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Қ. И. Сәтпаев атындағы Қазақ ұлттық техникалық зерттеу университеті

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# ИЗВЕСТИЯ

НАЦИОНАЛЬНОЙ АКАДЕМИИ НАУК
РЕСПУБЛИКИ КАЗАХСТАН
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# THE COMPARATIVE ANALYSIS OF COMPUTER MODEL OPERATION OF TOOTH GEARING WITH EVOLVENT GEARING AND NOVIKOV'S GEARING

**Abstract.** In the article settlement sizes of the maximum contact tension by techniques of various authors taking into account solutions of Hertz are specified, researches on comparison of load ability of elements of the gears with evolvent gearing applied in the drives of ball mills and gears with Novikov's gearing are considered. The numerical finite element models (FEM), calculated tooth gearing and boundary conditions and also calculation the stress-strain state of FEM by means of the programs MSC/Patran, MSC/Nastran are developed.

As a result of research the load ability of Novikov's gearing from conditions of contact endurance of active surfaces of teeths which is higher, than for gears with evolvent gearing, with the same overall dimensions, owing to the big specified radius of curvature of the contacting teeths is established.

**Keywords:** tooth gearing, Novikov's gearings, evolvent gearing, tooth module, finite element model, deformation, tension.

The efficiency and economic indicators of modern machines in various industries depend to a considerable extent on the performance of gear drives. Creation of tooth gearing with high performance criteria ensures the improvement of not only drives, but also machines in general, and this is relevant for modern mechanical engineering. One of the ways to improve gear drives is the development of tooth gears with increased load capacity and a long service life.

The creation by Michael Leontievich Novikov [1-3] in the middle of the 20th century of the original gearing system as an alternative to evolvent gearing was revolutionary character, accompanied by a surge of large-scale research and large investments, multifaceted industry projects and demonstrative examples of industrial implementation. One of the most significant advantages of Novikov's gearings was an extraordinarily large margin for contact resistance of the teeth and, as a result, high constructive flexibility [1].

There is a number of works on a comprehensive evaluation of Novikov's gearing, in the experience of theoretical research and practical results of using these gearings in the industry was generalized [4].

Comparative estimates of Novikov's gearings with evolvent gearing are most often based on understating evolvent gearing indicators and overvaluation of Novikov's gearing parameters. Kinematic principle of pointwise spatial engagement M.L. Novikov is based on axial intermating of the teeth (with the failure of the conditions for the end overlap with point conjugation of the end profiles of the interacting teeth and the approximation of the centers of their curvature to the instantaneous axis of rotation) and proceeds from the conclusions [3], which are presented in the following form:

- conclusion No. 1 - on the independence of the contact strength of the teeth at the linking pole from the shape of the tooth profiles and from the gearing system;

- conclusion No. 2 - about the absence of the possibility of a noticeable decrease in contact stresses in gears with an initial-linear touch of the teeth;
- conclusion No. 3 - about a particularly favorable (with a tenfold increase in the thickness of the oil layer) hydrodynamics of contact lubrication of Novikov's gearing;
- conclusion No. 4 - about the possibility of a significant reduction in contact stresses exclusively in the pointwise non-polar Novikov's gearing.

All the conclusions are completely justified within the framework of the accepted physical basis, in which even theoretically Novikov's point contact was initially modeled by the compression of parallel elastic cylinders-their radiuses are equal to the radiuses of the normal curvature of the screw contact lines of the interacting teeth.

The initial solution of the contact problem was proposed by Hertz for two possible variants of the initial contact: point and linear.

As is known in Hertz theory, the following calculated assumptions are accepted [5]:

1) The contacting bodies are smooth and uncoordinated (smoothness means no risks, scratches, defects in the contact zone, and inconsistency is a difference in the shape of the profiles of the contacting bodies).

2) The compressive force is normal to contact areas and its line of action passes through the centers of sections of curvilinear surfaces

3) Only normal pressures act within the contact area, while frictional forces are neglected

4) The dimensions of the contact pads are small in comparison with the surfaces of the contacting bodies and the radii of their curvature

5) Only elastic deformations occur in the contact zone, and the material of the bodies satisfies Hooke's law.

The result of solving the problems of Hertz depends in principle on the type of initial contact.

For a point contact, we have the following basic relationships:

$$p_m = \left[P\left(\frac{1}{R_1}+\frac{1}{R_2}\right)/\left(\frac{1}{E_1}+\frac{1}{E_2}\right)\right]^{\frac{1}{2}}, \tag{1}$$

where $R_1$, $R_2$ - the radiuses of the contacting cylinders, $P$ - the normal force of compression arising under compression, $E$ - the modulus of elasticity.

The greatest contact stress $\sigma_H$ is determined by the formula

$$\sigma_H = \frac{3}{2}\cdot\frac{P}{\pi ab}, \tag{2}$$

where $a$, $b$ - semiaxes of contact ellipses.

It should be noted that assumptions 1 and 2 suggest the inapplicability of classical Hertz formulas for point contact in Novikov's gearing, since at the tooth's beginning, the tooth profile of the gear wheel and the cogwheel has a geometrically consistent shape (figure 1) [6].

