Myron CHERNETS[*], Jerzy KIEŁBIŃSKI[*], Jurij CHERNETS[**]

A STUDY ON THE IMPACT OF TEETH MESHING CONDITIONS AND PROFILE CORRECTION ON THE CARRYING CAPACITY, WEAR, AND LIFE OF A CYLINDRICAL GEAR

WPŁYW WARUNKÓW INTERAKCJI ORAZ KOREKCJI UZĘBIENIA PRZEKŁADNI WALCOWEJ NA NOŚNOŚĆ, ZUŻYCIE ORAZ TRWAŁOŚĆ

Key words:

cylindrical helical gear, double tooth engagement, single tooth engagement, height and angular correction of gear profile, maximum contact pressures, tooth wear, gear life

Słowa kluczowe:

przekładnia walcowa o zębach skośnych**,** zazębienie dwuparowe, zazębienie jednoparowe, korekcja technologiczna oraz konstrukcyjna uzębienia, maksymalne naciski stykowe, zużycie zębów, trwałość przekładni

[*] Lublin University of Technology, Mechanical Engineering Faculty, Institute of Technological Systems of Information, ul. Nadbystrzycka 36, 20-618 Lublin, Poland, phone: (81) 538-42-76, e-mail:m.czerniec@pollub.pl

[**] Ivan Franko State Pedagogical University in Drohobych, Ivan Franko 24, 82100 Drohobych, Ukraine.

Abstract

Using an innovative method for determining wear and life of toothed gears, the author investigates the effect of teeth meshing in a mixed (double-single-double) tooth engagement of cylindrical helical gears on their contact strength, wear, and service life. The paper also presents a method for determining zones of double and single tooth engagement. It is found that gear profile correction leads to a reduction in maximum contact pressures by about 15–20%, depending on the applied type of correction. The distribution of pressures greatly depends on the conditions of tooth engagement. The wear of teeth is also significantly affected by their meshing conditions. Depending on the type of correction applied and its coefficients, the allowable limit of gear tooth wear will occur at different points of contact: at the beginning of double tooth engagement and at the beginning or end of single-tooth engagement. It is shown that the service life of gears can be considerably prolonged if optimum correction coefficients are applied. Specifically, height correction increases the gear life by 1.66 times while angular correction prolongs the service life of gears by up to two times.

INTRODUCTION

Toothed gears of various kinds are widely applied in different fields of human activity. Cylindrical involute gears are among the most widely used types of toothed gears. In order to improve the contact and bending strength of gear teeth as well as to prolong their life, gear tooth profiles are often modified.

In the literature, one can find very few publications on the effect of profile correction (i.e. height and angular correction) on contact and bending stresses. This problem is partly discussed in the works **[L. 1–5]**. The authors of **[L. 1]** use the standardized method ISO 6636-2 **[L. 2]** to investigate the effect of height correction of a spur cylindrical involute gear profile on the above two types of stress. This method was also employed by the authors of **[L. 3]**, who thereby demonstrated that profile correction considerably improves the strength of helical cylindrical gears.

In the studies **[L. 4, 5]**, the authors investigate the effect of angular correction of a power shift gear on contact and bending stresses. The present author has also investigated in **[L. 17, 18, 20]** the effect of both height and angular gear correction on the contact pressures in cylindrical gears. The results demonstrate that the profile correction decreases the maximum contact pressures by up to 20%, depending on the applied type of profile correction.

The methods cited in the literature **[L. 6–14 et al.]**, including the papers by the present author **[L. 15, 16 et al.]**, enable us to investigate the kinematics of wear in corrected gears, without taking account of the changes in teeth meshing conditions. Although the authors of the paper **[L. 11]** discuss the potential of profile correction, they do not conduct any studies in this respect.

The present author and his colleagues **[L. 17, 19, 21 et al.]** developed a method for numerical assessment of the effect of profile correction on the life of gears in single and double tooth engagement.

In what follows, we present the results of an investigation on the effect of height and angular correction of a cylindrical gear profile on the contact pressures, wear, and the service life of the gear in mixed (double-single-double) tooth engagement.

METHOD FOR SOLVING THE PROBLEM

The linear wear of gear teeth, h'_{kj}, at any point j of the profile in the tooth engagement time, t'_j, is determined using the following formula **[L. 15]**:

$$h'_{kj} = \frac{v_j t'_j (f p_{j\max})^{m_k}}{C_k (0.35 R_m)^{m_k}} \tag{1}$$

where:

$v_j = v$ is the sliding velocity at j-th point of the tooth profile,

$t'_j = 2b_j / v_0$ is the 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,

$j = 0, 1, 2, 3, ..., s$, are the contact points on the active face of the teeth,

$j = 0$, $j = s$ are the first and last point of tooth engagement, respectively,

$v_0 = \omega_1 r_1 \sin \alpha_t$ is the velocity of the contact point along the tooth profile,

α_t is the face pressure angle,

ω_1 is the angular velocity of the pinion,

k = 1; 2 are the numbers of the gears,

f is the sliding friction factor,

R_m is the immediate tensile strength of the material,

C_k, m_k are the characteristics of wear resistance of the gear materials determined in accordance with the methodology reported in **[L. 15]** based on the results of experimental tribological tests.

The wear of the teeth in the time t_* is calculated using the formula:

$$h_{kj} = 60 n_k h'_{kj} t_* \tag{2}$$

where n_k is the number of revolutions of the gears.

The service life of the gear for the allowable wear limit of the teeth, h_{k*}, is found as follows:

$$t = h_{k*} / \bar{h}_{kj} , \tag{3}$$

where $\bar{h}_{kj} = 60 n_k h'_{kj}$ denotes the linear wear of the teeth per one hour of the gear.

