

Myron CHERNETS

METHOD OF CALCULATION OF TRIBOTECHNICAL CHARACTERISTICS OF THE METAL-POLYMER GEAR, REINFORCED WITH GLASS FIBER, TAKING INTO ACCOUNT THE CORRECTION OF TOOTH

METODA OBLICZENIOWA TRIBOTECHNICZNYCH CHARAKTERYSTYK PRZEKŁADNI ZĘBATYCH METAL-POLIMEROWYCH Z POLIAMIDU WZMOCNIONEGO WŁÓKNEM SZKLANYM Z UWZGLĘDNIENIEM KOREKCJI ZĘBÓW

The paper proposes a new method for calculating the service life, wear and contact pressures of metal-polymer gear drives with a correction profile. The effects of height and angular modification in a gear drive made of dispersive glass fibre-reinforced polyamide and steel on its contact and tribocontact parameters are determined. A numerical solution obtained for the gear with height correction has shown that the life of such gear is the longest when the profile correction coefficients $x_1 = -x_2 = 0.1$. It has been found that the service life of the gear with angular correction is shorter than that of the gear with correction height. The effects of gear tooth height and angular correction on maximum contact pressures and pinion wear are examined and determined.

Keywords: *method for calculating service life, wear and contact pressures, metal-polymer spur gear drives, dispersive glass fibre-reinforced polyamide, height and angular correction, gear life, contact strength.*

Prezentowana opracowana nową metodę obliczeniową resursu, zużycia oraz nacisków stykowych przekładni walcowej metal – polimerowej z korekcją uzębienia. Dla przekładni z kołami zębatymi z poliamidu wzmocnionego dyspersyjnym włóknem szklanym i stali zostało przeprowadzone oszacowanie wpływu korekcji technologicznej oraz konstrukcyjnej uzębienia na wskazane parametry kontaktu oraz tribokontaktu. Na podstawie numerycznego rozwiązania zagadnienia dla przypadku korekcji technologicznej zębów kół określono, że największa trwałość przekładni będzie, gdy współczynniki korekcji $x_1 = -x_2 = 0.1$. Ustalono, że wtedy przy korekcji konstrukcyjnej zębów trwałość przekładni będzie mniejsza niżeli przy korekcji technologicznej. Został przebadany charakter wpływu korekcji technologicznej oraz konstrukcyjnej zębów na maksymalne naciski stykowe, zużycie zębniaka oraz ustalono jego prawidłowości.

Słowa kluczowe: *metoda obliczeniowa resursu, zużycia oraz nacisków stykowych, przekładnia metal – polimerowa walcowa o zębach prostych, poliamid wzmocniony dyspersyjnym włóknem szklanym, korekcja technologiczna oraz konstrukcyjna, trwałość przekładni, naciski stykowe.*

1. Introduction

Metal-polymer gears are widely used in different areas. For this reason, it is of practical importance to know how to determine their service life or wear with the use of calculation methods. However, neither the literature of the subject nor applied engineering offer such methods, particularly for gears that are made of composite polymers reinforced with glass or carbon particles or fibres to reduce their wear and extend their service life. Only in [23] a simplified calculation method for abrasive wear is applied to determine the life of a spur gear made of dispersive glass or carbon fibre-reinforced polyamide. However, it should be mentioned that abrasive wear is not the most common type of wear for such gears.

The literature of the subject offers methods for calculating the wear of spur gear drives with metal gears [2, 11, 12, 13, 15, 16, 20, 21 et al]. However, these methods are based on the Archard wear equation describing sliding wear, yet this type of wear does not occur in lubricated gear drives, not to mention gear drives with a closed case design. According to results of experimental studies on the wear of dispersive glass or carbon fibre-reinforced polyamide reported in the literature, the dominant type of wear in non-lubricated metal-polymer gear drives is fatigue wear while abrasive wear does not occur here at all. Consequently, it is necessary to devise methods for determining

service life and wear of metal-polymer gear drives that take account of the real wear mechanism.

The author and his co-authors have developed methods for calculating contact strength, wear and life of metal gears (straight and skew spur and bevel gears) [4 - 8] that are based on a tribokinetic, mathematical model of material wear caused by sliding friction and take account of their fatigue wear. The proposed calculation method was modified to suit metal-polymer gears, particularly those made of polyamide composites reinforced with dispersive glass or carbon fibres [9]. In [10] the authors report experimental results of their own studies investigating the wear resistance of reinforced polyamide composites used for metal-polymer gear drives to determine their wear resistance in order to be able to implement the proposed numerical solution. Results of both these and other studies reported in the literature point to fatigue wear of polyamide composites reinforced with dispersive glass and carbon fibres.

It should be stressed that the literature of the subject mentions only few experimental studies on polyamide composites used in the design of metal-polymer gear drives [3, 14, 17, 18, 19, 26]. For example, the results of numerical modelling of a metal-polymer straight spur gear drive with polyamide 66 gears are presented in [3, 14], where the distribution of load between the gears and its influence on contact and bending stresses is examined and then verified experimentally. The

influence of sliding speed on the frictional force in a tribojoint made of PA6 polymer and S355J2 steel is investigated by disc tests in [26], and the sliding velocities and Hertz contact pressures in the meshing of a metal-polymer spur gear are calculated. The friction coefficients and relative volumetric wear of the teeth of a metal-polymer gear drive from polyamide PA6-Mg, PA6-Na, PA66GF-30, polyoxymethylene POM-C and steel S355 under normal conditions in the air and abrasive media are experimentally investigated in the works [17, 18]. It has been established in [24] that the wear and life of glass and carbon composites largely depend on the volume content of a reinforcing phase. However, there are no other studies of this nature in the literature. More information about the contact and bending strengths of metal-polymer gears determined by well-known methods is given in [1, 23, 25, 27] dentition correction is used.