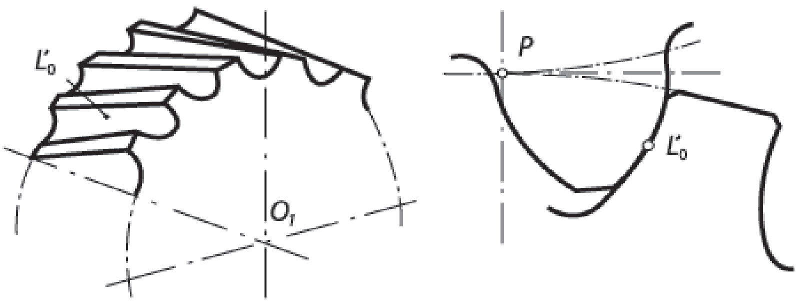



Figure 1 – Novikov's gearing with a convex concave butt-end profile of the tooth

The failure of the theory in the classical form is confirmed by the calculated values of $\sigma_{max}$, found by the methods of various authors developed with allowance for the Hertz solutions given in table 1 [1].

Table 1 – Settlement values of the maximum contact tension by techniques of various authors

| Determined values | Information sources | | | | |
|---|---|---|---|---|---|
| | Hertz [7] | Kovalev M.N. [5] | Makushin M.I. [6] | VNIINMACH [1] | IMACH [3] |
| $\sigma_{\max}$, МПа | 9771 | 3843 | 9818 | 4473 | 3206 |
| $\varphi_\sigma = \frac{\sigma_{\max}}{\sigma^*_{\max}}$ | 14,00 | 5,50 | 14,06 | 6,41 | 4,60 |
| $\varphi_F = \varphi_\sigma^3$ | 2743 | 167 | 2783 | 263 | 97 |

From table 1 it follows that the calculated values of the stresses are approximately 14 times higher than the voltage $\sigma_{max}$ = 698, taking place in Novikov's gearing. The indicated increase of $\sigma_{max}$ corresponds to a decrease in the loading ability of the Novikov's gearing by contact stresses.

To compare the physical nature of the contact in evolvent gearing and Novikov's gearing, we refer to the experimental data of the author M.I. Sakhanko [7], dealing with the issues of contact endurance of steels, depending on the geometric parameters of the dimensions of the contacting bodies. The author introduces the concept of the material's resistance in the contact zone and provides a formula for determining the contact voltage with allowance for the coefficients $\vartheta_1, \vartheta_2$- coefficients that take into account the curvature of the contacting bodies at the point of their mutual contact

$$\sigma_{max} = \frac{4100}{\vartheta_1 \cdot \vartheta_2} \sqrt{P \left(\sum_\rho \frac{b}{a}\right)^2}, \quad (3)$$

where $P$ - load, $\sum_\rho \frac{b}{a}$ - the sum of the curvature of the contacting bodies. Experiment was established for a circular contact $\frac{b}{a} = 1$ composition $\vartheta_1 \cdot \vartheta_2$=1, and for elliptic surfaces the pressure $\frac{b}{a} = 0{,}05$, for surfaces close to a linear contact, the product $\vartheta_1 \cdot \vartheta_2$=2 (figure 2).

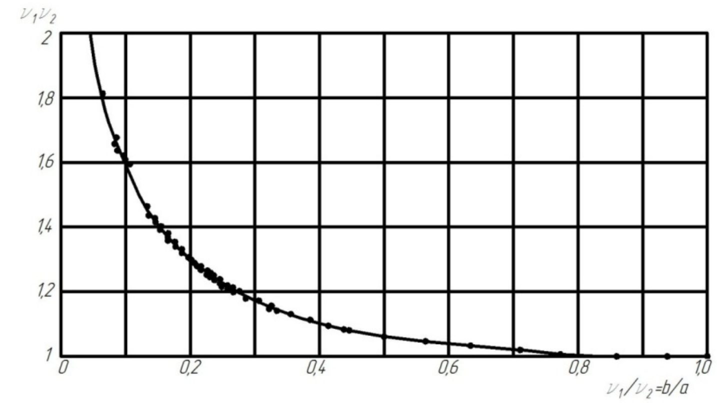



Figure 2 – The interrelation between the product of the coefficients $\vartheta_1, \vartheta_2$ and their ratio $\frac{\vartheta_1}{\vartheta_2} = \frac{b}{a}$

"Along the way," the author notes that when compressing hardened balls, the material in the zone of the circular pressure surface becomes stronger until the fracture at a coefficient $\frac{P}{D^2}$ = 50, while at compression of hardened rollers the material in the zone of linear contact already at the loading factor $\frac{P}{Dl}$ = 10-12 ($l$ - length) of the roller is in a state of fluidity.

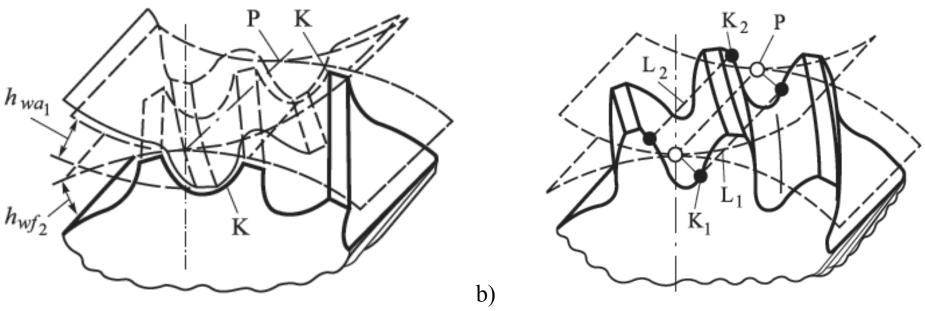
It is known that in the mutual deformation of contacting bodies, the material on the pressure surface is in a state of three-dimensional compression which characterizes three main stresses. If the contact is circular then volumetric compression on the pressure surface into linear compression, at this point the two principal stresses are zero and the contacting surface is the main, and from the theory of elasticity it is known that on the principal areas tangential stresses $\tau_{ij} = 0$. The most favorable slip of the parts of the crystal lattice or dislocation is determined by the value $\alpha = 45^0$ for which $\tau_{max} = 1/2\sigma$ [8]. This argument speaks in favor of Novikov's point gearings.

