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A study of factors affecting wear and destruction of teeth in gear mechanisms

Badanie czynników warunkujących stopień zużycia i zniszczenia zębów w mechanizmach przekładni zębatych

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ABSTRACT: Transmission gears are very widely used in the modern construction of the machine. Even though electrical transmissions are used, mainly mechanical transmissions are widely used in the machine building industry as they are independent and better than other transmissions. Distributing power between the transmission mechanisms and working units of various machines, by changing the angle of momentum by adjusting the speed of the parts, it is possible to change the type of movement, turn the mechanism on and off, and change the direction of the transmission mechanisms. One of the most widespread types of transmission technology is gear transmission. Tens of thousands of kilowatt-hours of gears can be transmitted. The advantages of gear transmission in comparison with others are their ability to maintain their reliability and long-term functioning of the teeth, high value of the working coefficient (0.90–0.99), simple construction, lack of special service, simplicity of the transmission number and compactness of dimensions. The missing aspects of these transmissions are high precision in their preparation and installation, and low noise level. The main criterion for the working ability of the closed gear transmissions is the contact endurance of the active surface of the tooth. For this reason, the main dimensions of the transmission are determined by the contact tension and by teeth bending.

Key words: gear wheel, tension, endurance, bending.

STRESZCZENIE: Przekładnie zębate znajdują szerokie zastosowanie w nowoczesnych maszynach. Chociaż stosuje się także przekładnie elektryczne, to w przemyśle maszynowym stosowane są głównie przekładnie mechaniczne, ponieważ są one niezależne i lepsze od innych rodzajów przekładni. Przekładnie zębate umożliwiają rozprowadzenie mocy między mechanizmami przekładni a różnymi jednostkami roboczymi poszczególnych maszyn, poprzez zmianę kąta pędu i regulację prędkości części. Dzięki temu można zmieniać rodzaj ruchu, włączać i wyłączać mechanizmy oraz zmieniać kierunek mechanizmów przekładni. Jednym z najbardziej powszechnych rodzajów przekładni są przekładnie zębate. Mogą one przenosić dziesiątki tysięcy kilowatogodzin mocy. Przewaga przekładni zębatych nad innymi mechanizmami tego typu tkwi w ich niezawodności, trwałości zębów, wysokiej wartości współczynnika pracy (0,90–0,99), prostej konstrukcji, braku wymogu specjalnego serwisowania, prostocie liczby przełożeń i niewielkich wymiarach. To, czego im brakuje, to zachowanie wysokiej precyzji wykonania i montażu oraz niski poziom hałasu. Głównym kryterium decydującym o sprawności zamkniętych przekładni zębatych jest wytrzymałość styku aktywnej powierzchni zęba. Z tego powodu główne wymiary przekładni zależą od naprężenia stykowego i zginania zębów.

Słowa kluczowe: koło zębate, naprężenie, wytrzymałość, zginanie.

Introduction

In both open and closed transmissions, the tension created by the abrasive particles on the working surfaces, teeth of which are not protected from contamination, are subjected to abrasive wear of the teeth.

When the worn teeth get stuck, the gap between them increases which causes an increase of dynamic loads and the noise and the kinematic accuracy of the transmission decreases. This leads to a decrease in the strength and cross-sectional area of the teeth and even to breakage.

To prevent corrosion, it is necessary to strengthen the surface of the teeth and reduce their roughness, protect the transmission from abrasive particles, reduce the relative sliding speed in the correction of engagement, and use high-viscosity oils in lubrication.

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The wear of the teeth in the contact zone depends mainly on the power expended to overcome the friction forces. The total wear in the contact zone depends on the phase of engagement, the time and intensity of wear of the contacting bodies.

The wear, which depends on the radial runout, provides wear of the tooth in areas with zero values of the sliding speed, in proportion to the value of the instantaneous temperature.

Drozdov et al. (1986). gave the most clear, systematic approach to the development of wear calculation methods. The durability of the designed transmission subject to fatigue failures is evaluated based on the transmissions experience operating under conditions of wear and high temperature (Kudryavtsev, 1973). An experimental study by Kragel'kiy et al. (1977) showed that the wear rate of the driven wheels is greater than the one of the driving wheels. The specific power expended to overcome the friction of the contact surfaces was used as a wear criterion (Mustafayev, 2013). The dependence of volumetric wear on the work of friction forces was also used by Fleischer (1973) when developing the energy theory of wear.

