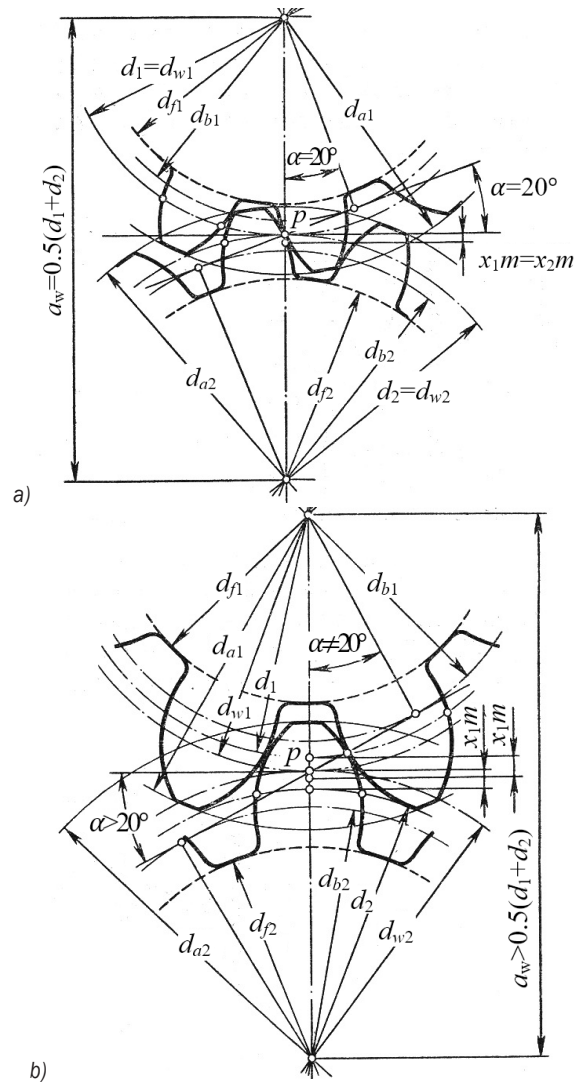


Fig. 1. a) Spur gears with height, and b) correction and angular correction

In gears with height correction, the shift coefficients are assumed to be the same $x_1 = -x_2$ and the total shift coefficient is $x_2 = x_1 + x_2 = 0$. Accordingly, the centre distance $a = r_1 + r_2$ remains the same as in the gears without offset. Only the addendum and dedendum radii will be variable

$$r_{a1} = r_1 + (1 + x_1)m, \quad r_{a2} = r_2 + (1 + x_2)m, \quad (8)$$

$$r_{f1} = r_1 + (1.25 - x_1)m, \quad r_{f2} = r_2 + (1.25 - x_2)m, \quad (9)$$

where $r_2 = mz_2$ is a pitch radius.

All formulas for calculating other parameters have the same structure as for uncorrected gears [22].

In gears with angular correction, the shift coefficients $x_1 \neq x_2$ (usually $x_1 > 0$, $x_2 > 0$); total shift coefficient $x_\Sigma > 0$; the corrected centre distance

$a_w=(r_{w1}+r_{w2})>a$; the corrected pressure angle $\alpha_w>\alpha$ on the initial pitch circle. Accordingly, the pitch radii of the pinion and the gear, radii of the addendum and dedendum are expressed as follows

$$r_{w1} = r_1 \frac{\cos \alpha}{\cos \alpha_w}, \quad r_{w2} = r_2 \frac{\cos \alpha}{\cos \alpha_w}, \quad (10)$$

$$r_{a1} = r_1 + (1 + x_1 - K)m, \quad r_{a2} = r_2 + (1 + x_2 - K)m, \quad (11)$$

$$\begin{aligned} r_{f1} &= r_1 + (1.25 - x_1 + K)m, \\ r_{f2} &= r_2 + (1.25 - x_2 + K)m, \end{aligned} \quad (12)$$

where $K=[(a-a_w)/m]+x_\Sigma$ is the reduction coefficient of the addendum heights.

At this type of correction due to changes in values of a_w , α_w , r_{w1} , r_{w2} , r_{a1} , r_{a2} , some formulas for calculation of geometrical parameters will have a changed structure [21] and [22], compared to uncorrected gears.