Tooth profiles in metal and metal-polymer gear drives are often corrected, which leads to reduced contact pressures and wear of the engaged teeth and increased gear life. The literature of the subject, however, contains no mention of numerical or experimental studies investigating the effect of gear tooth profile correction on the load capacity and tribotechnical parameters of metal-polymer gears. In light of the above, the author of this paper – using his own methods – has undertaken such study for a spur gear drive made of steel and dispersive glass fibre-reinforced polyamide with height and angular correction of the gear teeth, and obtained results are given below.

2. Method for solving the problem

The method for assessing the wear of metal-polymer gear drives is based on a tribokinetic mathematical model of sliding abrasive wear [4, 6] shown below. According to this model, the wear of engaged teeth is described with a system of linear differential equations:

$$\frac{1}{v} \frac{dh_k}{dt} = \Phi_k^{-1}(\tau), \quad k=1;2, \quad (1)$$

where $\tau = fp$.

Experimental values of the wear-resistance function $\Phi(\tau)$ of the materials are approximated by the relation:

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

where $\tau_S = R_{0,2} / 2$; $R_{0,2} = 0,7R_m$ (steel), $\tau_S = R_m / 2$ (filled polymer composites).

The wear-resistance function $\Phi_i(\tau_i)$ of the teeth materials is determined in the following way:

$$\Phi_i(\tau_i) = L / h_i.$$

Taking into account relation (2), after the separation of variables and system integration (1) on condition that $\tau = fp = \text{const}$, the following will arise:

$$t_k = \frac{C_k}{v} \left(\frac{\tau_S}{\tau} \right)^{m_k} h_k. \quad (3)$$

Then, the function of linear wear of the teeth at any point j of the working surface over a period t_j of their interaction:

$$h_k = \frac{v t_k}{C_k} \left(\frac{\tau}{\tau_S} \right)^{m_k}. \quad (4)$$

The linear wear of the gear teeth h'_{kj} at any point j of the profile in the tooth engagement time t'_j is determined using the following formula [4]:

$$h'_{kj} = \frac{v_j t'_j (f p_{j \max})^{m_k}}{C_k \tau_S^{m_k}} \quad (5)$$

where $j = 0,1,2,3,\dots, s$, $t'_j = 2b_j / v_0$, $v_0 = \omega_1 r_1 \sin \alpha_t$.

Tooth wear causes an increase in the curvature radii of tooth profiles, which leads to a decrease in the initial maximum contact pressures $p_{j \max}$, and the contact area width $2b_j$ at every j -th point of contact are calculated in accordance with the Hertz equations:

$$p_{j \max} = 0.564 \sqrt{N' \theta / \rho_j}, \quad 2b_j = 2.256 \sqrt{\theta N' \rho_j}, \quad (6)$$

where $N' = N / l_{\min} w$; $N = 9550P / r_1 n_1 \cos \alpha_t$; respectively, for the spur gear with the pinion width $l_{\min} = b_W$;
 $\theta = (1 - \mu_1^2) / E_1 + (1 - \mu_2^2) / E_2$.

The reduced radius of curvature of the involute spur gear is:

$$\rho_j = \frac{\rho_{1j} \rho_{2j}}{\rho_{1j} + \rho_{2j}}. \quad (7)$$

The formulas for calculating the radii of curvature for the modified pinion and gear profiles of the spur gear at j -th point of contact are [5]:

$$\rho_{1j} = \frac{\rho_{t1j}}{\cos \beta_b}, \quad \rho_{2j} = \frac{\rho_{t2j}}{\cos \beta_b}, \quad (8)$$

where for the bevel gear:

$$\begin{aligned} \beta_b &= \arctan(\tan \beta \cos \alpha_t), \quad \alpha_t = \arctan\left(\frac{\tan \alpha}{\cos \beta}\right), \\ \rho_{t1j} &= r_{b1} \tan \alpha_{t1j}, \quad \rho_{t2j} = r_2 \sqrt{(r_2 / r_2)^2 - \cos^2 \alpha_t}, \\ \alpha_{t1j} &= \arctan(\tan \alpha_{t10} + j \Delta \varphi), \quad \alpha_{t1s} = \arctan\sqrt{(r_{1s} / r_1)^2 - \cos^2 \alpha_t}, \\ \alpha_{t2j} &= \arccos\left[(r_2 / r_2) \cos \alpha_t\right], \\ r_{b1} &= r_1 \cos \alpha_t, \quad r_1 = m z_1 / 2 \cos \beta, \quad r_{b2} = r_2 \cos \alpha_t, \quad r_2 = m z_2 / 2 \cos \beta, \\ \tan \alpha_{t10} &= (1 + u) \tan \alpha_t - \frac{u}{\cos \alpha_t} \sqrt{(r_{20} / r_2)^2 - \cos^2 \alpha_t}; \quad r_{a2} = r_2 + m, \\ r_{20} &= r_{a2} - r, \quad r = 0.2m, \\ r_{2j} &= \sqrt{a_W^2 + r_{1j}^2 - 2a_W r_{1j} \cos(\alpha_t - \alpha_{t1j})}, \quad r_{1j} = r_1 \cos \alpha_t / \cos \alpha_{t1j}, \\ a_W &= (z_1 + z_2) m / 2 \cos \beta. \end{aligned}$$

