Computer-aided design and adoption of standard software on gearing

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Abstract. New automated systems for designing gear drives and adoption of standard computer software on gearing are considered. A review of different software solutions for designing gears, gear drives and gear forming tools is presented. Methodological aspects and numerical results of designing injection-moulded plastic gears and forming mould dies for their manufacturing are shown, including the techniques of tooth profile construction and adjusting gear accuracy parameters. The possibilities of a two-level calculation procedure for strength and wear of composite gears made of dispersion-filled plastics aimed at optimization of reinforcing filler content are noted. The results of application of known standard automated systems, such as IOSO, KIMoS and other for quality improvement of the front loader gearbox and transmissions of “Belarus” tractor are discussed.

1. Introduction

The article presents the results of development of software intended to automate the gear drive and gear forming tool calculation for manufacturing gears, including injection molded plastic ones. The developed computer programs provide for optimization of tooth meshing by the criterion of contact ratio maximization; designing the gear cutting tool with modified tooth profile; designing plastic cylindrical gear drives and computation of forming tools (mold dies) for manufacturing plastic gears.

Currently, the market of CAD/CAM/CAE systems offers a wide variety of software products of various levels for different purposes. Well known “universal” software products, such as Pro/Engineer, Unigraphics, CATIA, EUKLID, I-DEAS, ANSYS, T-ELEXCAD, APM WinMachine, Solid Works, Compas, AutoCAD, have special modules which provide for computing various types of gear wheels and drives.

APM WinMachine system (offered by Research and Software Development Center APM Ltd., Russia) is equipped with APM Trans module which allows to perform design and verification calculations of the gear drives by meshing (external and internal meshing cylindrical gear drives, external meshing helical gears, herringbone, bevel gears with straight and circular arc tooth, worm gears) [1]. Special software designed for gear drives calculation are based primarily on standard
calculations according to the National (DIN 3990, AGMA 610) and International (ISO 6363) standards and may have additional features specific to their use. Programs [2, 3] supplement AutoCAD, Compas, etc. are offered to calculate geometry, strength and reliability of external and internal meshing cylindrical spur and helical gear drives, bevel gears with spur and circular arc teeth, cylindrical worm gears.

The concept, presented in [4, 5] uses the modules invariant to the gear drive type and support the process of drive designing, manufacturing and operation. The software library includes 3D modeling and interactive visualization; gear drive scheme module; gear cutting process; tooth contact modeling; loaded tooth contact analysis; simulation of pinion and gear run-in which are based on the results of measurements of real teeth surfaces; predicting the gear drive condition (wear, noise, vibration activity).

Some software use the programs directly related to designing the plastic gears (in particular, such well-known programs as KISS Soft [6], StarGear [7], “Plastic Gearing” [8]).

A set of basic racks (4 modifications with various tooth heights) proposed by ABA-PGT Inc. [8] assures equal-sized thickness at the tooth dedendum of a pair of mating gears and modification of the profile at the top of tooth, excludes the teeth undercutting and decreases bending stresses owing to completely rounded tooth space; the software system is also available [8]. The basic rack parameters are standardized [9].

Methods and programs for calculation of load carrying capacity of the gear drive by safety criteria along with determination of safety factors for the tooth fatigue fracture, contact endurance, wear resistance, melting temperature of plastic gear material under long-term continuous operation, as well as for lubricant temperature are described (by the example of metal-polymer worm drive with plastic helical gear) in works [10, 11]. In VDI 2736 specifications software for automated computing worm drives with helical plastic worm wheels is proposed too [12].

The authors [13, 14] present a variant of gear drive calculations based on generalized parameters under the brand name of Direct Gear Design supplemented (for plastic gears) with the Genetic Mold Solution concept, providing a step-by-step attainment of required gear accuracy. The AKGears-developed software provides for calculation of various types of gear drives with the optimal shape of teeth and form of fillet by the criterion of minimal bending stresses, as well as drives with asymmetric teeth profiles assuring the increased load carrying capacity.