In practice, for a positive correction $x_\Sigma>0$ the inversely proportional method of resolution of x_Σ is often used [22]:

$$x_1 = \frac{z_2}{z_1 + z_2} x_\Sigma, \quad x_2 = x_\Sigma - x_1. \quad (13)$$

At height correction, the centre distance will not change, unlike the angular correction.

2 RESULTS OF THE SOLUTION, DISCUSSION

Data for calculation: $T_{nom}=4000$ Nmm, $n_1=700$ rpm; $B=420000$ revolutions (10 hours of operation) the size of the block with constant conditions of tribocontact; $K_g=1.2$; $m=4$ mm, $u=3$ gear ratio, $z_1=20$, $z_2=60$, $b=50$ mm, $h_{2*}=0.5$ mm acceptable teeth wear of the polymer gear.

Height correction: $x_1=-x_2=0, 0.1, 0.2, 0.3$; $a=160$ mm; $\alpha=20^\circ$. Angular correction: $x_\Sigma=0.3$; $x_1=0, x_2=0.3$; $x_1=0.05, x_2=0.25$; $x_1=0.1, x_2=0.2$; $x_1=0.2, x_2=0.1$; $x_1=0.25, x_2=0.05$; $a_w=161.169$ mm; $\alpha_w=21.11^\circ$. The recommended values of the shift coefficients for the specified number of teeth in the case of angular correction are $x_1=0.225, x_2=0.075$.

The characteristics of materials of MP gears:

- Pinion: carbon steel 0.45%C, normalizing, grinding, $E_1=2.1 \cdot 10^5$ MPa, $\nu_1=0.3$; $C_1=10^9$, $m_1=2$, $\tau_{S1}=365$ MPa.
- Gear: polyamide PA6, $E_2=2300$ MPa, $\nu_2=0.4$; $C_2=1.34 \cdot 10^6$, $m_2=1.15$, $\tau_{S2}=40$ MPa; $f=0.23$; polyamide composite PA6+30CF, filled with 30

% fine carbon fiber, $E_2=3300$ MPa, $\nu_2=0.41$; $C_2=3.67 \cdot 10^6$, $m_2=1.15$, $\tau_{S2}=40$ MPa; $f=0.25$.

The wear resistance of materials of metal-polymer tribocouples was established according to the pin-on-disk scheme in the conditions of dry friction according to the ISO 7148-2 standard ($T=23 \pm 1$ °C, relative humidity of air 50 ± 5 %) [27].

The results of research of height and angular correction influence on initial maximum contact p_{jmax} and wear contact p_{jhmax} pressures in engagement, linear wear h_{2j} of gear teeth, minimum gears resource t_{Bmin} are given in Figs. 2 to 6. Figs. 2 and 3 show the pressure level p_{jmax} in the meshing cycle for both types of correction. To the left and right are the areas of two-pair engagement (input and output), and in the centre, the area of one-pair engagement. Here $\Delta\phi$ is the angle of rotation of the pinion tooth from the point of the entry of the teeth into the engagement ($p.0$) to the next points with the selected step.