The minimum length of the line of contact is:

$$l_{\min} = \frac{b_W \varepsilon_\alpha}{\cos \beta_b} \left[1 - \frac{(1-n_\alpha)(1-n_\beta)}{\varepsilon_\alpha \varepsilon_\beta} \right] \text{ at } n_\alpha + n_\beta > 1, \\ l_{\min} = \frac{b_W \varepsilon_\alpha}{\cos \beta_b} \left[1 - \frac{n_\alpha n_\beta}{\varepsilon_\alpha \varepsilon_\beta} \right] \text{ at } n_\alpha + n_\beta \leq 1, \quad (9)$$

where: $\varepsilon_\alpha = \frac{t_1 + t_2}{t_z}$, $\varepsilon_\beta = \frac{b_W \sin \beta}{\pi m}$, $\varepsilon_\gamma = \varepsilon_\alpha + \varepsilon_\beta$, $t_1 = \frac{e_1}{\omega_1 r_{b1}}$, $t_2 = \frac{e_2}{\omega_1 r_{b1}}$, $t_z = \frac{2\pi}{z_1 \omega_1}$, $e_1 = \sqrt{r_{1s}^2 - r_{b1}^2} - r_1 \sin \alpha_t$, $e_2 = \sqrt{r_{20}^2 - r_{b2}^2} - r_2 \sin \alpha_t$, $r_{1s} = r_{a1} - r$, $r_{a1} = r_1 + m$.

In Fig. 1 presents spur gear engagement and transmission parameters. The sliding velocity of the engaged teeth is calculated as [5]:

$$v_j = \omega_1 r_{b1} (\tan \alpha_{t1j} - \tan \alpha_{t2j}). \quad (10)$$

In a simplified case, at constant output conditions, i.e., when the initial contact pressures $p_{j\max} = \text{const}$, the gear life t_* for a given acceptable tooth wear h_{k*} is calculated as:

$$t_* = h_{k*} / \bar{h}_{kj}, \quad (11)$$

where $\bar{h}_{kj} = 60 n_k h'_{kj}$.

In some of the above formulas one should consider the modified engagement parameters.

For the gears with height correction:

The addendum radii of the gear:

$$r_{a1} = r_1 + (1 + x_1)m, \quad r_{a2} = r_2 + (1 + x_2)m, \quad (12)$$

where $x_1 = -x_2$.

The remaining parameters of the gear are the same as those of the gear without profile correction.

For the gears with angular correction:

Here $x_1 \neq x_2$, and the total profile shift coefficient $x_\Sigma = x_1 + x_2$. The working (real) distance between the axes is:

$$a_{wk} = r_{w1} + r_{w2} > a_w. \quad (13)$$

The corrected profile pressure angle α_w depends on the real distance between the axes of the meshing gears and is higher (when $a_{wk} > a_w$) than the apparent pressure angle α_t . If the real distance between the axes is known, then:

$$\alpha_w = \arccos \frac{a_w}{a_{wk}} \cos \alpha_t. \quad (14)$$

The pitch radii of the pinion and gear teeth:

$$r_{w1} = r_1 \frac{\cos \alpha_t}{\cos \alpha_w}, \quad r_{w2} = r_2 \frac{\cos \alpha_t}{\cos \alpha_w}. \quad (15)$$

The addendum radii of the gear teeth:

$$r_{a1} = r_1 + (1 + x_1 - K)m, \quad r_{a2} = r_2 + (1 + x_2 - K)m, \quad K = \frac{a_w - a_{wk}}{m} + x_\Sigma. \quad (16)$$

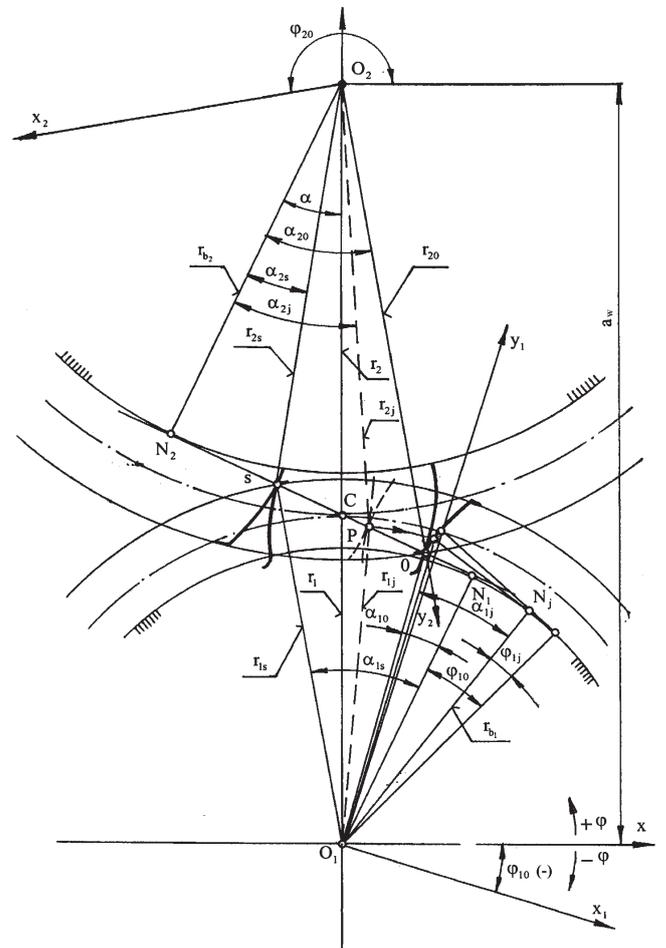


Fig. 1. Parameters of gear and engagement: N_1N_2 - line of engagement; $0, s$ - respectively, the points of entry of the teeth in the engagement teeth and exit; C - engagement mean