Advantages and disadvantages of Novikov's gearings:

- a helical cylindrical transmission with lines or close to the contact lines of the cogwheel, in which the convex surfaces of the initial teeth heads interact with the concave surfaces of the initial legs of the teeth and the coefficient of end overlap equal to or close to zero. Approaching a linear contact is provided by slightly less curvature of the profile of the concave surface of the tooth in comparison with the curvature of the convex surface of the profile of the conjugate tooth.

Smoothness of work is achieved due to axial overlap, the coefficient $\varepsilon_a$ which is chosen to be greater than unity.

There are Novikov's gearings from one (*scheme a*) and two (*scheme b*) lines of gearings.




a) b)

Figure 3 – The scheme of contact in Novikov's gearings

In *scheme a* the notation: *K* - the contact point moved translationally during the transmission operation ( *K* moves along the path parallel to the pole line, *P* - the contact line of the initial cylinders); $h_{wa1}$ and $h_{wf2}$ - respectively the height of the initial head of the tooth gear wheel and the height of the initial foot of the tooth cogwheel.

In *scheme b* the notation: *P* - the pole line; $K_1$ and $K_2$ - contact points, respectively, on the stem and head of the tooth; $L_1$ and $L_2$ - the gearing lines-the trajectories of the contact points $K_1$ and $K_{2,}$ respectively.

The contact points on the same gearing lines are moved one after another at an interval denoted by $q_{21}$. The spacing between two contact points on different gearing lines $q_{22}$ is the smallest distance between two end sections of the conjugate cogwheel drawn through contact points of the same surfaces of two adjacent cogwheel's teeths.

Apply Novikov's gearing with two lines of gearing. They have teeth with convex surfaces of the initial heads and concave surfaces of the initial legs. The teeth of the gear wheel and cogwheels can be cut with one tool, in contrast to Novikov's gearing with one gearing line.

The geometric calculation of the Novikov's gearings with two lines of gearing is performed depending on the parameters of the *initial contour* [7].

Calculation of geometric parameters is performed in accordance with GOST 15023-76 in figure 4 shows the profile of gearing [6].

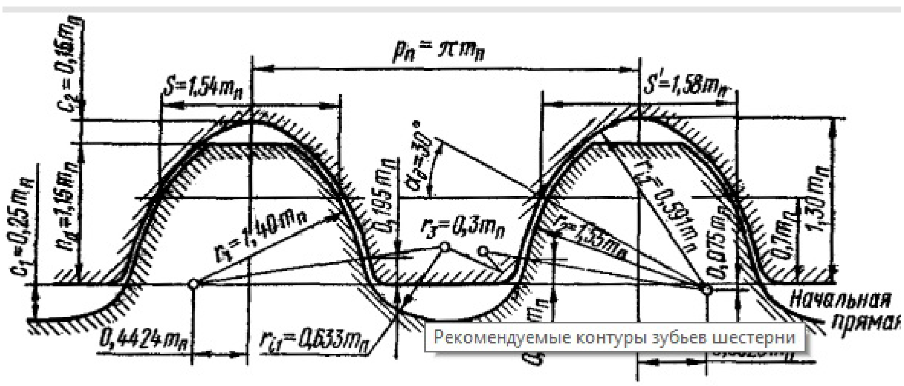



Figure 4 – Profile of Novikov's gearing

Dividing cogwheel diameters:

$$d_1 = \frac{mz_1}{cos\beta}; \quad d_2 = \frac{mz_2}{cos\beta}, \tag{4}$$

where $z_1$, $z_2$ - number of teeth of cogwheels, $m$ - module, $\beta$ - angle of inclination of the tooth line ( $\beta$ = 10 ... 22° in helical teeth and $\beta$ = 20 ... 30° in chevron gearings). Axial step $P_x$= π $m$ / sinβ. Width of a ring gear of a cogwheel $b_2$= (1 ... 1, 2) $P_x$ or $b_2$= (2 ... 2.2) $P_x$ or $b_2$= (3 ... 3,3) $Px$ or $b_2$= (4 ... 4.4) $P_x$ (with such values, the maximum transfer capacity is provided). Width of a ring gear of a wheel $b_1$ = $b_2$ + (0.4 ... 1.5) $m$.

Center distance

$$a = 0{,}5\ m(z_1 + z_2) / \cos\beta. \tag{5}$$

Diameters (circles) of vertices

$$d_{a1} = 2h_a; d_{a2} = d_2 + 2h_a. \tag{6}$$

Diameters (circles) of depressions

$$d_{r1} = d_1 - 2h_a - 2c; d_{r2} = d_2 - 2h_a - 2c. \tag{7}$$

Where $h_a$ and $c$ – the initial contour.