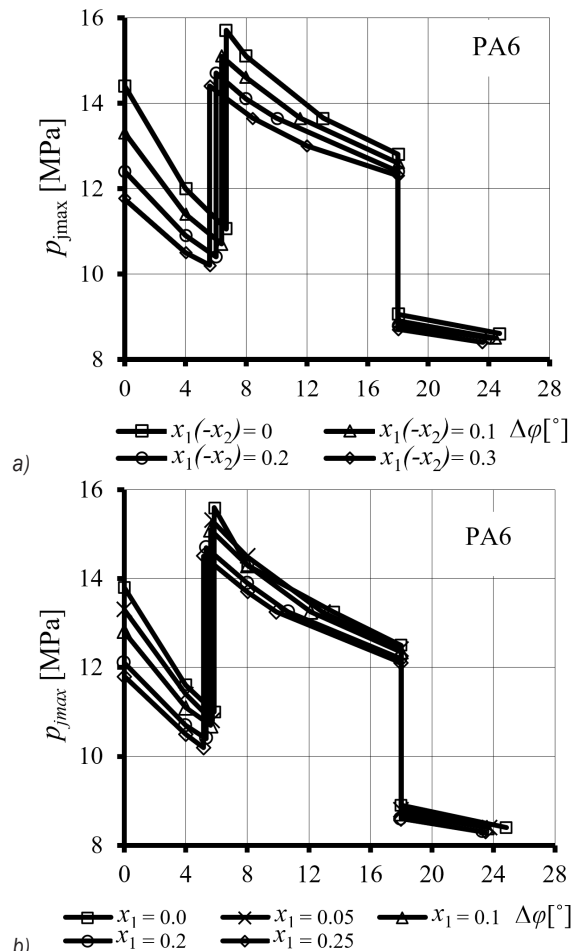


Fig. 2. Maximum contact pressures in engagement (PA6): a) height correction $[x_1(-x_2) \equiv (x_1 = -x_2)]$, b) angular correction

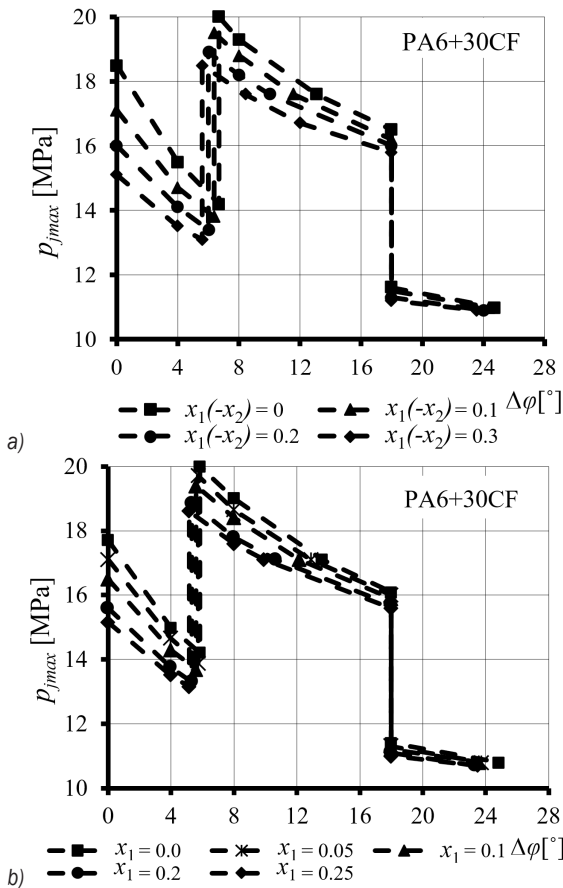


Fig. 3. Maximum contact pressures in engagement (PA6+30CF): a) height correction, b) angular correction

The obtained results of calculations show that at the entrance to the one-pair engagement, the initial contact pressures p_{jmax} reach the highest value. When the teeth are corrected, the contact pressures will decrease in different ways during the meshing process. At the entrance to the two-pair engagement up to 1.23 times for the height correction and up to 1.17 times for the angular correction, and then up to 1.14 times. In the area of one-pair engagement, the pressure drop due to tooth correction will be smaller. Instead, in the initial area of two-pair engagement, the pressure drop will be much less noticeable. The specified qualitative regularities of reduction of the initial maximum contact pressures in engagement of MP gears at both types of correction are close irrespective of a type of polymeric materials. In terms of quantification, the pressures will be about 1.29 times higher in the gears with a gear made of PA6 + 30CF than with a gear made of PA6 at different values of the shift coefficients. When the engagement is corrected, the angle of entry into the one-pair engagement and the pitch point are shifted to the left. With both types of

correction, the pressure level in the pitch point will not change.

The wear of the gear teeth of the corrected gears leads to a certain differentiated reduction of pressures p_{jmax} in the engagement. For the example of the gears with height correction trends of these changes are shown in Fig. 4. The figure shows the trends of these changes on the example of the gears with height correction.