Formulas in which the above profile correction parameters should be taken into account are as follows:

$$N = 9550PK_g / r_{w1} n_1 \cos \alpha_w, \\ \text{tg} \alpha_{t10} = (1 + u) \text{tg} \alpha_w - \frac{u}{\cos \alpha_w} \sqrt{(r_{20} / r_{w2})^2 - \cos^2 \alpha_w}, \\ \alpha_{t1s} = \arctg \sqrt{(r_{1s} / r_{w1})^2 - \cos^2 \alpha_w}, \quad r_{t2j} = r_{w2} \sqrt{(r_{2j} / r_{w2})^2 - \cos^2 \alpha_w}, \\ r_{2j} = \sqrt{a_w^2 + r_{1j}^2 - 2a_w r_{1j} \cos(\alpha_w - \alpha_{t1j})}, \quad r_{1j} = r_{w1} \cos \alpha_w / \cos \alpha_{t1j}, \\ \text{tg} \alpha_{t2s} = (1 + u^{-1}) \text{tg} \alpha_w - \frac{1}{u \cos \alpha_w} \sqrt{(r_{1s} / r_{w1})^2 - \cos^2 \alpha_w}, \\ \alpha_{t2j} = \arccos \left[(r_{w2} / r_{2j}) \cos \alpha_w \right], \quad e_1 = \sqrt{r_{1s}^2 - r_{b1}^2} - r_{w1} \sin \alpha_w, \\ e_2 = \sqrt{r_{20}^2 - r_{b2}^2} - r_{w2} \sin \alpha_w.$$

The angles of transition from double-pair engagement ($\Delta\phi_{1F_2}$) to single-pair engagement and, again, to double-pair engagement ($\Delta\phi_{1F_1}$) in the spur gear with a corrected profile are determined in the following way [6, 7]:

$$\Delta\phi_{1F_2} = \phi_{10} - \phi_{1F_2}, \Delta\phi_{1F_1} = \phi_{10} + \phi_{1F_1}; \quad (17)$$

where $\phi_{1F_2} = \tan\alpha_{F_2} - \tan\alpha_t, \phi_{1F_1} = \tan\alpha_{F_1} - \tan\alpha_t, \phi_{10} = \tan\alpha_{t10} - \tan\alpha_w$;

$$\tan\alpha_{F_2} = \frac{r_1 \sin\alpha_t - (p_b - e_1) + 0.5n_\beta p_b}{r_1 \cos\alpha}, \tan\alpha_{F_1} = \frac{r_1 \sin\alpha_t - (p_b - e_2) - 0.5n_\beta p_b}{r_1 \cos\alpha} \quad (18)$$

In the spur gear $e_\beta, n_\beta = 0$ and $l_{\min} = b_W, p_b = \pi m \cos\alpha_w$.

The angle $\Delta\phi_{1E}$ describing the moment of pair engagement exit is:

$$\Delta\phi_{1E} = \phi_{10} + \phi_{1E}, \quad (19)$$

where $\phi_{1E} = \tan\alpha_E - \tan\alpha_t, \alpha_E = \arccos(r_{b1} / r_{1s}); p_b = \pi m \cos\alpha_t / \cos\beta$.

For triple-double-triple pair engagement (helical gears):

$$\tan\alpha_{F_2} = \frac{r_1 \sin\alpha_t - (p_b - e_1) + 0.5\tilde{n}_\beta p_b}{r_1 \cos\alpha}, \tan\alpha_{F_1} = \frac{r_1 \sin\alpha_t - (p_b - e_2) - 0.5\tilde{n}_\beta p_b}{r_1 \cos\alpha} \quad (20)$$

$$\tilde{n}_\beta = \left\{ \begin{array}{l} n_\beta \text{ at } n_\alpha + n_\beta > 1, \\ 1 - n_\beta \text{ at } n_\alpha + n_\beta \leq 1 \end{array} \right\}$$

3. Numerical solution

The calculations were performed for a metal-polymer spur gear drive with a glass fibre-reinforced polyamide pinion and a steel gear after the application of height and angular correction of the gear teeth. Initial data applied in the calculations were as follows: $T_{nom} = 4000$ Nmm, $n_1 = 1000$ rpm; $K_g = 1.2; \beta = 0; m = 4$ mm, $u = 3, z_1 = 20, z_2 = 60, b_W = 50$ mm, $f = 0.3; h_{k*} = 0.5$ mm.

The applied profile correction coefficients were: a) height correction: $x_1 = -x_2 = 0; 0.1; 0.2; 0.3; 0.4; a_W = 160$ mm; b) angular correction: $x_1 = 0, x_2 > 0; x_1 > 0, x_2 = 0; x_1 = 0, x_2 < 0; a_{Wk} = 161$ mm.

The gear materials:

- 1) steel S45 in the state of delivery, $E = 2.1 \cdot 10^5$ MPa, $\mu = 0.3; C = 10^9, m = 2$ [10];
- 2) glass fibre-reinforced polyamide composite (30% vol.) PA6-LT-GF30-1, $E_G = 3,90$ GPa, $\mu_G = 0.42; C_G = 1.2 \cdot 10^6, m_G = 1.9$ [10]; $\tau_S^{(G)} = 52$ MPa.

In the above wear resistance characteristics of the gear materials, particularly of the polyamide composite with glass fibres PA6-LT-SW30-1 [10] for the unit friction force $\tau = fp$, the characteristic $m_G = 1.9$ indicates a nearly quadratic dependence between the experimental wear function (Eq. (2)) and the contact pressures $p = \tau / f$. This means that we are dealing here with fatigue wear rather than abrasive wear, because in the latter case $m_G = 1$. The wear of the steel gear is three times lower than that of the polyamide gear.

The results are given in Figs. 2 - 7. Fig. 1 illustrates the relationship between the minimal gear life t_{\min} at the contact point on the gear tooth profile where the maximum allowable wear occurs faster.