The aim of this paper is to review the results obtained by the authors in PC-aided design and adoption of known standard software in the field of gears, gear transmissions and gear forming tool.

2. Original PC-aided design of injection molded cylindrical plastic gears

2.1. Software subject and contents

The program is developed for PC-aided designing involute cylindrical drives with plastic gear wheels, basic rack as per GOST 13755-81 at $m > 1$ mm and GOST 9587-81 at $m < 1$ mm [15, 16].

The geometrical parameters limit values are: module, $m = 0.1 \div 10$ mm; reference diameter, $d$ – up to 400 mm; center distances, $a_w$ – up to 250 mm at $m > 1$ mm and up to 180 mm at $m < 1$ mm.

In the design gear drives the program is used the following calculation modes: calculations of nominal dimensions and geometrical parameters of gear wheels; accuracy grade and backlash norms; quality of engagement is evaluated; animated visual control over the gear wheel meshing process is implemented; full calculation protocol is generated; data for drawings are outputted; contact strength and bending fatigue strength are determined; calculation of gear forming mold dies is performed which results are used as the data for their manufacturing by CNC machines.

Nominal dimensions of cylindrical gear drive and gear wheels are calculated as per GOST 16532-83. The tolerances are selected automatically in accordance with the accepted grades of accuracy as per norms of kinematic accuracy, smoothness of operation, teeth contact, type of gear mating and type of backlash tolerance based on GOST 1643-81 at $m > 1$ and GOST 9178-81 at $m < 1$.

The procedure of strength calculations (including the design and verification calculations) meets GOST 21354-85 requirements. Strength calculation uses the basic calculation dependencies to determine contact strength of the active tooth flanks and teeth bending strength as specified in this standard.

At computation of the forming female dies, the following versions of design are possible [17].
The INTUS program permits to compute the forming female die by the values of the limiting mould shrinkage $\varepsilon_{\text{min}}$ and $\varepsilon_{\text{max}}$, available in reference books on plastics. Computation is performed according to the mean shrinkage value and mean parameters of plastic gear corresponding to the mean of tolerance range.

The IMITAT program is intended to compute the forming female die using the results of gear simulator measurements. First, statistical characteristics of the mould shrinkage are determined.

The HELIUS program is proposed to compute the helical gear female die. Just as the first two programs, in HELIUS program the reference data on limiting values $\varepsilon_{\text{min}}$ and $\varepsilon_{\text{max}}$ of mold shrinkage or sample-simulator diameter measurements are used to calculate the mean value $\bar{\varepsilon}$ and confidence interval $\Delta \varepsilon$. In addition to radial mold shrinkage, the data on axial mold shrinkage $\bar{\varepsilon}_{\text{ax}}$ are needed. If $\bar{\varepsilon}_{\text{ax}} \neq \bar{\varepsilon}$ the female die helical angle $\beta^m$ is corrected:

$$tg \beta^m = tg \beta^c \cdot (1 - \varepsilon_{\text{in}}) / (1 - \varepsilon)$$

The KORMAT program is mainly used to correct female die dimensions in order to improve accuracy of the molded plastic gears.

The developed computer programs help to plot the female die tooth profiles corresponding to the mean value and extreme limiting deviation of shrinkage, use tooth profile coordinates in the graphic files generating NC code to transfer information to the wire-cutting machine in order to cut the female die. The software program “PC-aided design of cylindrical gear drives with plastic gears” has been registered in the RB Center of Intellectual Property under No. 370 [18].