The problem of wear resistance of gears takes a special place in the process of occurrence and development of damage to the friction surface of the teeth. Depending on the conditions, wear at a low temperature during surface friction may stop in some cases; under extreme friction conditions (at high temperatures, loads, sliding speeds, and the presence of aggressive media), the process of wear of the contact surfaces is accompanied by wear at a high temperature, which is characterized by catastrophic wear of mating pairs and can lead to complete failure of the teeth. Jamming at a high temperature is the most dangerous type of gear tooth damage. At the moment of jamming the coefficient of sliding friction usually sharply increases, the temperature rises, dynamic processes are activated, and vibrational and acoustic activity increases.

The most theoretically based calculation for jamming, which has undergone extensive experimental testing in laboratory conditions, is the block temperature criterion, based on the hypothesis of the existence of a critical temperature of oil film break characteristic for each combination of materials and oil. The temperature in the contact is defined as the sum of the surface temperature of the bodies before contacting and the instantaneous temperature increase in the contact during the friction of the bodies – a temperature flash (Fleischer, 1973).

Onishchenko (1999) introduced the dynamic model of a variable gear ratio, due to distortion as a result of tooth wear, the shape of the profiles, by representing the movement of the tooth as the sum of the movement of its base in a rigid model with a variable gear ratio and relative oscillatory movement in an elastic model with four degrees of freedom.

The task of improving the gear designing method is reduced to approximating the physical and geometrical values of the gear and wheel, finding the desired combination of the tooth profile shape, allowing to have the maximum thickness of the lubricant layer and minimum stresses in the contact area of the teeth engagement.

The method of nonlinear programming was used, to solve this problem: the search for the minimum of the objective function that determines the shortest distance between many different points on the profiles of adjacent teeth (Dorofeev et al., 2014).

The solution to the problem of calculating contact stresses is analyzed by the method of fictitious load. The method is based on the dependence of contact stresses on the longitudinal modification of the teeth obtained by solving a system of equations. Near the contact area, the calculated contact stresses make up the main part of the total stresses, which allows them to be calculated and taken into account, ignoring other stress components (Dorofeev, 2016a). The problem of the distribution of contact loads in gearing cannot be solved only by calculating a fictitious load. When designing gears, the formulation of the problem of a mathematical model is reduced to calculating the vibration level of a gear, choosing a rational geometry, and optimal modification of the teeth. The solution to this problem is a more complex modification of the teeth. The calculation of contact stresses in the areas of edge engagement according to the proposed method is carried out by choosing a gear transmission that provides a resource for contact stresses, using computer simulation methods to calculate the shape of the tooth profile modification line (Dorofeev et al., 2012).

Possible options for modifying the side surface of the teeth of the wheels include the smoothness of the entry and exit from the engagement of the tooth profile, the redistribution of contact loads along the length of the teeth, the combined provision of smooth entry and exit from the engagement of the teeth, and the redistribution of contact loads along the profile and length of the tooth (Krotov, 2016). The reduction of contact loads and increase in the durability of the gear transmission and the formulas for the transformation of the parameters of the initial contour of the gear teeth are considered. It is proposed to replace the coefficients of the teeth height with the coefficient of the teeth thickness at the top of the studied gears, which will make it possible to obtain the optimal teeth geometry. Designing a gear transmission with a non-standard initial contour makes it possible to reduce the acting stresses and the kinetic error by 1.5 times (Dorofeev, 2016b). An assessment of the state tension of gears, mainly for single-pair gears of wheels, is given in the works of Kapelevich (1991), Ananiev et al. (2013), Dorofeev et al. (2012, 2014), Dorofeev (2016a, 2016b), Krotov (2016).

The developed computational program NOVKS-14 makes it possible to sequentially calculate the contact and bending tension of the teeth in various phases of a real multi-pair Novikov gearing with the identification of the most stressed zones that determine the load capacity along the entire length of the gear transmission (Korotkin et al., 2015).