Novikov’s gearing is used for hardness of tooth surfaces H ≤ HB 320 (3200 MPa); module $m \leq 16$ mm; the circumferential velocity $v \leq 20$ m/s. The load capacity of Novikov’s gearing from the condition of contact endurance of the active surfaces of the teeth is 2 times higher than for gears with evolvent gearing with the same overall dimensions, due to the greater reduced radius of curvature of the contacting teeth. But since Novikov’s gearing are used with low hardness of the surface due to technological difficulties of grinding the teeth, then by criteria of contact strength and endurance they can be inferior to gears with evolvent gearing. The strength of the teeth in bending is approximately 5 ... 15% lower than for gears with evolvent gearing.

Novikov’s gearing has a higher efficiency due to the rolling of the teeth without the geometry of sliding, but is sensitive to changes in the interaxial distance. Manufacturing error and/or deformation of the shaft/supports lead to differences in the gearing edge and the main transmission properties with involute gearing. Axial movement of the point contact in the Novikov’s gearing leads to a change in the reaction of the shaft support during one cycle of gearing and, accordingly, can cause additional vibration. For these reasons, Novikov's gearing requires a high precision of manufacture and high rigidity of shafts and supports [7].

To compare the load capacity of the gearing elements with evolvent gearing used in the drives of ball mills and gears with Novikov’s gearing, a numerical finite element method is used.

Work on the calculation of the ring gear of the ball mill drive was carried out in the following volume and sequence:

1. Drawing up of the design scheme corresponding to the test loading. Purpose of loading regimes .

2. Using the MSC/Patran finite element program model (hereinafter referred to as "KEM") of the design to be calculated, boundary conditions corresponding to the pinning of the gear in the experimental rig, as well as the external load model.

4. Carrying out calculation of the stress-strain state (hereinafter – VAT) of the CEM with the help of the SAE program. The programs MSC/Patran, MSC/Nastran were used .

The creation of the finite element model (KEM) of the ring gear, the external load model and the boundary conditions simulating the fastening of this cogwheel to the drum, the following sequence was performed.

Creating 3D model (figure 5) was carried out in CAD program KOMPAS module Shaft on the geometric parameter am calculated in the paper.

When creation of the calculation model, a fragment of the cogwheel was cut out, in order not to waste the resources of the machine, since the stresses will be localized over the area of the crown's tooth of a ring gear.

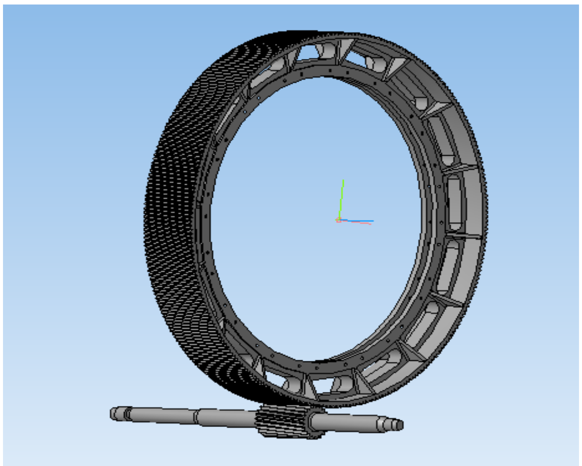


Figure 5 – 3D model of assembly of a cogwheel and gear wheel

The model created in the Russian CAD system was successfully imported through the exchange of files in the CAE system NASTRAN. Partitioning into finite elements is modeled by volume elements of the type tet. The calculation scheme for determining the operating voltages is shown in figure 6. The requirements for the type of elements and the quality of the grid should be increased, since for calculating the fatigue strength it is necessary to calculate the local stresses taking into account their concentration.

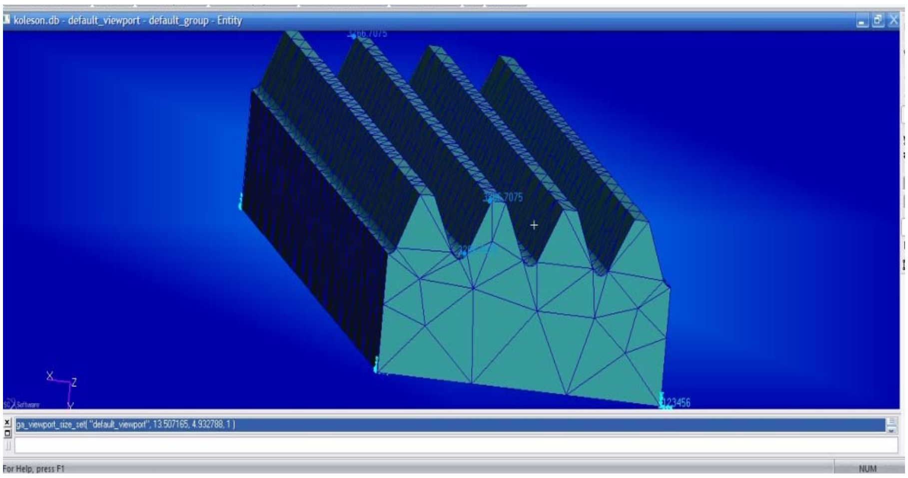



Figure 6 – The generated grid

Taking into account the specific features of the design, the calculated loading scheme is shown in figure 7.