2.2. PC-aided plotting of tooth profile

2.2.1. General approach and particular solutions

There are the techniques available for approximation of involute tooth profile, such as B-Spline interpolation [19] and tangent circular arc approximation [20]. As noted in [21], the latter technique has been improved by Yu.V. Shekhtman who proposed a software algorithm permitting to automatic select the number of approximating circle arcs for achieving a required accuracy. The peculiarities of symmetric and asymmetric tooth profile plotting described in [21] as follows:

- The gear accuracy is considered as a deviation from the coordinate point to the closest point of the arc and should be much less than the tooth profile tolerance (typically, the deviation is about 0.0005 mm);
- The tooth tip land arcs may have the centers not coinciding with the center of the gear, since the top land should be replaced with the tooth tip diameter arc and tooth tip radii or chambers;
- The ends of the connected tooth profiles may not coincide exactly, because the tangent arc should be fit to connect the neighboring tooth profiles;
- In the case of plotting the symmetric tooth, it makes sense to fit tangent arcs only to one-half of the tooth profile and then mirror it also fixing the tooth tip and root areas, because if the left and right profiles are plotted independently, they will be not exactly symmetric.

2.2.2. The technique for tooth profile involute section approximation

It is accepted, that the tooth profile for which the female die is made by a wire-cutting machine consists of the following sections [22]: arc of the root circle; fillet curve; straight line segment directed along the radius and connecting the fillet curve with the involute section; involute section of the tooth profile; rounded off arc near the tooth tip which radius is determined by the wire electrode diameter; the tip circle arc. In a general case, the overall possible versions of the tooth profiles are considered, namely, at $\alpha_L < 0$ (where $\alpha_L$ – angle determining the location of involute the starting point) and at $\alpha_L > 0$ – two versions are considered – at the different combination of central angles, determining the location of the space middle point and involute starting point, as well as location of the involute pressure angle and the angle between the radius passing through the middle of the space and perpendicular to the base circle [22]. The subprogram was developed for computation of different contours of tooth profile, namely: the gear tooth profile, the forming die one and (or) the electrode one. Taking into account normalizable deviations and tolerances the profile coordinates are computed with upper and lower
deviations corresponding to upper and lower limit deviations of the controlled dimension and mean the
deviation, corresponding to the middle of the tolerance zone for each above-mentioned object of
measurement. The tooth profile configuration is considered as independent of the rack tooth profile.

Involute section of tooth profile is approximated by circle arcs. Two versions of approximation
procedure are described in [22]. One of three types of the approximating circle arcs are used: the initial
and final arcs pass through the point of the fillet curve conjugation with the involute and the point of
the involute conjugation with the tooth tip rounded-off arc, intermediate circle arcs and conjugating circle
arcs. The centers of the intermediate circle arcs are located on the base circle; the centers of the initial,
final and conjugating circle arcs are located at some distance from it. Positions of the arc centers are
calculated by the involute roll angles: initial circle, intermediate circles, conjugating circles and the final
circle. Allowable approximation error is given by the designer. The number of approximating arc pairs is
chosen so that the real deviation of approximation is not more than the given one.

In accordance with another program version, more simple technique is used. Any section of the
involute is approximated, as a minimum, with one pair of arcs in condition of the minimal distance of
both arcs from an involute in the point, placed on passing the straight line extension connected curvature
radius centers (traced tangent to the base circle) with the first and second approximation circles. Quantity
sections approximated involute and associated pairs of arcs is chosen so in order provide given profile
accuracy. It is clear that the number of approximating circle arcs rises with the increasing of the design
module and defined accuracy and decreases with the tooth number grow (Table 1).

Table 1. Influence of gear geometrical parameters and given accuracy of approximation
on the number of circle arcs approximating involute part of tooth profile.

| Tooth number | Number of circle arcs (n) and real error of approximation (δ, µm) at a target profile error (δ, µm) and module (m, mm) |
|--------------|------------------------------------------------------------------------------------------------------------------|
| z            | m     | δ      | 100.0 | 5      | 10.0 | 5      | 1.0   | 5      | 0.10 | 5      |
| 9            | n     | δ      | 44.3  | 16.5   | 27.4  | 5.5   | 4.9   | 5.5   | 0.7   | 0.6   | 0.9   | 0.10  | 0.10  |
| 18           | n     | δ      | 19.3  | 16.0   | 12.6  | 2.4   | 7.5   | 3.7   | 0.7   | 0.9   | 0.8   | 0.09  | 0.08  |
| 36           | n     | δ      | 6.1   | 19.0   | 31.9  | 6.1   | 2.4   | 4.0   | 0.8   | 0.7   | 0.5   | 0.09  | 0.09  |

2.2.3. Fillet curve formation
The fillet curve is considered to be circular. A circular fillet curve is formed in accordance with the
principle of obtaining the maximum possible radius in order to obtain a minimal stress concentration at the
tooth root.