The results demonstrate that the application of height correction when $x_1 = -x_2 = 0.1$ increases the gear life t_{\min} by 1.2 times com-

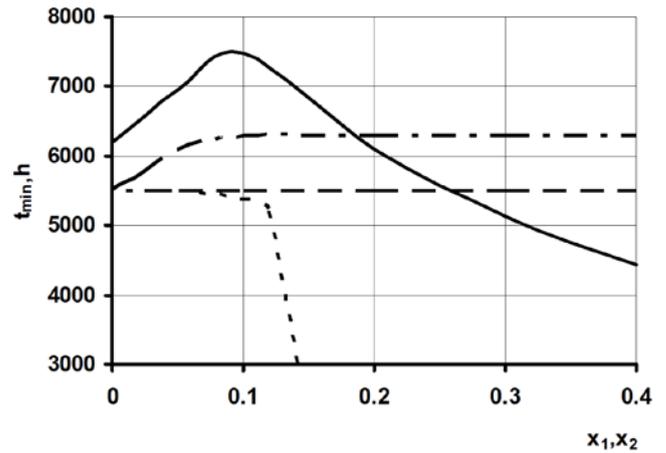


Fig. 2. Minimal life t_{\min} of a gear with profile engagement: solid line: $x_1 = -x_2 = 0 \dots 0.4$; dashed-dotted line: $x_1 = 0 \dots 0.4; x_2 = 0$; dashed line: $x_1 = 0, x_2 = 0 \dots 0.4$; dotted line: $x_1 = 0, x_2 = 0 \dots -0.1462$

pared to the life of the gear without profile correction. On the other hand, when $x_1 = -x_2 > 0.18$, the gear life decreases compared to both the gear without profile modification and the above-mentioned gear with angular correction (dashed-dotted line).

Among the analysed cases of a gear drive with angular correction where $a_{wk} = 161$ mm, the life of the gear drive somewhat increases when $x_1 = 0 \dots 0.1; x_2 = 0$. Nevertheless, the gear life is lower by 1.19 times than that of the gear with height correction. As regards two other cases of the gear with angular correction, the application of angular correction either has no effect on gear life ($x_1 = 0, x_2 = 0 \dots 0.4$) or leads to its sudden decrease ($x_1 = 0, x_2 = 0 \dots -0.1462$).

The minimal life t_{\min} of the gear drive with a steel pinion and a polyamide composite gear was determined. It has been found that the minimal life of the gear increases in direct proportion to the gear ratio u , i.e., by three times in this particular case.

An important parameter describing the meshing conditions is the overlap factor ϵ_α . Fig. 3 illustrates the effect of different types of profile correction on the overlap factor in double-single-double pair engagement.

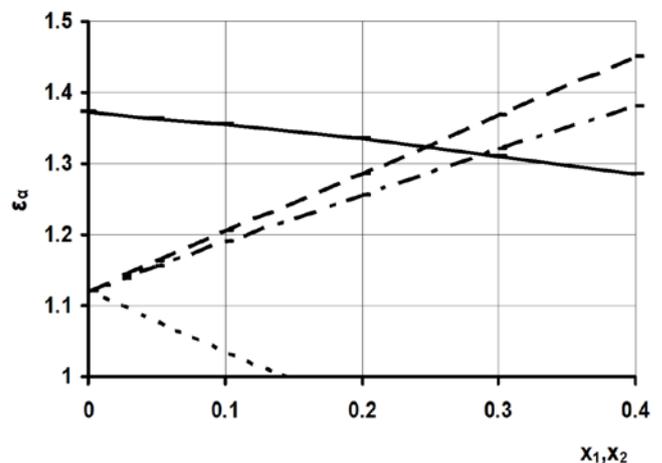


Fig. 3. Overlap factor vs. double-single-double pair engagement

The overlap factor ϵ_α of the gear with height correction for the optimal modification coefficients $x_1 = -x_2 = 0.1$ is definitely higher than in the case of the gear with angular correction when $x_1 = 0.1; x_2 = 0$

or $x_1 = 0$; $x_2 = 0.1$, which has a positive effect on service life of the gear drive (Fig. 2).

Fig. 4. illustrates variations in the maximum contact pressures for the meshing gears when height correction is applied as the optimal type of profile modification. Their values decrease between the entry and exit of pair engagement due to an increase in the reduced radius of curvature and the changes in gear pair engagement (double–single–double pair engagement).

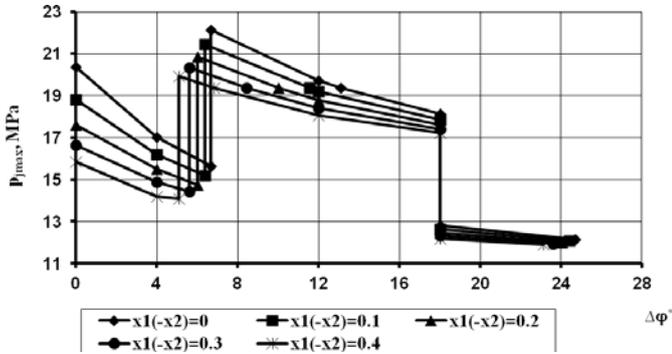


Fig. 4. Effect of height correction on maximum contact pressures in a gear meshing cycle

The results reveal a significant influence of height correction on the value of p_{jmax} in a gear meshing cycle. The greatest changes can be observed at the entry of double-pair engagement when $\Delta\phi = 0$ (left side of the figure) and at the entry of single-pair engagement in the central region. The highest contact pressures occurring at the entry of single-pair engagement are higher than in the centre of the meshing gears (19.36 MPa – all markers are on the same level). Therefore, it can be observed that the application of profile correction leads to decreasing the level of p_{max} in the entire tested range of addendum correction coefficients $x_1 = -x_2 > 0$, whereas at $x_1 = -x_2 \approx 0.2$ the life of the modified gear drive is lower than that of the gear without profile correction (Fig. 2). The selection of profile correction coefficients should depend on contact strength of the gear teeth or gear life, or both.