The maximum contact pressures $p_{j\max}$ and width of the contact zone $2b_j$ at j - th point of contact are determined from the Hertz formulas:

$$p_{j\max} = 0.564\sqrt{N' / \theta\rho_j} , \quad 2b_j = 2.256\sqrt{\theta N' \rho_j} \tag{4}$$

where:

$N' = N / l_{\min} w$,

$N = T_{nom} K_g / r_{w1} \cos\alpha_w$ is the force acting in tooth engagement,

$T_{nom} = 9550 P / n_1$ is the rated torque,

r_{w1} is the rolling radius of the pinion,

α_w is the pressure angle of the corrected profile,

P is the power on the drive shaft (pinion),

n_1 is the number of revolutions of the drive shaft,

K_g is the dynamic factor,

$\theta = \left(1-\mu_1^2\right)/E_1 + \left(1-\mu_2^2\right)/E_2$ denotes the gear material coefficient,

E, μ are, respectively, the Young’s moduli and Poisson’s ratios describing the materials of teeth,

$l_{\min}$ is the minimum length of the contact line,

w is the number of engaged tooth pairs,

ρ_j is the reduced radius of curvature of the gear profile.

The reduced radius of the curvature of the cylindrical gear profile is as follows:

$$\rho_j = \frac{\rho_{1j}\rho_{2j}}{\rho_{1j} + \rho_{2j}} \tag{5}$$

where ρ_{1j}, ρ_{2j} are the radii of curvature of the tooth flank profiles of the pinion and the gear, respectively.

The formulas for calculating the radii of the curvature of the corrected pinion and gear profiles of a cylindrical gear at *j* - th point of contact are as follows **[L. 20, 21]**:

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

where:

$$\beta_b = \arctan\left(\tan\beta\cos\alpha_t\right), \quad \alpha_t = \arctan\left(\frac{\tan\alpha}{\cos\beta}\right),$$

$$\rho_{t1j} = r_{b1}\tan\alpha_{t1j}, \quad \rho_{t2j} = r_{w2}\sqrt{\left(r_{2j}/r_{w2}\right)^2 - \cos^2\alpha_w},$$

$$\alpha_{t1j} = \arctan\left(\tan\alpha_{t10} + j\Delta\varphi\right), \qquad \alpha_{t1s} = \arctan\sqrt{\left(r_{1s}/r_{w1}\right)^2 - \cos^2\alpha_w},$$

$$\alpha_{t2j} = \arccos\left[\left(r_{2w}/r_{2j}\right)\cos\alpha_w\right],$$

$$r_{b1} = r_1\cos\alpha_t, \quad r_1 = mz_1/2\cos\beta, \quad r_{b2} = r_2\cos\alpha_t, \quad r_2 = mz_2/2\cos\beta,$$

$$\tan\alpha_{t10} = \left(1+u\right)\tan\alpha_w - \frac{u}{\cos\alpha_w}\sqrt{\left(r_{20}/r_{w2}\right)^2 - \cos^2\alpha_w}, \qquad r_{a2} = r_2 + m,$$

$$r_{20} = r_{a2} - r, \quad r = 0.2m,$$

$$r_{2j} = \sqrt{a_w^2 + r_{1j}^2 - 2a_w r_{1j}\cos\left(\alpha_w - \alpha_{t1j}\right)}, \qquad r_{1j} = r_{w1}\cos\alpha_w/\cos\alpha_{t1j},$$

$$a_W = \left(z_1 + z_2\right)m/2\cos\beta,$$

where β is helix angle of the teeth

r_1, r_2 are the radii of pitch circles of the pinion and gear, respectively,

r_{b1}, r_{b2} are the radii of base circles of the pinion and gear, respectively,

r_{a1}, r_{a2} are the radii of addendum circles of the pinion and gear, respectively,

r is the radius of the gear tooth fillet,

u is the gear ratio,

m is the engagement modulus,

z_1, z_2 are the numbers of teeth in the wheel pinion,

a_w is the distance between the gear axes,

$\Delta\varphi$ is the applied angle of rotation of the teeth of the pinion from the point of initial contact (point 0) to point 1, and so on,

α_{t10}, α_{t1s} are the angles indicating location of the first and the last point of engagement of the pinion teeth on the contact line,

α_{t20}, α_{t2s} are the angles indicating location of the first and last points of engagement of the gear teeth on the contact line.

The minimum length of the line of contact is calculated as follows:

$$l_{\min} = \frac{b_W \varepsilon_\alpha}{\cos\beta_b}\left[1 - \frac{(1-n_\alpha)(1-n_\beta)}{\varepsilon_\alpha \varepsilon_\beta}\right] \text{ at } n_\alpha + n_\beta \rangle 1,\; l_{\min} = \frac{b_W \varepsilon_\alpha}{\cos\beta_b}\left[1 - \frac{n_\alpha n_\beta}{\varepsilon_\alpha \varepsilon_\beta}\right] \text{ at } n_\alpha + n_\beta \le 1, \tag{7}$$

where

b_W is the width of the pinion;

$\varepsilon_\alpha, \varepsilon_\beta$ are the coefficients describing the top and step-by-step overlaps of the gear;

n_α, n_β are the fractional parts of coefficients $\varepsilon_\alpha, \varepsilon_\beta$;

$\varepsilon_\alpha = \frac{t_1 + t_2}{t_z}$, $\varepsilon_\beta = \frac{b_W \sin\beta}{\pi m}$, $\varepsilon_\gamma = \varepsilon_\alpha + \varepsilon_\beta$, $t_1 = \frac{e_1}{\omega_1 r_{b1}}$, $t_2 = \frac{e_2}{\omega_1 r_{b1}}$,

$t_z = \frac{2\pi}{z_1 \omega_1}$,

$e_1 = \sqrt{r_{1s}^2 - r_{b1}^2} - r_1 \sin\alpha_t$ is the length of engagement segment at the end of tooth engagement zone,

$e_2 = \sqrt{r_{20}^2 - r_{b2}^2} - r_2 \sin\alpha_t$ is the length of engagement segment at the beginning of tooth engagement zone,

$r_{1s} = r_{a1} - r$, $r_{a1} = r_1 + m$.