The effectiveness of the longitudinal modification of the teeth surfaces, providing the required load capacity of the transmission, was evaluated. The problem posed is reduced to finding a spatial curve of the tooth line, taking into account the nature of the distribution of the conducted load over the contact areas in a multi-pair engagement and a stressed state in various sections along the tooth length, as well as at the ends at the moment of the axial pairing of the teeth. The efficiency of longitudinal modification increases with decreasing precision in the manufacture and assembly of the gear transmission. The longitudinal modification of the teeth makes it possible to achieve an increase in the load capacity in terms of contact and bending stresses by several times (Korotkin, 2015).

As a result of the study of the contact interaction of gears, the initial contour of the teeth was determined, which is the basis of the parameters for solving contact problems. The contact tension and endurance of the tooth surfaces were also determined (Korotkin, 2016).

Methods for eliminating the defect are considered and the parameters of the longitudinal modification of the teeth are determined, which ensure the elimination of local zones of peak contact stresses. The research work was carried out as a measure to eliminate the chipping defect, using the finite element method (Kalinin et al., 2013).

Teeth shaping methods that affect the performance of gears are presented in the works of Cherepakhin and Vinogradov (2014) and Kalashnikov et al. (2010). The geometry of the tooth profile is formed during gear cutting and depends on the type of gear-cutting tool used, its geometry, and the parameters of the wheel being cut. It is noted that, depending on the chosen method of cutting the tooth profile, it is possible to reduce the stress concentration at the base of the teeth, increasing the bending endurance due to chemical-thermal treatment (Kalashnikov et al., 2013). The hardening of the surface layer and the viscosity of the wheel core can inhibit the development of fatigue failure at the base of the tooth. Chemical-thermal treatment significantly increases the resistance of gear materials to the occurrence of plastic deformation and reduces the possibility of metal adhesion in interacting metal surfaces (Semenov and Ryzhova, 2012).

Measures taken at the design and production stage, through the implementation of feedback from the operation stage, have the greatest potential effect of increasing the durability of gears. Technological methods do not allow checking whether all the possibilities are realized for the designed gear, therefore, to increase the bending, contact strength, and durability of gears, it is necessary to conduct several computational experiments aimed at finding the optimal gearing, which will provide uniform load and temperature distribution over the entire tooth width.

Task setting: Application of factors affecting the destruction of teeth in gear transmissions.

Task solving: Coefficients affecting the strength of the tooth subjected to dynamic loads. Damage characteristic as overloading, low-speed, and medium-speed gears cause wear on the surface of the teeth. Corrosion of the upper surface of the teeth occurs due to the breaking of the oil layer between the touching profiles and the generation of contact stresses between the metal surfaces caused by the effect of high pressure and temperature. If the number of microcontacts between the touching surfaces is small, then small areas on the upper layer of the teeth are destroyed when the contact is separated, and the temperatures generated in the working profiles are rapidly reduced, the thermal conductivity inside the metal particles and the oil layer between the teeth are restored, and a mild form of tooth wear occurs. Because the metal particles are rubbed off only on one side of the tooth surface, tooth wear occurs gradually.

When the number of microcontacts on the tooth increases, the process of heat release from the surface is delayed, and the amount of heat starts to increase during the rotation of the gear; after a certain period of time, the thin layer of oil on the tooth is no longer restored, and as a result of the effect of high temperature, soft metal particles break off from the gear in contact with its tooth profile and stick to the tooth profile of the other wheel, then adhesions form between the touching surfaces. Bubbles in a hard tooth form grooves in the relatively soft tooth it contacts in sliding, causing the transmission to fail for a short period. In addition to choosing a rational material for the gear mechanism, the moment that ensures the prevention of wear on the upper layer of the tooth is the use of special anti-friction oils with the inclusion of highly viscous and chemically active substances.

Plastic sliding deformation can occur on the contact surfaces of the teeth of soft material gears in overloaded transmissions. This sliding leads to a violation of the regularity of the sticking and disintegration of the teeth in the sticking line of the gear teeth.

To prevent the occurrence of plastic sliding, it is required to increase the strength of the material of the gear (Mustafayev, 2015).