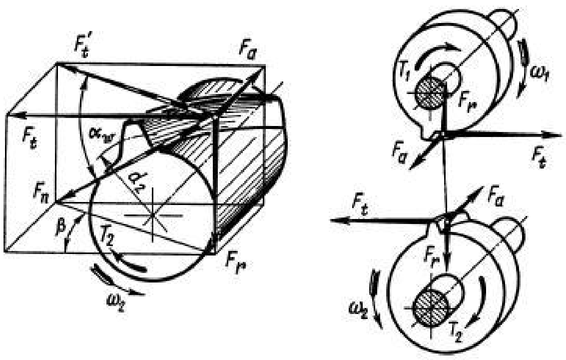



Figure 7 – Scheme of loading

Data on the forces acting in the gearing ring gear and gear shaft were calculated [9, 10].
- circumferential force: $F_t$ = 5362315 H;
- radial force in the gearing: $F_t$ = 195,972 H;
- Axial force: $F_a$ = 49333 H.

The mechanical design scheme included a rigid anchorage, presupposing a superposition of the coupling along six degrees of freedom and loading as a resultant distributed load along the tooth edge. The axle load module was calculated using the following formula:

$$q_x = \frac{F_t}{s} = \frac{536231}{1.9141} = 280114\frac{H}{м}, \tag{8}$$

$$q_y = \frac{F_t}{s} = \frac{195972}{1.9141} = 102388\frac{H}{м}, \quad q_z = \frac{F_\alpha}{s} = \frac{49333}{1.9141} = 25773\frac{H}{м}.$$

Thus, for definition by means of the MSC/Nastran program of a ring gear the static problem (figure 8) was solved. The stress calculation results are saved in one file with the extension*. xdb.

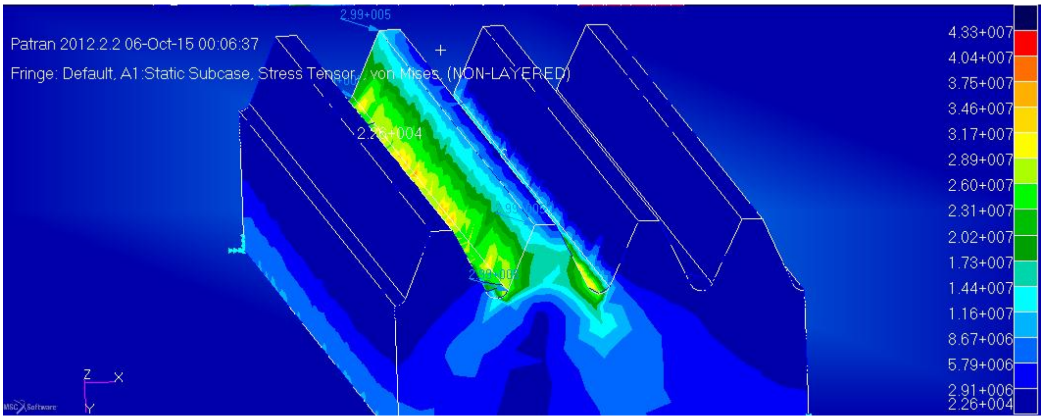



Figure 8 – Distribution of equivalent tension in a ring gear

After the program MSC/Nastran has successfully completed the solution of the problem "coleso.bdf ", the results of calculation for each step of the load application stored in one file "coleso.xdb" will be available for estimating the longevity. The distribution of equivalent stresses, calculated by the Mises formula. The voltage scale in Figure 8 corresponds to the dimension [Pa].

The stress results show a sufficient supply of static strength. Results in the diagram has a value of maximum stress $\sigma_{max}$ = 44,3 MPa allowable stress $[\sigma]$ = 600 MPa, the strength of the static coefficient is determined by the formula:

$$k = \frac{[\sigma]}{\sigma} = 1{,}72 \tag{9}$$

The stress distribution pattern shows good resistance to the bending of the profile teeth with evolvent stress.

However, it is known from practical observations that the failure of wheels occurs not because of an insufficient coefficient of static strength, but because of surface disruptions such as chipping, spalling, etc.

Fracture of the tooth occurs in the zone of maximum stress (at the fillet of the tooth) along the longitudinal section and has a curvilinear transverse contour which, depending on the type of load and heat treatment, can be either concave or convex. The fatigue fracture has a concave profile, that is, directed into the body of the wheel. For cogwheels improved or volume-hardened - without a step, for cogwheels with surface hardening with a step.

Statistical fracture, which occurs less frequently, has a convex contour, also without a step and with a step. In the case of surface hardening, the initial crack at the depth of the hardened layer is formed from the action of normal stresses, and further damage occurs under the influence of tangential stresses.

A fatigue fracture is preceded by a fatigue crack, which begins to form on the loaded side at the base of the tooth, more often on the edge of the butt; Further, the crack develops along the tooth's leg along the normal to the transition curve in the direction of the compressed side.

The pattern of stress isolines confirms these conclusions - the places of stress concentration at the base of the tooth are visible (figure 9). To assess the level of shearing stress influence on the fatigue strength of the tooth, it is advisable to estimate the values of the stress tensor components, which allows us to make the NASTRAN program.