2.3. Updating accuracy parameters of molded plastic gears
2.3.1. Versions of mold shrinkage computation
Computation presumes the determining mold shrinkage components in terms of generalized parameters [23], when the base geometry of the tooth ring is computed independently of the parameters of the basic rack tooth profile. As independent parameters, the base diameters \( d^n_n \), angular thicknesses on the base circle \( \theta^w_n \) and tip diameters \( d^n_t \) of the gear (index “g”) and mold die (index “m”) are accepted. According to the generalized parameters, the coefficients of mold shrinkage \( S_n, S_m, S \), are computed [24]:

\[
S_n = (d^n_n - d^n_t) / d^n_n
\]

\[
S_m = (d^n_m - d^n_t) / d^n_m
\]

\[
S = (\theta^w_n - \theta^w_t) / \theta^w_n
\]
Parameter \( d_{gb} \) is directly measured. To compute the remaining \( d_{gb} \) and \( \theta_{gb} \), one of the following methods described earlier in [25, 26] is used:

- Mold shrinkage computation by coordinates of tooth profile points;
- Computation of involute parameters by measuring dimensions over the balls with different diameters;
- Method of difference of the base tangent lengths.

2.3.2. Procedure conformance between real accuracy indices and computed ones

Deviation of any actual mean controlled dimension \( \bar{X} \) from a specified mean dimension in drawing \( \bar{X}^* \) is compared to confidence interval \( \Delta X \) for the mean one

\[
|\bar{X} - \bar{X}^*| < \Delta X / 2\sqrt{n}
\]

(5)

on the base of comparing the deviation of dimension \( \bar{X} \) from \( \bar{X}^* \) with tolerance \( T \), the probability of obtaining gears of required accuracy is evaluated

\[
P(X) = \Phi_{\sigma}(z_1) - \Phi_{\sigma}(z_2) \geq P(X)^*,
\]

(6)

where

\[
Z_1 = \frac{\bar{X}^* + T}{\frac{2 - \bar{X}}{\sigma_z}}; \quad Z_2 = \frac{\bar{X}^* - T}{\frac{2 - \bar{X}}{\sigma_z}}
\]

(7)

Confidence interval is computed by equation:

\[
\Delta X = \bar{X} \cdot t(\alpha)
\]

(8)

where \( \sigma_z \) – mean square deviation confidence, \( t(\alpha) \) – Student’s factor.

3. Improvement of plastic gear quality parameters using the Direct Gear Design System

The development of the system of generalized parameters proposed by E.B. Vulgakov with its modern representation in the form of a software and computer-aided design is connected with the name of his disciple, A.L. Kapelevich engaged in extension of this direction under a common brand the Direct Gear Design [27]. The system Direct Gear Design uses CAD for the involute gear drive design with the optimal parameters meeting the requirements specification by presenting the zone of existence (zone of feasible design solutions) of the drive with asymmetric tooth profile in the form of limiting contours in coordinates \( \nu_1 \) and \( \nu_2 \) (\( \nu_1,2 \) – tip circle profile angles for pinion (1) and gear wheel (2) correspondingly). The existence zone for a gear pair with teeth numbers \( z_1 \) and \( z_2 \), preset asymmetry factor \( k \) and coefficients of the tooth top thickness \( m_{a1} = m_w/d_{b1} \) and \( m_{a2} = m_w/d_{b2} \). After determining all parameters of the involute tooth profile by the Direct Gear Design the fillet curve parameters are optimized using the following procedures [27, 28]: a stochastic method of arranging fillet points so as to ensure the minimal bending stresses; the method of trigonometric functions for fillet curve approximation; the finite element method for stress computation. The optimized fillet curve provides for a minimized radial clearance, eliminates profile interference, inducing thereby bending stress distribution over a large portion of the fillet curve due to a resultant maximal curvature radii along its whole length, which brings about a significant stress concentration reduction. One of the latest versions of the Direct Gear Design [28] considers a CAD variant that along with the traditional method (creation of the existence zone of the involute gear pair mesh having teeth numbers \( z_1 \) and \( z_2 \), and parameter \( k \)) involves an accelerated technique. This method presumes the preliminary optimization of the fillet curve over an ellipse in order to ensure the choice of a required variant of the gearing, using which the curve fillet is then optimized finally following above-described procedure.