The investigation of the effect of angular correction when $x_1 = 0 \dots 0.4$; $x_2 = 0$ demonstrates that the gear life increases while the contact pressures p_{jmax} decrease. Results of these calculations are given in Fig. 5.

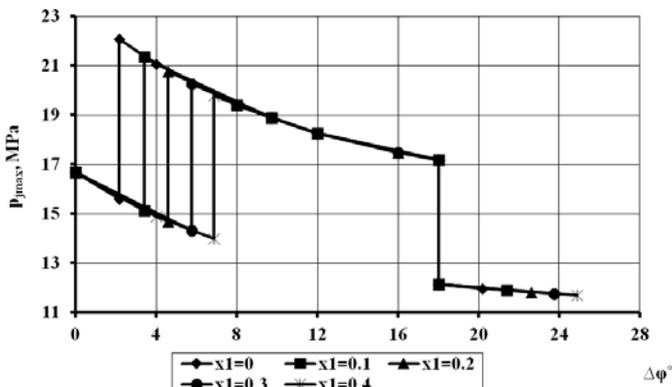


Fig. 5. Effect of angular correction on maximum contact pressures in a gear meshing cycle when $x_1 = 0 \dots 0.4$; $x_2 = 0$

The correction of the pinion teeth only affects the maximum contact pressures p_{jmax} at the exit of double-pair engagement and at the entry of single-pair engagement. Contrary to height correction, increasing the coefficient x_1 leads to increasing the double-pair engage-

ment area, which results in a slight reduction in the contact pressures at this point without any changes in the service life of the gear drive (Fig. 2), which remains unchanged starting from $x_1 \approx 0.1$. Hence, this type of angular correction can be applied to reduce maximum contact pressure when the service life of the gear with height correction of $x_1 = -x_2 > 0.18$ decreases (Fig. 2).

In effect of the application of angular correction when $x_1 = 0, x_2 = 0 \dots 0.4$, both the highest contact pressures occurring at the entry of single-pair engagement and the gear life remain unchanged (Fig. 6). As in the previous case angular correction increasing the coefficient x_2 leads to increasing the double-pair engagement area, but this does not increase the durability of the transmission. Instead, the negative consequence of the correction of the teeth of the metal wheel is the increase in contact pressures at the entrance to the double-pair engagement. Therefore, this case of angular correction engagement is not appropriate.

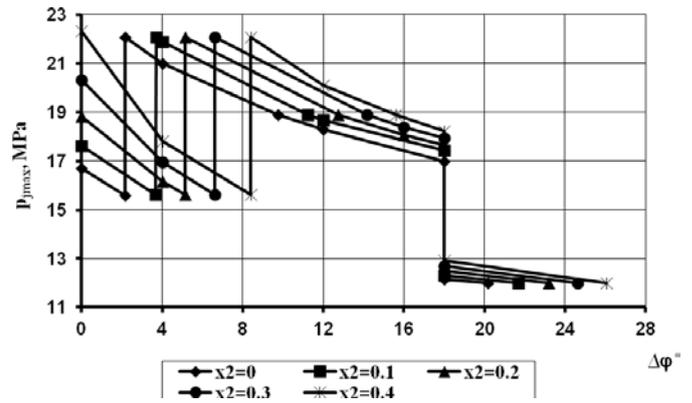


Fig. 6. Effect of angular correction on maximum contact pressures in a gear meshing cycle when $x_1 = 0, x_2 = 0 \dots 0.4$

Fig. 7. shows the linear wear of the pinion teeth when height correction is applied.

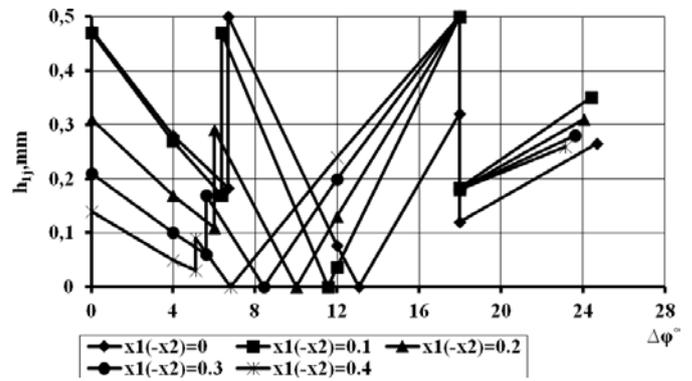


Fig. 7. Linear wear of the pinion teeth in a gear meshing cycle

The results of wear obtained for the non-corrected gear drive and the modified gear with the correction coefficients $x_1 = -x_2 = 0.1$ are similar. The latter case is optimal, and the wear is almost identical at three points of the tooth profile: at the entry of both double- and single-pair engagement as well as at the exit of single-pair engagement (acceptable wear), which undoubtedly leads to a longer service life of the gear drive (Fig. 2). In other cases, the wear at these three particular points varies, while the acceptable wear is reached at the exit of single-pair engagement.

Given the results of maximum contact pressures, service life and wear of metal-polymer gear drives obtained with the developed cal-

ulation method, it is recommended to apply height correction with relatively small modification coefficients ($x_1 = -x_2 \approx 0.15$) in order to increase the load carrying capacity and service life of a gear drive.

5. Conclusions

1. The proposed calculation method is employed to thoroughly investigate metal-polymer gear drives with gears made of dispersive glass fibre-reinforced polyamide composite in order to assess their strength properties and tribotechnical parameters as well as to verify whether profile correction is necessary, which is of great practical importance at the design stage.
2. The results of problem solution demonstrate that the application of height correction in a limited range of variation of addendum correction coefficients leads to an increase in the gear life compared to the gear without profile correction (Fig. 2).
3. When $x_1 = -x_2 = 0.1$, the gear life increases by 12% and is the highest possible for the gear with angular correction.
4. The maximum contact pressures in the engagement cycle vary significantly, depending on the engagement area and the pin-

ion rotation angle. They reach the greatest value at the entry of single-pair engagement (Fig. 4 – height correction, Fig. 5 – angular correction).