The sliding velocity of the teeth in a mesh is calculated as

$$v_j = \omega_1 r_{b1}\left(\tan\alpha_{t1j} - \tan\alpha_{t2j}\right) \tag{8}$$

The angles of transition from double tooth engagement ($\Delta\varphi_{1F_2}$) (segment E_1F_1) to single tooth engagement (segment F_1F_2) and, again, to double tooth engagement ($\Delta\varphi_{1F_1}$) (segment F_2E_2) in a corrected cylindrical gear with helical teeth are determined the following from:

$$\Delta\varphi_{1F_2} = \varphi_{10} - \varphi_{1F_2},\ \Delta\varphi_{1F_1} = \varphi_{10} + \varphi_{1F_1} \tag{9}$$

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

$$\tan\alpha_{F_2} = \frac{r_{w1}\sin\alpha_w - (p_b - e_1) + 0.5b_W\tan\beta_b}{r_1\cos\alpha},$$

$$\tan\alpha_{F_1} = \frac{r_{w1}\sin\alpha_w - (p_b - e_2) - 0.5b_W\tan\beta_b}{r_1\cos\alpha};$$

$p_b = \pi m\cos\alpha_w / \cos\beta$ is the pitch of teeth;

$$e_1 = \sqrt{r_{1s}^2 - r_{b1}^2} - r_{w1}\sin\alpha_w, \quad e_2 = \sqrt{r_{20}^2 - r_{b2}^2} - r_{w2}\sin\alpha_w.$$

The angles $\Delta\varphi_{1E}$ describing the teeth at the end of engagement are determined similarly to the above method, namely:

$$\Delta\varphi_{1E} = \varphi_{10} + \varphi_{1E} \tag{10}$$

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

Some of the above formulas must include the parameters of gear correction.

Height correction:

The radii of tooth addendums:

$$r_{a1} = r_1 + (y_{a1} + x_1)m, \quad r_{a2} = r_2 + (y_{a2} + x_2)m \tag{11}$$

where $x_1 = -x_2$ are the coefficients of addendum correction,

y_{a1}, y_{a2} are the coefficients of tooth addendum height of gears 1 and 2, respectively.

The remaining parameters of the gear are the same as those of the uncorrected gear.

Angular correction:

Here, $x_1 \neq x_2$, whereas the total $x_\Sigma = x_1 + x_2$.

The real distance between the axes is

$$a_{wk} = r_{w1} + r_{w2} \rangle a_w \tag{12}$$

The rolling pressure angle α_w will depend on the real distance between the axes of the cooperating gears and will be higher (at $\alpha_{wk} > \alpha_w$) than the transverse pressure angle α_t. Once we know the real distance between the axes, then

$$\alpha_w = \arccos \frac{a_w}{a_{wk}} \cos \alpha_t \tag{13}$$

The radii of the pinion and gear:

$$r_{w1} = r_1 \frac{\cos \alpha_t}{\cos \alpha_w}, \quad r_{w2} = r_2 \frac{\cos \alpha_t}{\cos \alpha_w} \tag{14}$$

The radii of the tooth addendums:

$$r_{a1} = r_1 + (1 + x_1 - K)m, \quad r_{a2} = r_2 + (1 + x_2 - K)m, \tag{15}$$

where: $K = \frac{a_w - a_{wk}}{m} + x_\Sigma$ denotes the addendum reduction coefficient.

The above parameters of profile correction should be included in the following formulas:

– $N = 9550 P K_g / r_{w1} n_1 \cos \alpha_w$,

– $\tan \alpha_{t10} = (1 + u) \tan \alpha_w - \frac{u}{\cos \alpha_w} \sqrt{\left(r_{20} / r_{w2}\right)^2 - \cos^2 \alpha_w}$,

– $\alpha_{t1s} = arctg \sqrt{\left(r_{1s} / r_{w1}\right)^2 - \cos^2 \alpha_w}$ $\rho_{t2j} = r_{w2} \sqrt{\left(r_{2j} / r_{w2}\right)^2 - \cos^2 \alpha_w}$,

– $r_{2j} = \sqrt{a_w^2 + r_{1j}^2 - 2a_w r_{1j}\cos\left(\alpha_w - \alpha_{t1j}\right)}$,
– $r_{1j} = r_{w1}\cos\alpha_w / \cos\alpha_{t1j}$,
– $\tan\alpha_{t2s} = \left(1+u^{-1}\right)\tan\alpha_w - \dfrac{1}{u\cos\alpha_w}\sqrt{\left(r_{1s}/r_{w1}\right)^2 - \cos^2\alpha_w}$,
– $\alpha_{t2j} = \arccos\left[\left(r_{2w}/r_{2j}\right)\cos\alpha_w\right]$,
– $e_1 = \sqrt{r_{1s}^2 - r_{b1}^2} - r_{w1}\sin\alpha_w$, $e_2 = \sqrt{r_{20}^2 - r_{b2}^2} - r_{w2}\sin\alpha_w$.

NUMERICAL SOLUTION

The input data included the following: $z_1 = 20$; $z_2 = 80$; $m = 3$ mm; $u = 4$; $n_1 = 750$ revolutions per minute; $P = 6$ kW; $f = 0.05$; $b = 30$ mm; $\beta =$ 0^0, 10^0, 12^0; K_g = 1.6. The following materials were used: the pinion was made of 38HMJA steel after nitriding with 58 HRC; $R_m = 1040$ MPa, C_1 = 3.510^6, m_1 = 2; the gear was made of 40H steel after bulk heat treatment with 53 HRC, $R_m = 981$ MPa, $C_2 = 0.1710^6$, $m_2 = 2.5$; $E = 2.110^5$ MPa, $\mu = 0.3$. As a lubricant, we used engine oil with an anti-wear additive described by a kinematic viscosity of $\nu_{+50^\circ} \approx 15$ cSt; $h_{k*} = 0.5$ mm; step $\Delta\varphi =$ 4^0. The correction coefficients and geometrical parameters of the gear were as follows:
a) Height correction: $x_1 = -x_2$ = 0; 0.2; 0.4; 0.6; $x_\Sigma = 0$; a_w = 150 mm;
b) Angular correction:
$\beta =$ 0^0: $x_1 = 0.4 \ldots 1$, $x_2 = 0.4566 \ldots 1.0566$; $x_\Sigma = 1.4566$; a_w = 150 mm; $a_{wk} = 154$ mm; $\alpha_w = 23.754^0$;
$\beta =$ 10^0: $x_1 = 0 \ldots 0.5$, $x_2 = 0 \ldots 0.584$; $x_\Sigma = 0.584$; a_w = 152.314 mm; $a_{wk} = 154$ mm; $\alpha_w = 21.918^0$;
$\beta = 12^0$: $x_1 = 0 \ldots 0.2$, $x_2 = 0 \ldots 0.2196$; $x_\Sigma = 0.2196$; a_w = 153.351 mm; $a_{wk} = 154$ mm; $\alpha_w = 21.049^0$.