Strength of the gears with asymmetric teeth was calculated by the software Direct Gear Design. To attain the maximal probable benefit from using the asymmetric profile we have realized the conditions of the maximal contact ratio by admitting an undercut of the non-operating tooth profile part under the
conditions of roughly similar bending strength of the pinion and gear teeth. Results of comparative computation of the gearings with symmetric and asymmetric teeth (Table 2) prove the growth of elevated bearing capacity of the gears with asymmetric teeth due to reasons noted in [29]:

- Higher pressure angle ($\alpha_{wd} = 23\ldots25^\circ$ in contrast to $\alpha_{w} \approx 20\ldots22^\circ$ for symmetric teeth);
- Increased face contact ratio ($\varepsilon_{a} = 1.34\ldots1.45$ for symmetric teeth and $\varepsilon_{ad} = 1.7\ldots1.8$ for asymmetric teeth);
- Balanced bearing capacity due the rational pinion to gear wheel tooth thickness ratio ($c_{5} \approx 1.55\ldots1.9$ for asymmetric teeth and $c_{5} \approx 1.35\ldots1.79$ for symmetric ones);
- Reduction of bending stress for asymmetric teeth in comparison with symmetric ones as follows 30.0…58.6%.

### Table 2. Comparison of quality parameters of gear pairs with symmetric and asymmetric teeth.

| Parameters | Parameter value for gear pair | 1$^\text{)}$ | 2$^\text{)}$ | 3$^\text{)}$ |
|------------|-------------------------------|---------|---------|---------|
| Tooth number, $z$ | pinion gear wheel | 13 50 | 13 65 | 11 57 |
| Operating center distance, $a_{w}$, mm | | 25.506 39.525 | 51.0 |
| Face width, $b_{w}$, mm | | 15 14 | 21 |
| Gear ratio, $i$ | | 3.846 5.0 | 5.182 |

#### Symmetric tooth profile

- Module $m_{n}$, mm: 0.8 1.0 1.5
- Pressure angle, $\alpha_{w}$,°: 21.813 21.994 20
- Contact ratio, $\varepsilon_{a}$: 1.435 1.343 1.447
- Ratio $C_{3} = S_{w1}/S_{w2}$: 1.35 1.79 1.53
- Bending stress, $\sigma_{FS}$: 8.8 9.2 18.8 24.05 33.0 36.0

#### Asymmetric tooth profile

- Module at working circle, $m_{w}$, mm: 0.80973 1.01345 1.50

#### Drive flank

- Pressure angle, $\alpha_{wd}$,°: 23 23 25
- Contact ratio, $\varepsilon_{wd}$: 1.695 1.791 1.723

#### Coast flank

- Pressure angle, $\alpha_{we}$,°: 15 15 16
- Contact ratio, $\varepsilon_{we}$: 1.387 1.474 1.342
- Ratio $C_{3} = S_{w1}/S_{w2}$: 1.55 1.85 1.90
- Bending stress, $\sigma_{F}$, MPa: 5.6 5.8 13.3 16.5 25.4 28.2

Notes: 1. For designations “operating flank” and “non-operating flank” the “drive flank” and “coast flank” notions, proposed by A.L. Kapelevich, are used. 2. Bending stress $\sigma_{FS}$ for symmetric teeth is computed by the software “Gear Pair”; $\sigma_{F}$ for asymmetric teeth is computed by the software “Direct Gear Design”. 3. $^\text{1)}$, $^\text{2)}$, $^\text{3)}$ are the first, second and third reducer steps, correspondingly.