5. In the case of the gear drive with height correction when $x_1 = -x_2 = 0.1$, the wear of the pinion teeth is almost identical at three points on the tooth profile – at the entry of double- and single-pair engagement as well as at the exit of single-pair engagement (Fig. 7). In other cases, when $x_1 = -x_2 > 0.1$, the wear at these three points of engagement varies and the acceptable wear is reached at the exit of single-pair engagement.
6. It is rational to apply height correction in compliance with the criteria of gear life and contact pressures when $x_1 = -x_2 < 0.18$, while angular correction should be applied according to these criteria when $x_1 > 0.18$, $x_2 = 0$.
7. In metal-polymer gears, heat can occur due to the action of frictional forces in the engagement. In particular, such a phenomenon is intensified with an increase in power or load. This can increase the wear of the wheel teeth. Therefore, this should be taken into account when operating this kind of gear.

References

1. Bharat G, Abhishek C, and Gautam V V. Contact stress analysis of spur gear. *International Journal of Engineering Research and Technology* 2012; 1: 1-7.
2. Brauer J, Andersson S. Simulation of wear in gears with flank interference - a mixed FE and analytical approach. *Wear* 2003; 254: 1216-1232, [https://doi.org/10.1016/S0043-1648\(03\)00338-7](https://doi.org/10.1016/S0043-1648(03)00338-7).
3. Cathelin J, Letzelter E, Guingand M, De Vaujany J P, Chazeau L. Experimental and Numerical Study a Loaded Cylindrical PA66 Gear. *Journal of Mechanical Design* 2013; 135: 89-98, <https://doi.org/10.1115/1.4023634>.
4. Chernets M V, Kelbinski J, Jarema R Ja. Generalized method for the evaluation of cylindrical involute gears. *Materials Science* 2011; 1: 45-51, <https://doi.org/10.1007/s11003-011-9366-9>.
5. Chernets M., Yarema R Y, Chernets J M. A method for the evaluation of the influence of correction and wear of the teeth of a cylindrical gear on its durability and strength. Part 1. Service life and wear. *Materials Science* 2012; 3: 289-300, <https://doi.org/10.1007/s11003-012-9505-y>.
6. Chernets M V, Chernets J M. Evaluation of the strength, wear, and durability of a corrected cylindrical involute gearing, with due regard for the engagement conditions. *Journal of Friction and Wear* 2016; 37 (1): 71-77, <https://doi.org/10.3103/S1068366616010050>.
7. Chernets M V, Chernets Y M. A technique for calculating tribotechnical characteristics of tractive cylindrical gear of VL - 10 locomotive. *Journal of Friction and Wear* 2017; 37 (6): 566-572, <https://doi.org/10.3103/S1068366616060040>.
8. Chernets M, Chernets J. The simulation of influence of engagement conditions and technological teeth correction on contact strength, wear and durability of cylindrical spur gear of electric locomotive. *Proc. J. Mech. E. Part J: Journal of Engineering Tribology* 2017; 231 (1): 57-62, <https://doi.org/10.1177/1350650116645024>.
9. Chernets M, Shil'ko S, Pashechko M. Study of wear resistance of reinforced polyamide composites for metal-polymer gear drives. *Tribologia* 2018; 3: 19 - 23, <https://doi.org/10.5604/01.3001.0012.7003>.
10. Chernets M V, Shil'ko S V, Pashechko M I, and Barshch M. Wear resistance of glass- and carbon-filled polyamide composites for metal-polymer gears. *Journal of Friction and Wear* 2018; 39 (5): 361 - 364, <https://doi.org/10.3103/S1068366618050069>.
11. Drozdov Y. To the development of calculation methods on friction wear and modeling. *Wear resistance*. Science, Moscow: Science, 1975: 120-135.
12. Flodin A, Andersson S. Wear simulation of spur gears. *Tribotest J.* 1999; 3(5): 225-250, <https://doi.org/10.1002/tt.3020050303>.
13. Flodin A, Andersson S. A simplified model for wear prediction in helical gears. *Wear* 2001; 249 (3-4): 285-292, [https://doi.org/10.1016/S0043-1648\(01\)00556-7](https://doi.org/10.1016/S0043-1648(01)00556-7).
14. Hooke C J, Kukureka S N, Liao P, Rao M, Chen Y K. The Friction and Wear of Polymers in Non-Conformal Contacts. *Wear* 1996; 200: 83-94, [https://doi.org/10.1016/S0043-1648\(96\)07270-5](https://doi.org/10.1016/S0043-1648(96)07270-5).
15. Grib V. Solution of tribotechnical tasks with numerous methods. Moscow: Science, 1982.
16. Kahraman A, Bajpai P, Anderson N E. Influence of tooth profile deviations on helical gear wear. *J. Mech. Des.* 2005; 127 (4): 656-663, <https://doi.org/10.1115/1.1899688>
17. Kalacska G, et al. Friction and Wear of Engineering Polymer Gears. *Proceedings of WTC2005 World Tribology Congress III*, Sept. 12-16, 2005, Washington, <https://doi.org/10.1115/WTC2005-63961>.
18. Keresztes R, Kalacska G. Friction of Polymer/Steel Gear Pairs. *Plastics and Rubber* 2008; 45: 236-242.
19. Kindrachuk M V, Volchenko A I, Volchenko D A, Zhuravlev D Y, Chufus V M. Elektrodynamics of the Termal Contact Friction Interaction in Metal-Polymer Friction Couples. *Material Science* 2018; 54 (1): 69 - 77, <https://doi.org/10.1007/s11003-018-0159-2>.
20. Kolivand M, Kahraman A. An ease-off based method for loaded tooth contact analysis of hypoid gears having local and global surface deviations. *J. Mech. Des.* 2010; 132 (7): 0710041-0710048, <https://doi.org/10.1115/1.4001722>.
21. Pasta A, Mariotti Virzi G. Finite element method analysis of a spur gear with a corrected profile. *J. Strain Analysis* 2007; 42: 281-292, <https://doi.org/10.1243/03093247JSA284>.
22. Pronikov A. Reliability of machines. Moscow: Mashinostroenie, 1978.