Low-quality thermal and chemical thermal treatment of the working surface of the teeth sometimes leads to the protection of metal particles on the surface of the tooth in the form of a layer. Such cases occur due to defects in the nitrogenized or cemented upper layer of the tooth during teeth grinding, or if the core part of the tooth does not have the required strength, the fragile working surface is compressed under the influence of a large load. One of the causes of rubbing is the overloading of the transmission.

The main reason for tooth wear in gears that are sufficiently lubricated and protected from dust and dirt in a closed circuit is the wear and tear of working surfaces due to changing contact stresses.

Abrasion usually occurs due to small cracks at the bottom of the teeth, near the attachment pole. The load is transferred to this zone by only one pair of teeth, in the direction of sliding and rolling of the teeth, the oil is squeezed into the cracks, allowing metal particles to break off. If initial corrosion occurs in the pores caused by defects during the production of wheels, then after some time, due to this corrosion and plastic deformation, the number of micro-cracks and the concentration of the load decreases, and the pores expand. Such abrasions do not adversely affect the operation of the gear transmission. Small pores formed at the beginning of the attachment line multiply and spread along the working surface of the bottom of the tooth, and sometimes to the top of the tooth.

Although the teeth that wear out without fatigue still retain the ability to stick for a long time in the gear transmission, increased dynamic loads occur that accelerate the disintegration of the teeth in the sticking engagement.

In the teeth of transmissions which are subjected to a certain degree of wear, the cracks that are caused by the friction of the upper layers of the tooth do not have the opportunity to expand. To prevent rubbing on the surface of the teeth, it is required to report the tolerance according to their contact stresses, strengthen the surface of the material and increase the precision in the preparation of gears (Yerokhin, 2005).

Stresses from bending can cause teeth to break. Fractures can occur in two cases, without fatigue and additional loading.

Fatigue fracture is caused by the disintegration of the base of the tooth under the influence of stresses caused by a large number of repeated loads that exceed the material's endurance limit. Fracture caused by overloading occurs because of static and dynamic forces. Non-fatigue fractures may occur due to improper bending reporting methods, while non-fatigue fractures may occur due to failure of random factors which could cause a tooth to be stuck against this load. The gears used in various mechanisms and devices are made of different materials depending on the conditions.

To reduce the weight and the dimensions of the gear, heat-treated alloy steels are preferred. Steel gear teeth can be cut before and after heat treatment. If the teeth are cut after heat treatment, this will not affect the dimensions of the teeth and the accuracy of the transmission. Heat-treated steels are used for teeth cutting in small gears up to HB ≤ 350 and in medium and large gears HB < 280.

When choosing gear material and type of heat treatment, the main focus is on increasing the load in the transmission.

The force generated in the sticking causes not only the generation of additional dynamic loads due to the uneven distribution of the load along the contact line but also the deformation of shafts, bodies, and supports (Kurmaz and Kurmaz, 2007).

Such loads have a negative effect on the occurrence of certain errors in the installation and preparation of parts included in the transmission.

Several factors affect the breakdown and failure of teeth during the operation of gear mechanisms.

Most studies show that more than 50% of tooth loss is caused by crooking.

In the reviewed article, the issue of investigating the causes of tooth decay and taking concrete measures to eliminate them was set.

When gears work, heat is released from the heat source along the contact area of the teeth.

Temperatures in the contact area of the teeth in contact increase by approximately 30–40% compared to non-functional areas of the tooth. This causes the working area of the teeth to become even hotter. Under the influence of compressive forces, the temperatures take part in the heating of the teeth again without being able to leave the surface.

To determine the distribution of temperatures in the contact areas of the teeth, a metallographic analysis was performed on the structure of the materials (Yakimov et al., 1974).

During the conducted studies, the heat source in the top part of the tooth is considered as a thin-walled part with a displacement speed of $V_z = 0.001/0.0005$ m/sec. and a thickness of 3.5/4.0 mm. For gears with module $m = 2$–2.5 mm, the temperature from the opposite side of the tooth has an effect on the operation of the tooth.

Studies show that the temperatures generated in contact areas exceed the temperatures of austenitic transformations.

The displacements of the heat source in the contact area of the teeth in gear mechanisms vary according to the sinusoidal law.