To solve the problem, the following conditions are taken into account:

The teeth are in double-single-double engagement.

The dynamic character of work is defined by the dynamic factor K_g .

The maximum contact pressures $p_{j\max}$ are maintained constant during gear operation.

Boundary lubrication conditions are ensured.

The results are given in **Figs. 1–7**. **Fig. 1** shows the variations in $p_{j\,max}$ in the uncorrected gear versus pitch angle. On increasing the pitch angle, the double tooth engagement zone increases too (left and right sides of the plot), which provides favourable conditions for gear operation due to a decrease in the maximum contact pressures. Subsequently, the maximum pressures are nearly the same at the beginning of both the double (left side) and single tooth engagement (centre of the plot), which increases the reliability of the gear.

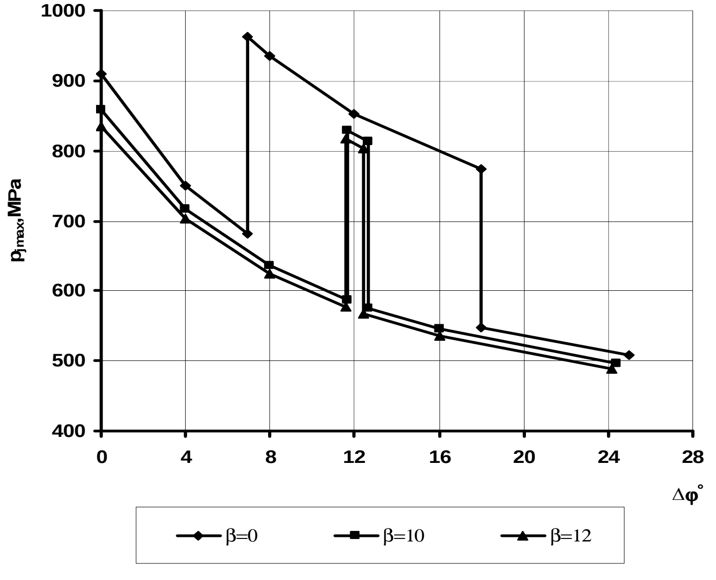

Fig. 1. Teeth inclination angle versus maximum contact pressures in a gear with no profile correction

Rys. 1. Wpływ kąta pochylenia zębów na maksymalne naciski stykowe w przekładni niekorygowanej

The analysis shows that the greatest attention should be paid to the beginning of the tooth engagement zone (from the beginning to the centre of tooth engagement). The highest contact pressures occur in this zone, and thus we can expect that the highest wear of tooth surface will be observed here, too. The use of helical gears, even with a small angle of teeth line inclination, substantially reduces the zone of single tooth engagement. The results of uncorrected gear indicate that the contact pressures have similar values at the beginning of tooth engagement and in its centre in the case of helical gears described by the angle $\beta = 12^\circ$ (and higher), which enables calculating tooth surface wear using simplified methods connected with the geometric parameters of the tooth engagement centre.

Height correction:

Figs. 2–4 show the regularities concerning the effect of height correction of the gear profile on the examined parameters, depending on pitch angle. **Fig. 2** illustrates the change in $p_{j\,max}$ of the teeth in mesh at the applied angles β. The height correction has a positive effect on these pressures in the course of tooth engagement, as it makes them decrease by about 15%.