4. Original two-level calculation method for design and optimization of polymer composite gears of dispersion-filled plastics

Polymer materials, first of all, polyamides and ultra-high molecular weight polyethylene, may be successfully used in gear manufacturing due to a good workability, very low friction coefficient, high impact and corrosion resistance. Traditionally, strength and wear resistance of mentioned above basic plastics are increased by the addition of micron size disperse particles and short fibres of inorganic materials (described in [30, 31]). For designing stress-strain state and wear resistance of disperse-filled composite joints, including gears, the effective two-level method has been developed [32, 33]. The method allows us to determine the stress-strain state, compliance, strength and wear resistance of composite gears, as well as to find an optimum composition of the material on the basis of requirements for the gearing without geometrical corrections of the wheels. It is proposed directly in the micromechanical models to determine the effective elastic and strength characteristics of disperse-
filled plastics taking into account their structure and characteristics of the matrix, filler and interphase components. Thereafter, such characteristics as an initial data for optimal design of gears are used.

The effectiveness of the proposed two-level method for gears consists of a rational combination of analytical micromechanical approach of the elasticity theory and numerical macromechanical techniques (first of all, finite element approximation) of machine part strength analysis. Due to the first level it is suitable to describe the deforming process of essentially heterogeneous composite materials and due to the second one – to determine the parameters of the stress-strain state of intricate in shape parts, i.e., the gear. The benefits of the method are a possibility to make computer design to optimize material composition, elastic properties of the polymeric matrix, form and dimensions of reinforcing and other functional filler particles, as well as the parameters of the interphase layer according to the deformation, strength and tribological criteria.

5. Adoption of IOSO program complex for optimization of gear pair meshing parameters

5.1. General provisions

IOSO software is intended for engineering multicriterion multiparametric optimization of technical objects. According to IOSO technology, in the first stage the simulation model of the optimized object is created. The common simulation models correspond to technical objects and computations realized in the standard CAD-software (SolidWorks, SolidEdge, Inventor, ProEngineer, UniGraphics and others) or software written at Pascal, Delphi, C++ or in the Excel programming environment. In search of an optimal solution of the mathematical model the following iteration process takes place: IOSO program complex generates in automatic mode the input data values and record in the file of the data input, then the IOSO program package runs the mathematical model, which records the calculated output parameters (criteria) in the output file; the IOSO-system pickups the output file, and analyzes the received results; then the IOSO algorithm accepts a decision about changing input parameters.

5.2. Optimization of front loader gearbox back row geometrical parameters

The goal of optimization is the reduction of the gearbox noise, an oblique figure of which is a contact ratio. The contact ratio value is necessary to maximize, by preserving the accepted close to the former center distances and gear ratios of the back row gear pairs, as the most loaded unit of the front loader gearbox (Fig. 1). The procedure is the following.

- For reduction of noise in a three gear pair meshing, three criteria of optimization are introduced, namely, contact ratios $\varepsilon_\alpha$ (5-10), $\varepsilon_\alpha$ (7-10), $\varepsilon_\alpha$ (12-10), which values should be maximized.
- To avoid the absence of tooth undercutting, mesh interference and tooth sharpening should be provided the existence of every gear pair meshing (twelve limitations).

Figure 1. A model of the back row gear pairs of the front loader gearbox.

*Developers of IOSO-system is Moscow company “Sigma Technology” (www.iostech.com) and its Minsk subsidiary production unit “IRION” (www.irion-m.com, www.engineeringoptimization.com)
To provide durability of gears according to bending and contact stress safety it is necessary to input limitations on the corresponding safety factors for the bending state \((K_F5; K_F7; K_{F10}; K_{F12} > 1.1)\) and for contact state \((K_{H5}; K_{H7}; K_{H10}; K_{H12} > 1.1)\).

The optimization parameters are worked out, including module \(m\); pressure angle \(\alpha\); addendum coefficient \(h_a^*\); shift coefficient \(x\); and tooth number of the central gear \(z10\), under which all other gears are engaged.