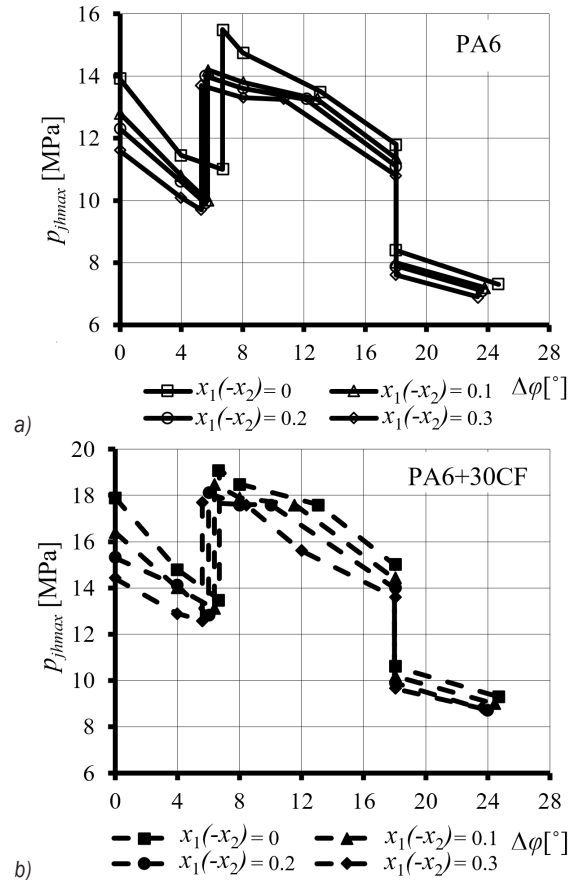


Fig. 4. Maximum wear contact pressures: a) PA6, b) PA6+30CF

The smallest decrease in p_{jmax} (up to 1.053 times) is observed in the input area of engagement and in the transition to the central region. In contrast, at the exit from the central region the ducton will be up to 1.15 times and much more up to 1.24 times at the exit from its right region. Wear-contact pressures at angular correction have similar quantitative and qualitative laws, as in case of height correction.

Correction of the engagement affects the course of linear wear h_{2j} of the gear polymer teeth in different ways at selected points of their profile. The research results are shown in Fig. 5.

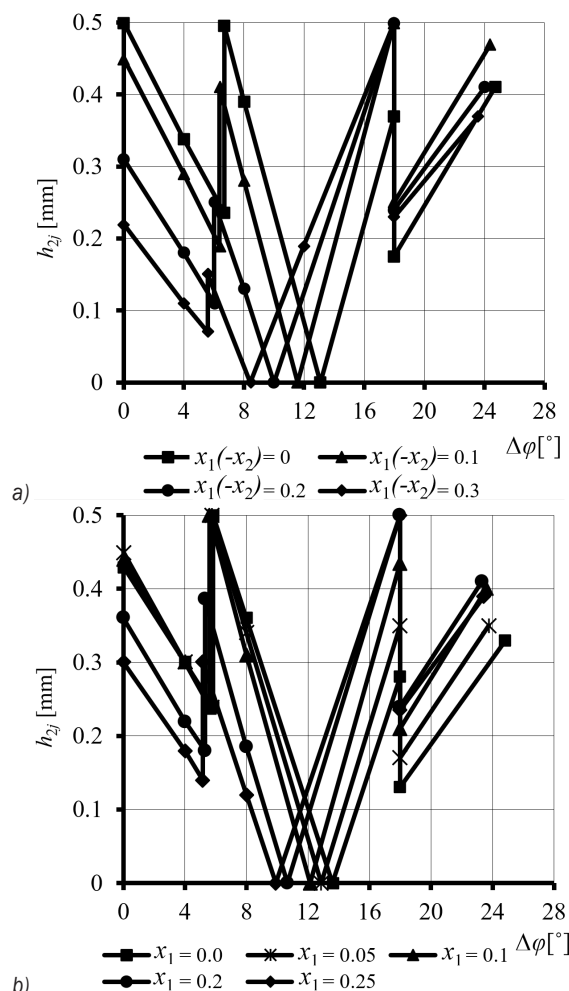


Fig. 5. Linear wear of the teeth profile of the polymer gears during engagement: a) height correction, b) angular correction