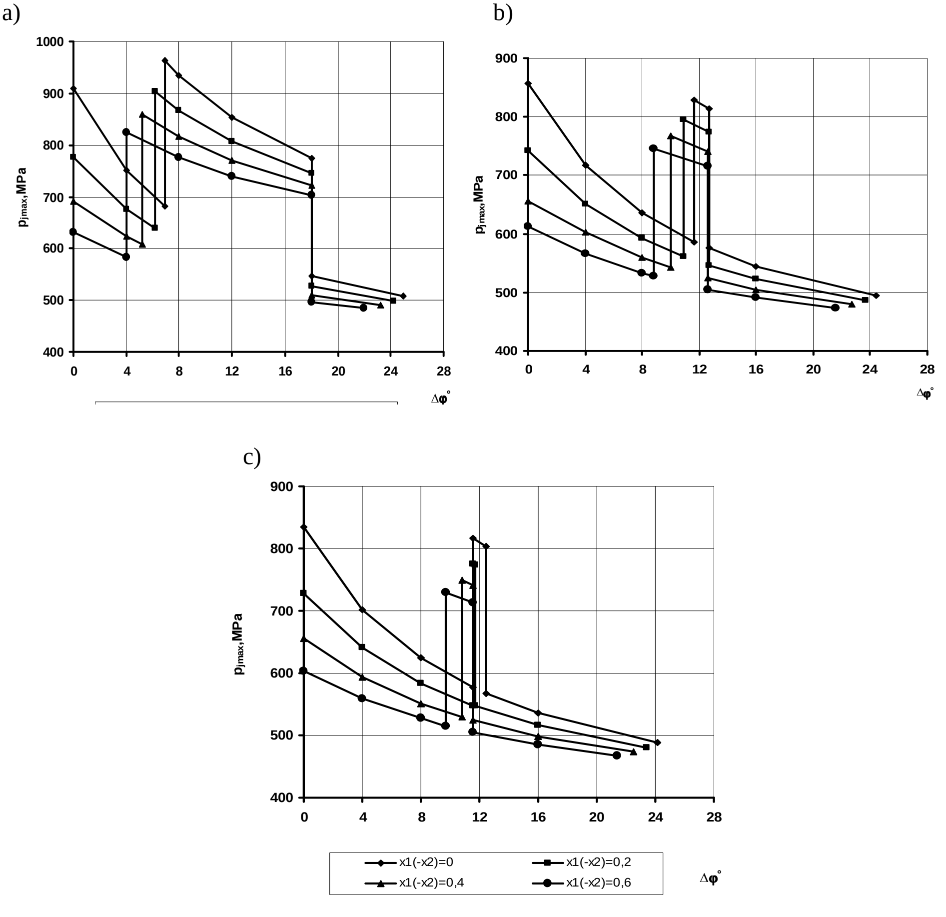

Fig. 2. Height correction of the gear profile versus maximal contact pressures: a) $\beta = 0^o$, b) $\beta = 10^o$, c) $\beta = 12^o$

Fig. 2. Wpływ korekcji technologicznej zebów na maksymalne naciski stykowe: a) $\beta =$ 0^o, b) $\beta =$ 10^o, c) $\beta =$ 12^o

The linear wear h_{2j} of the gear teeth is illustrated in **Fig. 3.**

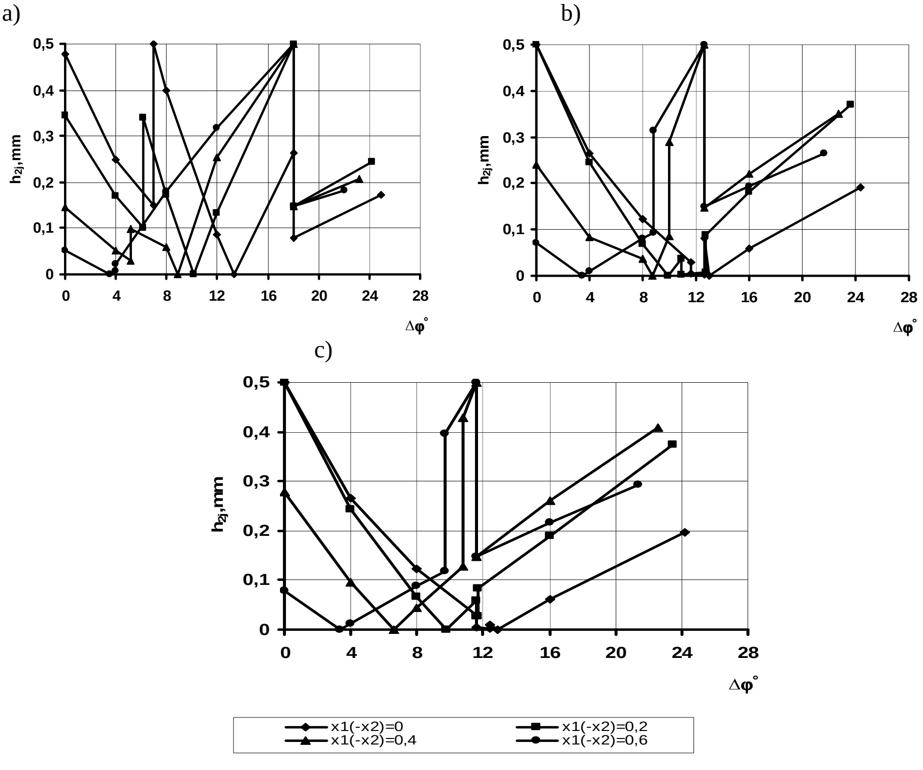

Fig. 3. Height correction of the gear profile versus linear wear of the gear teeth: a) $\beta = 0^o$, b) $\beta = 10^o$, c) $\beta = 12^o$

Rys. 3. Wpływ korekcji technologicznej zębów na ich zużycie liniowe: a) $\beta = 0^o$, b) $\beta = 10^o$, c) $\beta = 12^o$

Height correction has a significant effect on contact pressures, particularly at the beginning of the tooth engagement zone. The maximum contact pressures occur at the point of transition from double to single tooth engagement. The impact of this correction increases with increasing the correction coefficient. The decrease in contact pressures at the beginning of tooth engagement (the beginning of tooth-pair contact) is estimated to be 20%–40%. At the end of tooth engagement (the end of tooth-pair contact), the correction has a much lower effect on the values of contact pressure.

As a result, this type of correction also affects the values of wear in individual regions of the tooth engagement zone in contrast to the uncorrected gear profile. The analysis of the plots shows that, in the case of corrected tooth engagement, the maximum wear h_{2*} is usually located at the end of the single tooth engagement. The wear h_{1j} of the pinion teeth is similar to the wear of gear teeth yet it proceeds about two-times more slowly.

The variations in the gear life t_{min} for the allowable wear limit h_{2*} at one of the points of contact are plotted in **Fig. 4.**

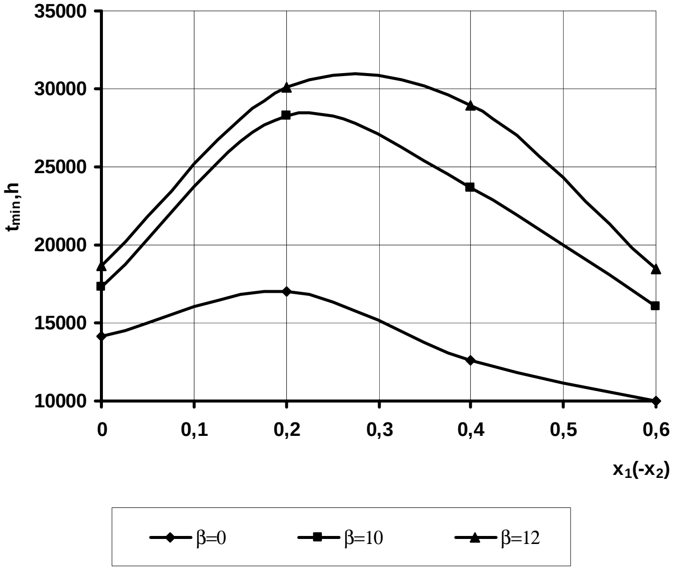

Fig. 4. Height correction of the gear profile versus gear life
Rys. 4. Wpływ korekcji zębów na trwałość przekładni

It can be observed that the service life of gears improves with increasing the pitch angle of teeth. The solution of the problem demonstrated that the profile correction coefficients are optimum when t_{min} reaches its maximum value, which is by 1.66 times higher than that of the gear where no profile correction was applied.