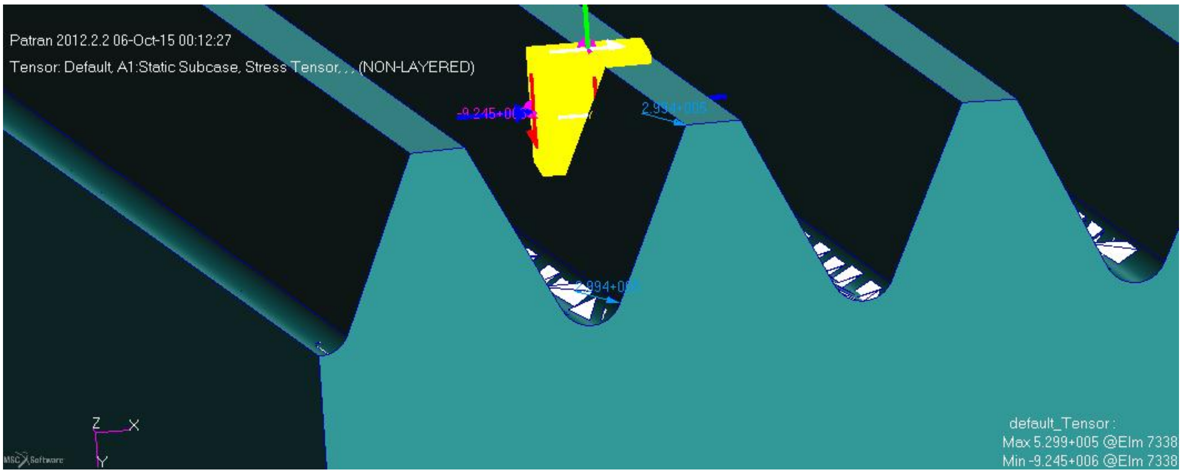



Figure 9 – Elementary volume on the tooth surface

An elementary volume was extracted from the tooth surface and all components of the stress tensor were visualized.

Figure 10 shows the components of the stress tensor in the elementary volume of the evolvent gearing tooth.

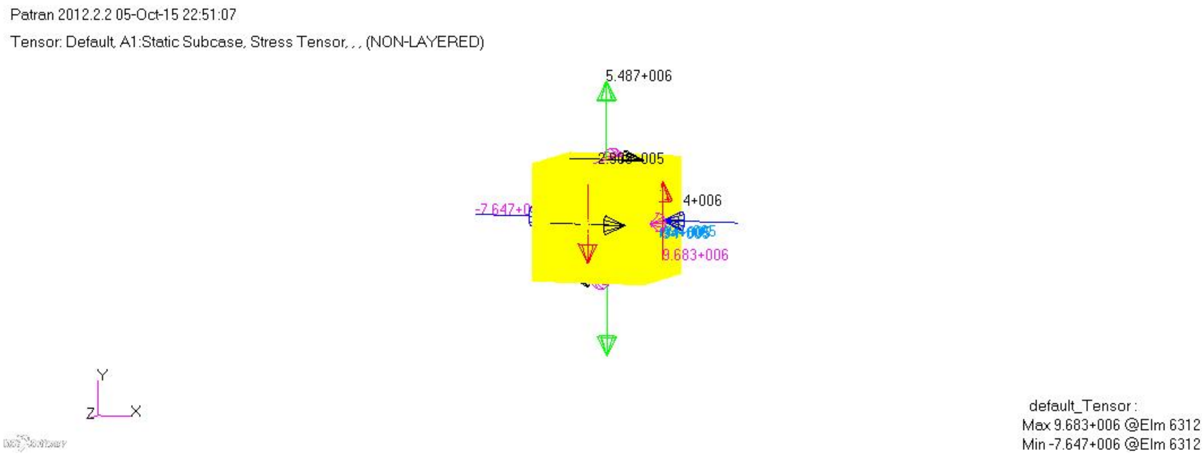



Figure 10 – Components of a stress of tension in the elementary volume of tooth of evolvent gearing

The diagram shows that the shearing stresses $\tau_{ig}$ on the areas of the linear contact has values greater than the values of normal stresses $\sigma_i$, (perpendicular vectors). It is known that macrocracks on contacting surfaces develop under the action of tangential stresses.

On the surface of the fatigue fracture, more often because of the small size of the tooth, one can observe not five, but three zones of development of the crack:

- smooth, where the origin of the crack and its slow growth occurred;
- Rough , where crack growth accelerates;
- rough , where there was a brittle fracture of the tooth.

The strained state in the dangerous section of the tooth depends on the geometric parameters of the tooth.

In order, to evaluate the load capacity and resistance to surface damage wheels with Novikov's gearing, according to GOST 15023-76 were counted geometric parameters of the teeth and solid model constructed in KOMPAS Russian program.

Figure 11 presents the 3D gearing model of a part of the Novikov's gearing of cogwheel.

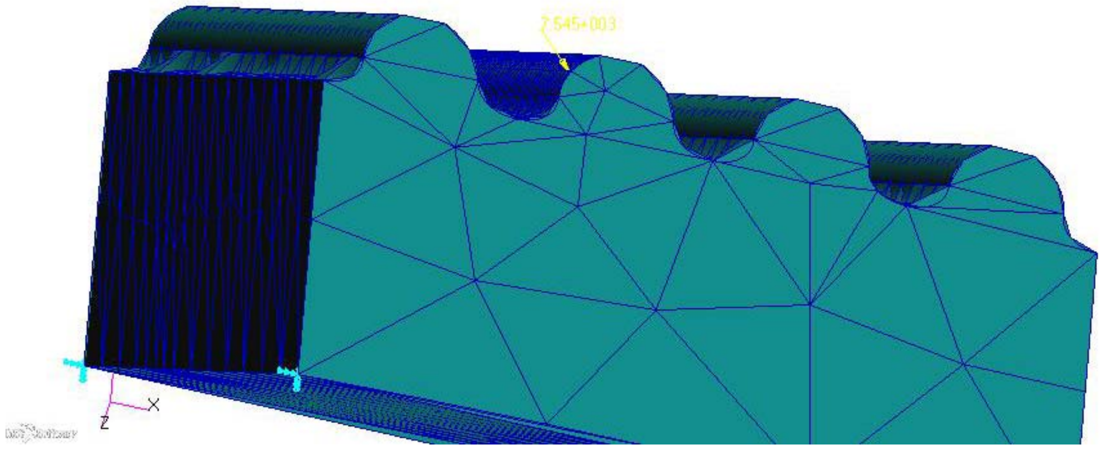



Figure 11 – 3D model of gearing of a part of a cogwheel of Novikov's tooth gearing