Correction of engagement leads to changes in tooth wear at characteristic points: at the entrance to the left area of two-pair engagement, at the entrance to the area of one-pair engagement, at the exit from it, and at the exit from the right area of two-pair engagement. In all cases of height correction, the acceptable wear is achieved at the exit of the teeth from the area of one-pair engagement. For angular correction, the acceptable wear is achieved at the point of entrance of the teeth into one-pair engagement at $x_1 = 0, 0.05, 0.1$, and at $x_1 = 0.2, 0.25$ the acceptable wear is achieved at the exit from this engagement. As already noted, the correction leads to a shift of the pitch point to the left, which is clearly seen in the figures because here the wear of the teeth is close to zero. This is because the slippage speed at the pitch point is almost zero.

The estimated minimum resource $t_{B\min}$ of MP gears for both types of correction and the studied gear

materials is shown in Fig. 6. Of the gears resource determined at all characteristic points of engagement, the minimum will be that where the accepted allowable wear of the gear teeth is achieved (Fig. 5).

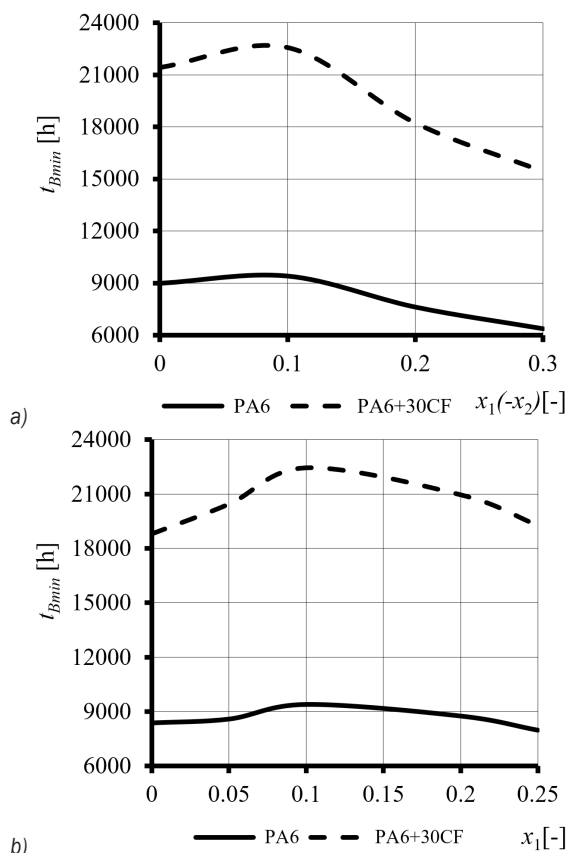


Fig. 6. Influence of the teeth correction on the minimum resource $t_{B\min}$ of MP gears at a teeth inclination: a) height correction, b) angular correction

The gears with a gear made of PA6 + 30CF will have a significantly higher resource $t_{B\min}$ than a gears with a gear made of PA6 in the whole range of shift coefficients. It is 2.4 times larger at the optimal values of the shift coefficients of height correction $x_1 = -x_2 = 0.1$ and angular correction $x_1 = 0.1, x_2 = 0.2$. Therefore, the results show that the correction of the engagement in relation to the gears resource will be most effective at these coefficients. It should be noted that with these optimal shift coefficients, the gears resource with height correction increases about 1.05 times, and with angular correction in 1.1 times (gear made of PA6) and in 1.22 times (gear made of PA6+30CF). A comparison of the effectiveness of engagement correction by both methods shows that at height correction at $x_1 = -x_2 = 0.126$ the gears resource will be lower than at $x_1 = -x_2 = 0$. That is, in the future such

correction will not make sense given the reduction of gears resource. Instead, with angular correction in the gears at $x_1=0.3$, $x_2=0$, its resource will be the same as the uncorrected gears. For the angular correction in the case of normative distribution of coefficients $x_1=0.225$, $x_2=0.075$, the gears resource will be somewhat higher, as at $x_1=0$, $x_2=0.3$, but less in 1.13 times than at the optimal values of the coefficients $x_1=0.1$, $x_2=0.2$. Presented in Fig. 6, the nature of the change in the gears resource indicates that the pre-optimal shift coefficients are better for height correction, and the post-optimal are better for angular correction.