The temperatures generated at the base of the tooth are determined by the following formula (Pisarenko and Lebedev, 1976):

$$T = \sum_{k=0}^{k=n_1-1} 1.3\frac{q_1 a}{\lambda v_z}\cdot \int_{-y-H}^{-y+H} l^{\xi}\cdot K_0 \xi d\xi + \sum_{k=0}^{k=n_1-1} \frac{q_2 a}{\lambda v_z}\cdot \int_{y-H}^{-y+H} l^{\xi}\cdot K_0 \xi d\xi + \frac{q_2 a}{\lambda v_z}\cdot \int_{-2H}^{0} l^{-\xi}\cdot K_0 \xi d\xi \qquad (1)$$

$$y = \left[z-(n_1-k)L\right]-H$$

Here:

$z = n_1 \cdot L,\ n_1 = \dfrac{\sqrt{D_d \cdot 0.7m + (0.7m)^2}}{S}$ – is the number of teeth on the gear,

q_1 and q_2 – is the intensity of the heat flow generated in the transition from one tooth to another,
a – the heat transfer coefficient,
λ – the heat transfer coefficient,
v_z – the speed of propagation of the heat source in the head of the tooth,
n – the total number of heats,
S – the area of the width of the tooth.

The temperatures generated in the distribution circle of the tooth are determined by the following formula:

$$T=\sum_{k=0}^{k=n_2-1}1.3\frac{q_1 a}{\lambda v_{b\cdot a}}\cdot\int_{-y-H}^{-y+H} l^{-\xi}\cdot K_0\xi d\xi+$$
$$+\sum_{k=0}^{k=n_2-1}\frac{q_2 a}{\lambda v_{b\cdot a}}\cdot\int_{-y-H}^{-y+H} l^{-\xi}\cdot K_0\xi d\xi+\frac{q_2 a}{\lambda v_{b\cdot a}}\cdot\int_{-2H}^{0} l^{-\xi}\cdot K_0\xi d\xi \quad (2)$$
$$y=\left[z-(n_1-k)L\right]$$

$z=n_2\cdot L,\ n_2=\frac{\sqrt{D_d\cdot 0.5m+(0.5m)^2}}{S}$ v_{bottom} – gear rolling speed on the division circle.

The temperatures generated in the contact zone of the bottom part of the tooth are determined by the following formula:

$$T=\sum_{k=0}^{k=n_3-1}1.3\frac{q_1 a}{\lambda v_{bottom}}\cdot\int_{-y-H}^{-y+H} l^{-\xi}\cdot K_0\xi d\xi+$$
$$+\frac{q_2 a}{\lambda v_{bottom}}\cdot\int_{-2H}^{0} l^{-\xi}\cdot K_0\xi d\xi \quad (3)$$
$$z=n_3\cdot L,\ n_3=\frac{\sqrt{D_d\cdot h+h^2}}{s}$$

$h = 0.5$– 0.7 mm, for instrumentation $h=\sqrt{Dd\cdot t}$

The above equations allow measuring the temperatures in the contact areas of all types of gears.

Let's consider the formulas that determine the temperatures for gears with a contact cut at an angle of 15°.

The temperatures generated in the contact zone of the base of the tooth are determined by the following formula:

$$T=1.3\frac{q_1 a}{\lambda v_{top}}\cdot\int_{y_1-H}^{y_1+H} l^{-\xi}\cdot K_0\xi d\xi+\frac{q_2 a}{\lambda v_{top}}\cdot\int_{-2H}^{0} l^{-\xi}\cdot K_0\xi d\xi \quad (4)$$
$$y_1=6H$$