Angular correction:

Apart from height correction, which ensures that the distance between the gear axes is constant, one can also use angular correction to ensure the required interaxial spacing for a given pitch angle. The main role of angular correction is to keep the required distance between the gear axes and – similarly to height correction – to optimize teeth meshing conditions with respect to gear life.

Figs. 5–7 illustrate the results of angular correction, comparing these results with those obtained in height correction.

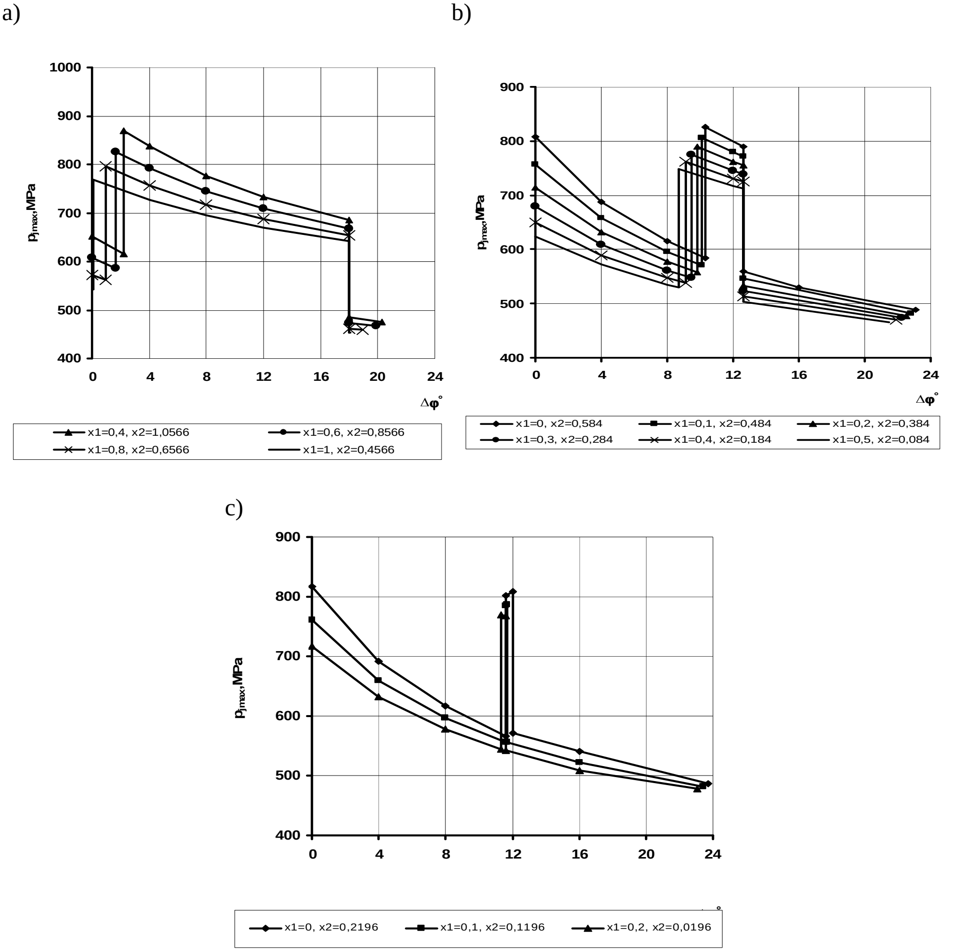

Fig. 5. Angular correction of the gear profile versus maximum contact pressures: a) $\beta = 0^o$, b) $\beta = 10^o$, c) $\beta = 12^o$

Rys. 5. Wpływ korekcji konstrukcyjnej zębów na maksymalne naciski stykowe: a) $\beta = 0^o$, b) $\beta = 10^o$, c) $\beta = 12^o$

The angular correction of the gear profile is characterized by the fact that the values of $p_{j\,max}$ in the spur gear are much higher at the beginning of the single tooth engagement than at the beginning of the double tooth engagement. Due to the teeth pitch, the differences in $p_{j\,max}$ are somewhat lower. This type of correction reduces the maximum contact pressures by about 20%. In addition, the zone of single tooth engagement in the spur gear is much wider.

The observed regularities with respect to the teeth wear h_{2j} are plotted in **Fig. 6.** These regularities agree to some degree with those obtained using the angular correction.