The results of this type of analytical research realize the possible optimization of gears by three criteria:

- 1) The load carrying capacity of the gears by p_{\max} or $p_{h\max}$.

This confirms the well-known trend that the correction of engagement leads to a decrease in the level of p_{\max} , and the above also shows their change during tooth wear. Contact pressures will be lowest at the highest values of the shift coefficients.

- 2) Linear wear h_{2j} of the tooth profile

Given this criterion, larger coefficients are more rational for height correction $x_1=-x_2$ or angular correction x_1 . Then the acceptable wear will not be at the entrance to the two-pair engagement, as is the case of uncorrected gears, but at the exit of the teeth from the one-pair engagement or at the entrance to it.

- 3) Gears resource $t_{B\min}$

As established, the longest life of corrected gears will be achieved with the specified optimal shift coefficients. Since the service life of the gears is its most important characteristic from a practical point of view, the other two criteria will be subordinate. It should be noted that with the specified optimal shift coefficients, the second criterion will actually be provided as well.

3 CONCLUSIONS

1. The initial maximum contact pressures $p_{j\max}$ in the studied MP gears with height and angular correction have converged quantitative and qualitative patterns of change during the cycles of two-one-two-pair engagement. Correction of teeth has the most significant effect on their reduction in the entrance area of two-pair engagement, and less in one-pair engagement. The contact pressure in the gears with a gear

made of PA6 + 30CF will be 29 % higher than with a gear made of PA6 regardless of the shift coefficients. Due to the wear of the polymer gear teeth, there is no noticeable effect on the change of initial pressures $p_{j\max}$ in the first and second areas of engagement. Instead, such a change in pressure is more significant in the initial region of two-pair engagement. Quantitative and qualitative patterns of changes in wear and contact pressures are similar in both types of tooth correction.

2. The linear wear h_{2j} of the polymer gear teeth profiles depends not only on the specific friction force (nonlinear) and sliding speed (linear), which are variable at each point of contact, but also on the number of pairs of teeth simultaneously engaged. Therefore, it systematically changes with successive contact of the teeth during their rotation, reaching the acceptable value of h_{2*} at any of the three characteristic points: at the entrance to the two-pair engagement, at the entrance or exit of the one-pair engagement. This fact depends on the combination of the shift coefficients. In fact, for both types of correction, the maximum wear h_{2*} will be achieved at the edges of one-pair engagement
3. Along with the contact teeth strength, which must be ensured when designing uncorrected gears according to ISO 6336: 2006-10, DIN 3990: 1987-12 [1] and [2] for calculating the load capacity of gears with metal toothed wheels or also the recommendations of VDI 2736 Blatt 2:2014-06: 2014 [28] for thermoplastic gears, from a practical point of view, the most desirable tribotechnical parameter is their forecast resources. According to the developed calculation method of research of MP gears, such estimation of resources was carried out both for uncorrected, and for corrected gears. Accordingly, both quantitative and qualitative patterns of the impact of height and angular correction of engagement on the resource are established above. It is shown that there are optimal shift coefficients that provide a higher gears life compared to uncorrected gears. Also, expedient areas of shift coefficients values at height and angular correction are established. In fact, this method can be a reasonable basis for the rational choice of shift coefficients, both metal and metal-polymer gears.
4. It should be noted that it is known from practice that, as a result of the increased bending of the teeth of polymer composite gears, a positive effect arises, which consists in an increased area of two-pair engagement, that is, in an increase

in the overlap coefficient. Moreover, this should lead to some reduction of the contact pressures at the entrance to the one-pair engagement. Perhaps they will then have values close to or lower than the pressures at the entrance to the two-pair engagement (Figs. 1 and 2). Due to the reduction of the maximum contact pressures, the gears resources will increase. The method of calculating MP gears developed by us does not take into account the influence of tooth deflection on the studied contact and tribocontact parameters. However, based on it, it is possible to carry out their full-scale quantitative and qualitative analysis, which has not yet been carried out by other researchers. Its results should be interpreted as reflecting worse conditions of contact and wear-contact interaction of teeth in engagement. The deflection of the teeth in the actual state of engagement of MP gears even improves in view of the above-mentioned typical geometric condition.

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