The temperature generated in the distribution circle of the tooth is determined by the following formula:

$$T=\frac{q_1 a}{\lambda v_{d.c.}}\cdot\int_{y_2-H}^{y_2+H} l^{-\xi}\cdot K_0\xi d\xi+\frac{q_2 a}{\lambda v_{d.c.}}\cdot\int_{-2H}^{0} l^{-\xi}\cdot K_0\xi d\xi \quad (5)$$

$$y_2=(2m+5h)\cdot\frac{v_{av}}{2a},\ v_{av}=\frac{v_{top}+v_{d.c.}}{2}$$

The temperatures generated in the root part of the tooth are determined by the following formula:

$$T=\sum_{k=0}^{k=n_3-1}\frac{q_1 a}{\lambda v_{bottom}}\cdot\int_{y_3-H}^{y_3+H} l^{-\xi}\cdot K_0\xi d\xi+$$
$$+\frac{q_2 a}{\lambda v_{bottom}}\cdot\int_{-2H}^{0} l^{-\xi}\cdot K_0\xi d\xi \quad (6)$$
$$y_3=(2.2m+2h)\cdot\frac{v_{av}}{a},\ v_{av}=\frac{v_{bottom}+v_{top}+v_{d.c.}}{3}$$

Equations that determine temperatures in the contact zone of gears with a contact profile used in devices:

$$T=1.3\frac{q_1 a}{\lambda v}\cdot\int_{y_4-H}^{y_4+H} l^{-\xi}\cdot K_0\xi d\xi+\frac{q_2 a}{\lambda v}\cdot\int_{-2H}^{0} l^{-\xi}\cdot K_0\xi d\xi \quad (7)$$
$$y_4=5H$$

Cracks are often formed on the side surfaces of the gear profiles due to friction. It is necessary to control the heat intensity of the process to eliminate the defects caused by the increase of temperatures in the contact areas of the gears during transmission. Therefore, the temperatures generated in the contact zone are expressed as follows (Yakimov et al., 1981; Yakimov, 2003):

$$T=\frac{qa}{\lambda v_c}\cdot\int_{-y-H}^{-y+H} l^{-\xi}\cdot K_0\xi^2 d\xi+\frac{qa}{\lambda v_c}\cdot\int_{-2H}^{0} l^{-\xi}\cdot K_0\xi^2 d\xi \quad (8)$$

$H=\frac{h\cdot v_c}{2a}$ – the dimensionless half-width of the source,

$q=\frac{P_z\cdot V_d}{S}$ – the heat flow intensity,

λ – the heat transfer coefficient,

$y=\frac{v_c}{a}\sqrt{D_d\cdot 2.2m+(2.2m)^2}$.

K_0 – the first type is the Bessel function.

From the results of the research, it was determined that the temperatures generated in the contour areas of the teeth exceed the temperatures of the austenite, which strengthens the tooth, and these temperatures reach 800–1300°C at the top and bottom of the tooth. The obtained results were confirmed as a result of metallographic studies. The percentage of carbon at the initial and final temperatures of martensite transformation has been determined (Figure 1).

With the percentage of carbon being $C = 0.8\%$, the starting temperatures for transformation to martensite are 250°C. Since the percentage of carbon is $C = 1.3\%$, the starting temperature of transformation to martensite is 100°C.

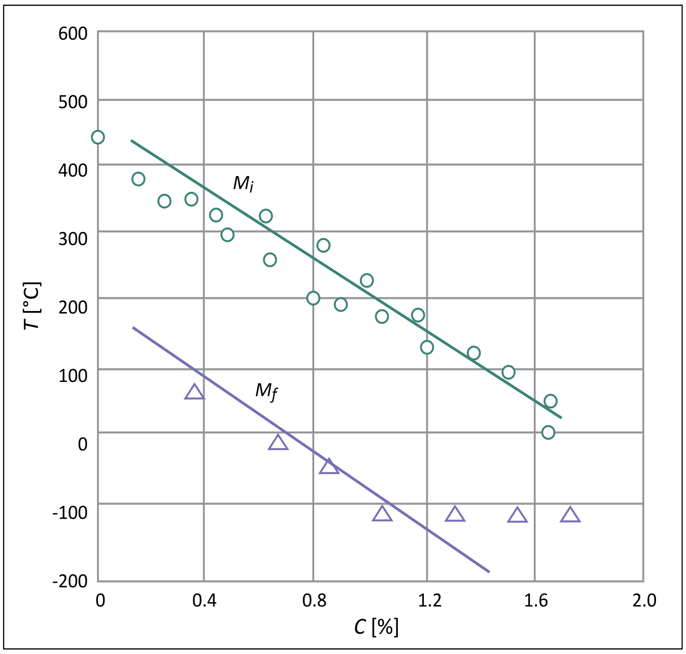

Figure 1. A graph showing the dependence of the initial (M_i) and final (M_f) temperatures of martensite transformation on the percentage of carbon