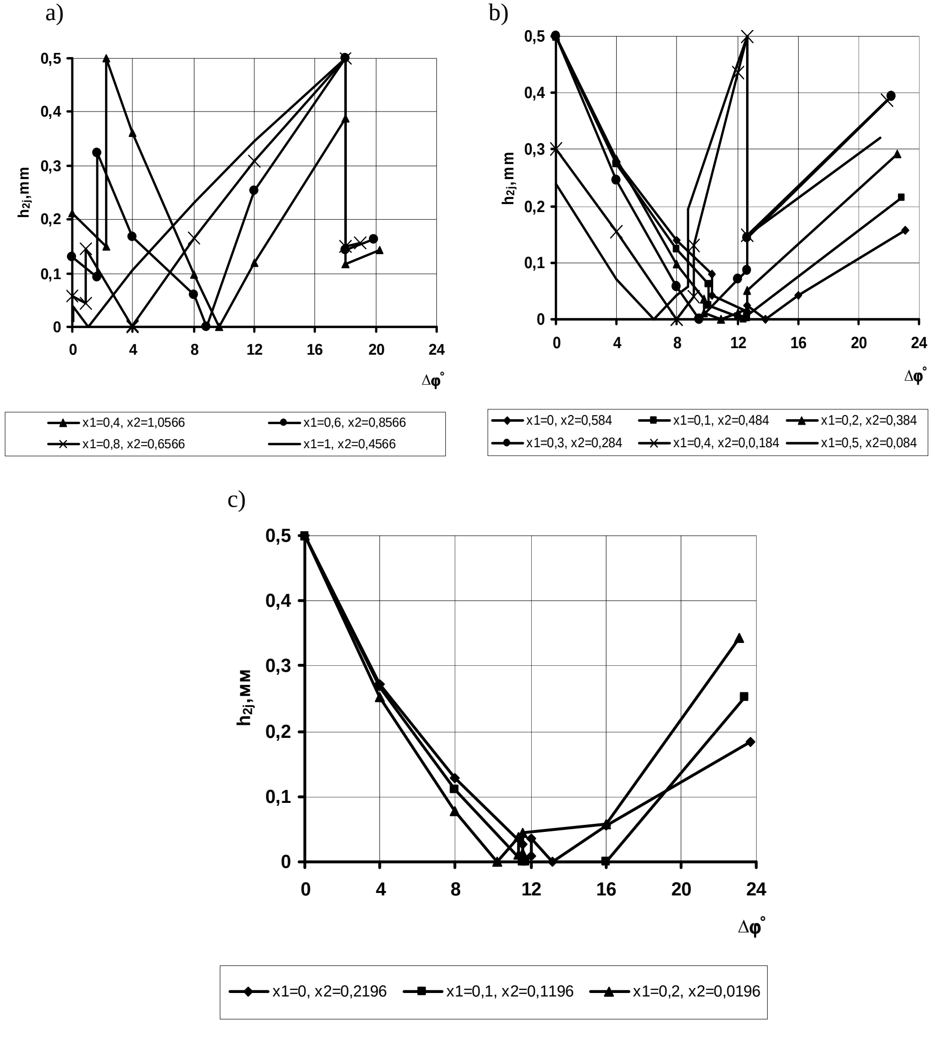

Fig. 6. Angular correction of the gear profile versus linear wear of the gear teeth: a) $\beta = 0^0$, b) $\beta = 10^0$, c) $\beta = 12^0$

Rys. 6. Wpływ korekcji konstrukcyjnej zębów na ich zużycie liniowe: a) $\beta = 0^o$, b) $\beta = 10^o$, c) $\beta = 12^o$

However, for a certain range of variations in the profile correction coefficient x_1 of the pinion, the allowable wear limit is reached at the beginning of single tooth engagement for spur gears (**Fig. 6a**), while for helical gears the allowable wear limit is first reached at the beginning of double tooth engagement (**Figs. 6b, c**), a trend which was not observed for height correction, and then at the end of double tooth engagement (**Fig. 3**).

The relationship between the gear life t_{min} and selected values of the β angle is plotted in **Fig. 7**. When $\beta = 0^\circ;10^\circ$, t_{min} is optimum due to the application of certain values of the profile correction coefficients x_1, x_2. When $\beta = 12^\circ$, the above trend does not apply. The above method for selecting profile correction coefficients x_1, x_2 improves the gear life by two times.

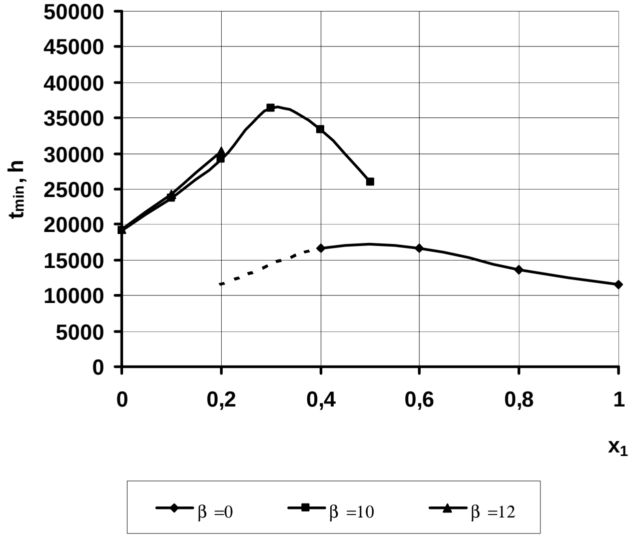

Fig. 7. Angular correction of the gear teeth versus gear life
Rys. 7. Wpływ korekcji konstrukcyjnej zębów na trwałość przekładni

CONCLUSIONS

1. In the paper, we presented a new method for estimating the wear and life as well as teeth meshing conditions of cylindrical gears with corrected profiles.
2. The following regularities regarding the maximum contact pressures in mixed (double-single-double) tooth engagement were observed:
   – With increasing the tooth line inclination angle β, the contact pressures gradually decrease during the entire working cycle of a single tooth pair.

This is connected with the fact that the tooth contact line increases proportionally to the angle β. This regularity is observed for all investigated types of profile correction.

- The maximum contact pressures are observed at the beginning of tooth engagement, where the profile correction has a great impact on the variations in these pressures. The contact pressures decrease by 15% due to the height correction and by 20% when the angular correction of the gear profile is applied. In both cases, the pinion is assigned a positive value of the engagement coefficient.
- The impact of profile correction on the values of contact pressures in the second phase of engagement (end of engagement) is insignificant.