23. Sajad H D, Vivek A, Mohammad J K, Arunish M. Investigation of bending stress on a spur gear tooth at design stage by finite element modelling. International Journal on Mechanical Engineering and Robotics 2015; 3: 13-18.
24. Shil'ko S V, Starzhinskii V E. Prediction of Wear Resistance of Gearing with Wheels Made of Reinforced Composites. Journal of Friction and Wear 1993; 14 (3): 7-13.
25. Shil'ko S V, Starzhinsky V E, Petrokovets E M, Chernous D A. Two-Level Calculation Method for Tribojoints Made of Disperse-Reinforced Composites: Part 1. Journal of Friction and Wear 2013; 34 (1): 65-69, <https://doi.org/10.3103/S1068366613010133>.
26. Sukumaran J, et al. Modelling Gear Contact with Twin-Disc Setup. Tribology International 2012; 49: 1-7, <https://doi.org/10.1016/j.triboint.2011.12.007>.
27. Sushil Kumar T, Upendra Kumar J. Stress analysis of mating involute spur gear teeth. International Journal of Engineering Research and Technology 2012; 1: 1-12.

Notations

- a_w – centre distance of the gear pair
- a_{wk} – real distance between the axes
- b_w – width of the pinion
- C_k, m_k – wear resistance characteristics of the gear materials
- E, μ – Young's modulus and the Poisson's ratio of the gear tooth materials, respectively
- f – sliding friction factor
- h – linear wear of the material of the tribosystem elements
- h_i – linear wear of the material samples
- \bar{h}_{kj} – linear wear of the gear teeth per hour
- h'_{kj} – single linear wear of the teeth at any j -th point of the profile
- h_{k*} – acceptable wear of the composite gear
- $i = 1, 2, 3, \dots$ – loading ratios
- j – contact points on the active face of the teeth
- $j = 0, j = s$ – first and last point of tooth engagement, respectively
- K_g – dynamic factor
- $k = 1; 2$ – numbers of the gears (1 – pinion, 2 – gear)
- K – addendum modification coefficient
- L – friction length
- l_{\min} – minimum length of the contact line
- m – engagement module
- $n = n_k = 1, 2, 3, \dots$ – number of gear revolutions
- n_1 – number of pinion revolution
- n_α, n_β – fractional parts of the coefficients $\varepsilon_\alpha, \varepsilon_\beta$
- N – force acting in the engagement
- p – maximum tribocontact pressure
- $p_{j\max}$ – maximum tribocontact pressure (at tooth wear) at j -th point of contact
- p_b – pitch of the teeth
- P – power on the drive shaft (pinion)
- r – radius of the gear tooth fillet
- r_1, r_2 – pitch circle radii of the pinion and gear, respectively
- r_{b1}, r_{b2} – base circle radii of the pinion and gear, respectively
- r_{a1}, r_{a2} – addendum circle radii of the pinion and gear, respectively
- r_{w1} – rolling radius of the pinion
- $R_{0,2}$ – conventional yield strength of the material in tension
- R_m – immediate tensile strength of the material
- t – time of wear
- t'_j – time of wear of the teeth in the course of motion of j -th point of their contact along the tooth by the width of contact zone $2b_j$
- $t_{*\min}$ – minimal gear life $t_{*\min}$ for the highest tooth wear \bar{h}_{kj}
- T_{nom} – rated torque
- u – gear ratio
- $v_j = v$ – sliding velocity at j -th point of the tooth profile
- v_0 – velocity of the contact point along the tooth profile
- w – number of the engaged tooth pairs
- x_1, x_2 – addendum modification coefficients
- z_1, z_2 – number of teeth of the pinion and the gear, respectively
- α_t – apparent pressure angle
- α_w – pressure angle of the modified profile
- $\alpha = 20^\circ$ – pressure angle of the engaged teeth
- α_{t10} – angle of the first point on the contact line
- α_{t1s} – angle describing the location of the last point of engagement of the pinion teeth on the contact line
- $\alpha_{t20}, \alpha_{t2s}$ – angles describing the location of the first and last points of engagement of the gear teeth on the contact line
- β – tooth pitch angle
- $\varepsilon_\alpha, \varepsilon_\beta$ – coefficients describing the top and step-by-step overlap of the gear
- $\Phi(\tau)$ – function of wear resistance of the gear drive materials
- $\Delta\varphi$ – selected angle of rotation of the teeth of the pinion from the point of initial contact (point 0) to point 1, and so on
- ρ_j – reduced radius of curvature of the gear profile changeable due to wear, in a normal section
- ρ_{1j}, ρ_j – changeable curvature radii of the pinion and gear tooth profiles, respectively
- ρ_{1j}, ρ_{2j} – curvature radii of the unworn tooth flank profiles of the pinion and the gear, respectively
- τ – specific friction force
- τ_s – shear strength of the material
- ω_1 – angular velocity of the pinion

Myron CHERNETS

Department of Mechanical Engineering
Lublin University of Technology
ul. Nadbystrzycka 36, 20-816 Lublin, Poland

E-mail: m.czerniec@pollub.pl