Rysunek 1. Wykres przedstawiający zależność początkowej (M_i) i końcowej (M_f) temperatury przemiany martenzytycznej od procentowej zawartości węgla

When the percentage of carbon is $C \approx 1.3$–1.4%, the initial temperatures in the transformation to martensite are 70–100°C. During the subsequent cooling stages, the temperature was lowered from 850° to 100°C and the formation of temporary thermoelastic stresses on the surface has been observed (Figure 2).

Temporary thermoelastic stresses increase up to the initial temperatures in the martensite transformation. After the martensite transformation point, the temporary thermoelastic stresses decrease and the process is accompanied by an increase in the volume of the surface material structure.

The following results have been obtained from the conducted theoretical studies.

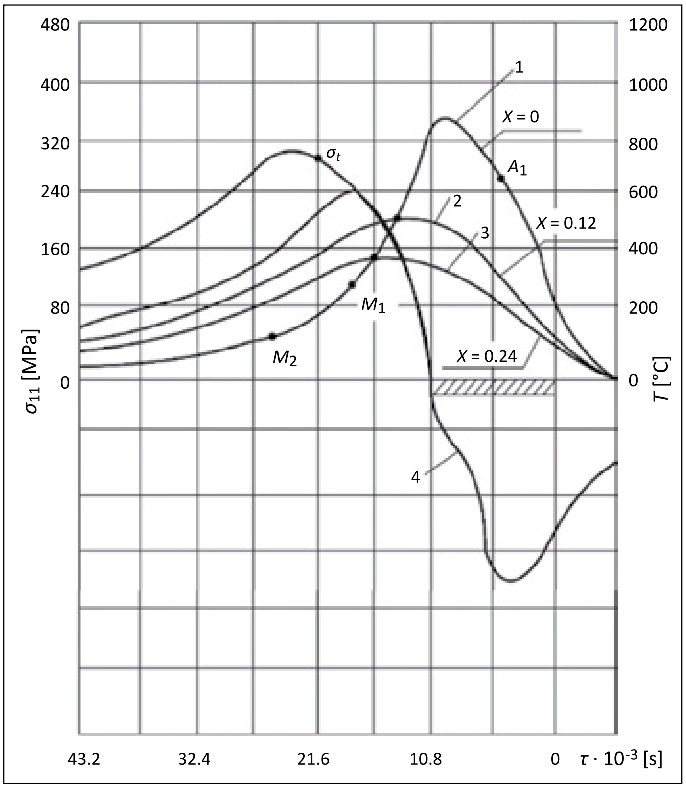

Figure 2. The nature of the temperature change on the contact surfaces of the tooth and the curve of the change of the temporary thermoelastic stresses during the cooling stages: 1, 2, 3, 4 – deep layers of the tooth; M_1, M_2 – martensitic transformation; A_1 – austenitic transformation; σ_t – temporary thermoelastic stresses during the teeth cooling period

Rysunek 2. Charakter zmian temperatury na powierzchniach styku zęba i krzywa zmian chwilowych naprężeń termoelastycznych podczas etapów chłodzenia: 1, 2, 3, 4 – głębsze warstwy zęba; M_1, M_2 – przemiana martenzytyczna; A_1 – przemiana austenityczna; σ_t – tymczasowe naprężenia termoelastyczne podczas fazy chłodzenia zębów

Conclusions

1. The cause of damage on the working surface of the teeth is the variable contact stresses and the friction created in the profiles of the teeth.
2. It was determined that the areas of microcracks in the tooth layer between the touching surfaces are disintegrated and that corrosion occurs between the touching profiles due to the effect of high pressure and temperatures.
3. Deriving of the equations that determine the temperatures generated at the top, division part, and root of the tooth have been determined.
4. To prevent the top layer of the tooth from being worn, it is considered appropriate to use special anti-rubbing oils with the inclusion of high viscosity and chemical substances.

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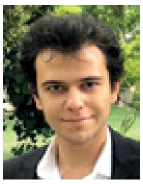

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