3. As for height correction, the highest wear occurs at two points: at the beginning of tooth engagement, where the double tooth engagement changes to the single tooth engagement, and at the end of engagement, i.e. at the final point of single tooth engagement.
   An increase in the correction coefficient causes a decrease in the gear wear at the beginning of tooth engagement and, at the same time, a slight increase in the gear wear at the end of tooth engagement.
4. The application of angular correction produces similar effects as height correction, particularly in spur gears.
   Regarding helical gears, the profile correction has a similar effect on their linear wear; therefore, we can conclude that the correction coefficient value assigned to the pinion exerts the greatest impact here. With increasing the correction coefficient, the gear wear decreases at the beginning of tooth engagement, while it significantly increases at the end of tooth engagement.
   By describing the pinion teeth and gear teeth with suitable values of the correction coefficient, we can produce uniform wear of the tooth surface in both single and double engagement zones.
5. It was found that gear teeth reach the allowable wear limits two times faster. Depending on the applied profile correction coefficients and pitch angle, the wear occurs at different points of contact: at the beginning of the double tooth engagement and at the beginning or end of the single tooth engagement.
6. Profile correction can prolong gear life. It was found that the service life of the examined gears can be prolonged if optimum profile correction coefficients are applied:
   - Height correction increases the gear life by 1.66 times.
   - Angular correction increases the gear life by up to 2.0 times.
7. On increasing the correction coefficient, the double tooth engagement zone becomes slightly shorter.

REFERENCES

1. Pasta A., Mariotti Virzi G.: Finite element method analysis of a spur gear with a corrected profile. J. Strain Analysis. 41/2007, p. 281–292.
2. ISO 6336-2. Calculation of load capacity of spur and helical gears – Part 2: calculation of surface durability (pitting), International Standard, 1st edition, 15 May 1996 (International Organization for standardization, Geneva).
3. Ulaga S. M., Ulbin M., Flasker J.: Contact problems of gears using Overhauser splines. Int. J. Mech. Sci, 41/1999, p. 385-395.
4. Zwolak J., Martyna M.: Analiza naprężeń kontaktowych i naprężeń zginających występujących w przekładniach zębatych power shift. Tribologia, 3/2011, s. 155–165.
5. Zwolak J., Witek M.: Optymalizacja parametrów geometrycznych kół zębatych w aspekcie minimalizacji naprężeń kontaktowych. Tribologia, 6/2011, s. 283–291.
6. Drozdov Yu.: To the development of calculation methods on friction wear and modeling . Wear resistance. Science, Moscow, 1975, p. 120–135.
7. Pronikov A.: Reliability of machines. Mashinostroenie, Moscow, 1978.
8. Grib V.: Solution of tribotechnical tasks with numerous methods. Science, Moscow, 1982.
9. Flodin A., Andersson S.: Simulation of mild wear in spur gears. Wear, 207/1997, p. 16-23.
10. Flodin A., Andersson S.: Wear simulation of spur gears. Tribotest Journal, 5/1999, p. 225–250.
11. Brauer J., Andersson S.: Simulation of wear in gears with flank interference – a mixed FE and analytical approach. Wear, 254/2003, p.1216–1232.
12. Kahraman A., Bajpai P., Anderson N.E.: Influence of tooth profile deviations on helical gear wear. J. Mech. Des, 127/2005, p. 656–663.
13. 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., 132/2010, p. 132–142.
14. Mekhalfa A., Bonaricha A., Kallouche A., Hadjadj E.: Teeth's Gear Correction. J. Rev. of Mech. Ing., 3/2009, p. 271–274.
15. Czerniec M., Kiełbiński J.: Prognozowanie trwałości tribologicznej kół zębatych walcowych ewolwentowych. Wyd. Politechniki Lubelskiej, 2003.
16. Chernets M.V., Kelbinski J., Jarema R.J.: Generalized method for the evaluation of cylindrical involute gears. Materials Science, 1/2011, p. 45–51.
17. Czerniec M., Kiełbiński J., Jarema R.: Oszacowanie wpływu korekcji zębów na zużycie, trwałość oraz wytrzymałość kontaktową ewolwentowych przekładni walcowych o zębach prostych. Tribologia, 3/2011, s. 17–30.
18. Chernets M.V., Jarema R.Ja.: Generalized value method of teeth correction influence on resource, wear and contact durability of cylindrical involute gears. Materials Science, 4 /2011, p. 56–58.
19. Chernets M.V., Yarema R.Ya., Chernets Yu.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 live and wear. Materials Science, 3/2012, p. 289–300.

20. Chernets M.V., Yarema R.Ya., Chernets Yu.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 2. Contact strength. Materials Science, 6/2012, p. 752–756.
21. Czerniec M., Kiełbiński J.: Wpływ korekcji przekładni walcowej o zębach ewolwentowych na zużycie oraz trwałość. Tribologia, 1/ 2015, s. 13–24.

Streszczenie

Na podstawie autorskiej metody obliczenia zużycia warstwy wierzchniej na wysokości czynnej zębów i trwałości przekładni zębatych przeprowadzono badania wpływu warunków interakcji zębów z uwzględnieniem parowości ich zazębienia w walcowej przekładni zębatej o zębach skośnych. Rozważono także wpływ korekcji technologicznej (P-O) i konstrukcyjnej (P) na wytrzymałość kontaktową, zużycie oraz trwałość. Przedstawiono sposób określenia stref występowania dwuparowego oraz jednoparowego zazębienia. Ustalono, że korekcja zazębienia powoduje obniżenie maksymalnych nacisków stykowych w granicach 15–20% zależnie od jej rodzaju. Rozkład nacisków w znacznym stopniu zależy od warunków zazębienia zębów. Na przebieg zużywania zębów znacząco wpływają warunki ich interakcji. W zależności od wartości współczynników korekcji oraz jej rodzaju zużycie dopuszczalne zębów koła zębatego wystąpi w różnych charakterystycznych punktach styku: na wejściu w zazębienie dwuparowe, na wejściu lub wyjściu z zazębienia jednoparowego. W wyniku rozwiązania numerycznego ustalono również, że resurs przekładni przy wyborze optymalnych wartości współczynników korekcji może ulec znacznemu podwyższeniu: przy korekcji technologicznej do 1,66 razy, a przy korekcji konstrukcyjnej ok. 2-krotnie.