The load on the tooth was simulated as being equal in absolute value $F_n = \sqrt{F_t^2 + F_r^2 + F_a^2}$ and directed perpendicular to the tangent to the profile of the tooth, applied at the point of contact of the Novikov's gearing.

The fastening was modeled as a five-way link and a rotation around the axis Z.

The static calculation parameters were selected and the following results were obtained (figure 12).

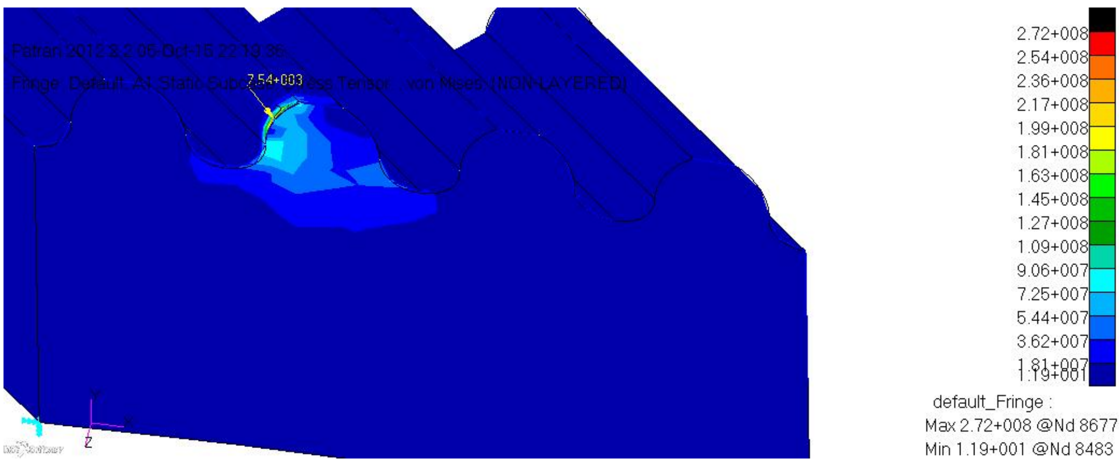



Figure 12 – Stress distribution diagram

The diagram shows that the limit of the maximum equivalent stresses is much higher in the Novikov's gearing than in the evolvent gearing.

$\sigma_{max}$ = 272 MPa for the Novikov's gearing

$\sigma_{max}$ = 44.3 MPa for gears with evolvent gearing

Which explains the best resistance of deformation to the bending of the evolvent teeth.

To assess the possibility of plastic deformations and physical causes of tangential stresses at the contact site, the capabilities of the NASTRAN program were used, which allows to decompose the stress tensor into components and visualize this result.

Figures 13 and 14 show the elementary volume on the surface of the tooth of cogwheel the Novikov's gearing and the components of the stress tensor.

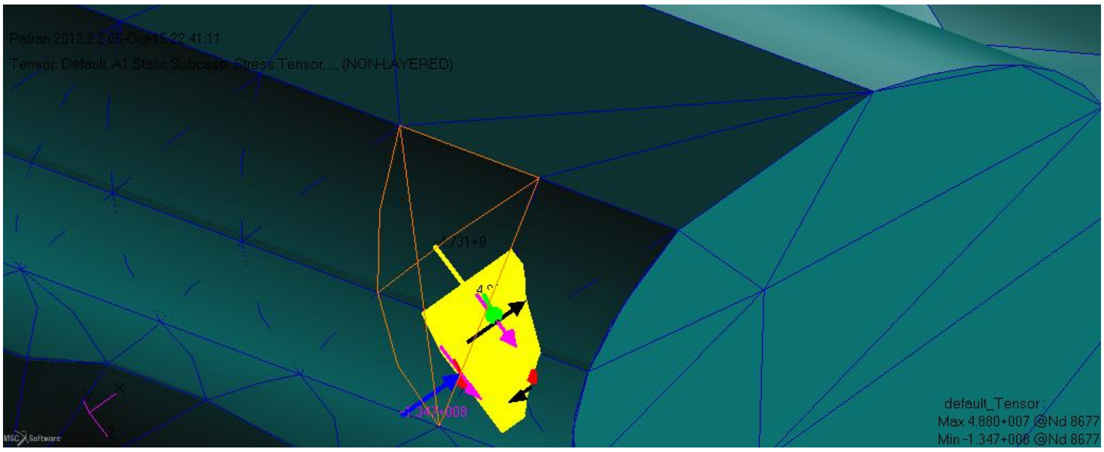



Figure 13 – Elementary volume on the surface of tooth of a cogwheel of Novikov

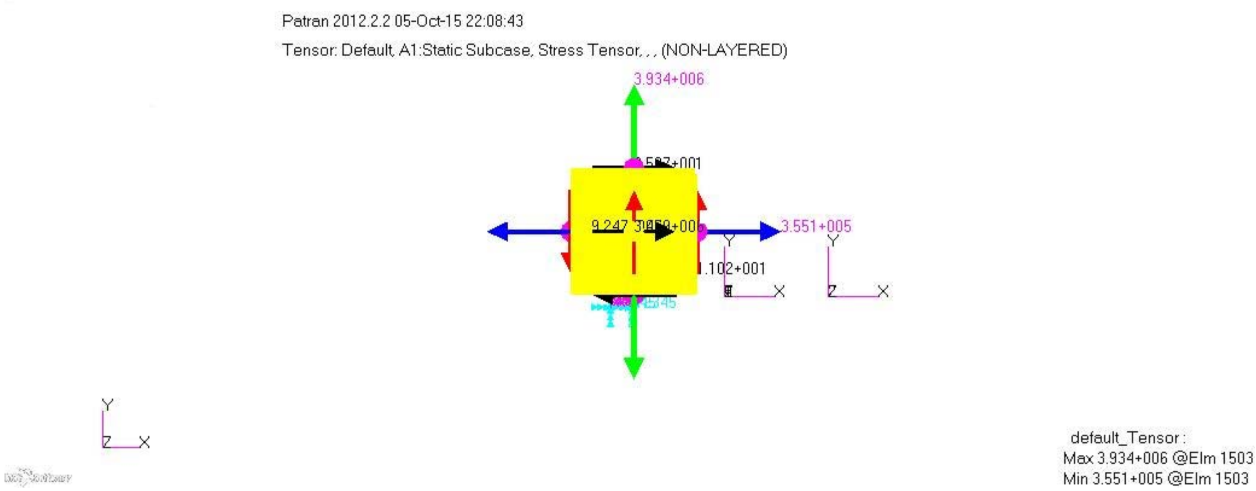



Figure 14 – Components of the stress tensor

The results show that tangential stresses located in the contact plane have very small values. Indeed, in case of point contact or contact of bodies with approximately equal radiuses of curvature of the body, they undergo uniaxial compression where the tangential stress should be absent on the main areas, ie, perpendicular to the force vector, which is approximately reflected in the results of the program, the value of $\tau_{xy} = 10$ Pa.

Based on a comparison based on computer modeling, we can draw the following conclusions:

- the load capacity of Novikov's gearing from the conditions of contact endurance of the active surfaces of the teeth is higher than that of gears with evolvent gearing, with the same overall dimensions, due to the large reduced radius of curvature of the contacting teeth;

- the strength of the teeth of Novikov's gearing is lower in bending than in gears with evolvent gearing.

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## ЭВОЛЬВЕНТТІ ІЛІНІС ЖӘНЕ НОВИКОВ ІЛІНІС ТІСТІ БЕРІЛІСТЕРІН КОМПЬЮТЕРЛІК МОДЕЛЬДЕУ АРҚЫЛЫ САЛЫСТЫРМАЛЫ ТАЛДАУ

**Аннотация.** Мақалада Герц шешімдерін ескере отырып әртүрлі авторлардың әдістеріне сәйкес максималды байланыс кернеулерінің есептелген мәндері берілген, шарлы диірмендер мен жетектерге арналған Новиков ілініс тісті берілісімен пайдаланылатын беріліс элементтердің жүктемелік қабілетін кернеуге салыстыру арқылы зерттеу жүргізілді. MSC/Patran, MSC/Nastran бағдарламаларының көмегімен есептелінетін тісті берілістің шекаралық жағдайда сандық соңғы элементтердің моделі (СЭМ), сондай-ақ СЭМ кернеулі-деформацияланған күйін есептеу жүргізілді. Зерттеу нәтижесінде белсенді тіс беттерінің контактілерге төзімділігі жағдайында Новиков ілiнiстің жүктемелік қабілеті анықталды, бұл байланыстағы тістің келтірілген үлкен қисықтық радиусы салдарынан бірдей жалпы өлшемдерімен эвольвентті тісті берілістерге қарағанда жоғары болып табылды.

**Түйін сөздер:** тісті беріліс, Новиков ілінісі, эвольвентті ілінісі, тіс модулі, соңғы элементтік моделі, деформация, кернеу.

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## СРАВНИТЕЛЬНЫЙ АНАЛИЗ КОМПЬЮТЕРНОГО МОДЕЛИРОВАНИЯ ЗУБЧАТЫХ ПЕРЕДАЧ С ЭВОЛЬВЕНТНЫМ ЗАЦЕПЛЕНИЕМ И ПЕРЕДАЧИ НОВИКОВА

**Аннотация.** Приведены расчетные величины максимальных контактных напряжений по методикам различных авторов с учетом решений Герца, рассмотрены исследования по сравнению нагрузочной способности элементов передач с эвольвентным зацеплением, применяемых в приводах шаровых мельниц и передач с зацеплением Новикова. Разработаны численные конечно-элементные модели (КЭМ), рассчитываемой зубчатой передачи и граничных условий, а также расчет напряженно-деформированного состояния КЭМ с помощью программ MSC/Patran, MSC/Nastran. В результате исследовании, установлена нагрузочная способность передачи Новикова из условий контактной выносливости активных поверхностей зубьев, которая выше, чем у передач с эвольвентным зацеплением, с теми же габаритными размерами, вследствие большого приведенного радиуса кривизны контактирующих зубьев.

**Ключевые слова:** зубчатые передачи, зацепления Новикова, эвольвентное зацепление, модуль зуба, конечная элементная модель, деформация, напряжения.

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