The rest numerous geometrical factors will be derived from above-mentioned ones. Finally, the decreasing noise of the front loader gearbox represents a multicriteria (three criteria of maximization of the contact ratio for every gear pair), multiparameter (five optimization parameters – \(m, \alpha, h_a^*, z10, x10\)) problem with twelve limitations (tooth undercutting, mesh interference and tooth sharpening for each gears of the four).

When creating the mathematical model of each gear pair in the gearbox the following standard calculation software is used, namely: the program of involute gear drive geometrical calculation according to GOST 16532-83 and the program of strength calculation for the involute cylindrical gear drives in accordance with GOST 21354-87. Further, by omitting details related to the final mathematical model, a structural graph and the diagram of optimizing the project are worked out at the final stage of optimization using IOSO provisions the input and optimization parameters are assigned, input and output files are used, along with above-mentioned calculation programs.

The results of solution are presented in Table 3, where one can see, that the optimal version is obtained for the gear pairs with a nonstandard basic rack tooth profile \((m = 4,25 \text{ mm}, \alpha = 16^0)\). As a consequence, the optimal solution for the front loader gearbox row gear pairs based on criteria noise and strength was found. This solution was obtained by increasing contact ratios for all gear pairs at preserve approximately accepted formerly center distances and gear ratios, giving the possibility not to change the hauling balance of the front loader.

| Characteristics of optimization version | Output parameter values for version of gear pair \(z_{5-10}\) | \(z_{7-10}\) | \(z_{12-10}\) |
|------------------------------------------|---------------------------------|---------------|---------------|
| Initial version \((m = 5 \text{ mm}, z_i = 43, 20, 23, 34)\) | \(\varepsilon\) \(1.450\) | \(1.418\) | \(1.538\) |
| Optimized initial version \((m = 5 \text{ mm}, z_i = 43, 20, 23, 34)\) | \(\varepsilon\) \(1.541\) | \(1.528\) | \(1.576\) |
| Version with transformed module and tooth number \((m = 4.5 \text{ mm}, z_i = 48, 23, 26, 38)\) | \(\Delta \varepsilon\), % \(6.3\) | \(7.8\) | \(2.5\) |
| Version with transformed module and tooth number \((m = 4.25 \text{ mm}, z_i = 51, 24, 27, 40)\) | \(\Delta \varepsilon\), % \(13.1\) | \(7.5\) | \(7.3\) |
| Version with transformed module, tooth number and pressure angle \((m = 4.25 \text{ mm}, \alpha = 16^0)\) | \(\Delta \varepsilon\), % \(34.5\) | \(31.2\) | \(33.9\) |

Notes: \(\Delta \varepsilon\) – increasing of contact ratio in comparison to initial version

6. Adoption of KIMoS System at design and manufacturing

“Belarus” tractor final drives of front and back axles

The KIMoS (Klingelnberg Integrated Manufacturing of Spiral Bevel Gears) is a module system of the computer program enabling to realize planning and integrating of production of bevel gears with computer support and different methods of machining. The data are controlled in the database determined by the user and preserved in the neutral format. The data describe the geometry of the bevel gear pair, adjustment and kinematics of the gear-cutting machine, as well as adjustment of the gear-cutting hob regardless of the manufacturing process and the system. The open joint-stock company “Minsk Tractor Plant” employs the computer program KIMoS in order to support quality of circular tooth bevel gears, used in “Belarus” tractor spiral bevel final drives of the front and back axle.

The software system KIMoS is used at all stages of design and manufacturing bevel gears. The following steps are realized using this program:
computation of geometrical parameters of bevel gear pairs and base adjustment for gear-cutter and gear-grinding machines;
- analysis of gear pair meshing without load and under load;
- correction of development stages based on the results of meshing analysis;
- computation of tooth flank surface coordinates for measurement on the coordinate measurement machine;
- correction of development settings on the base of tooth flank surface control.

Tool grinding machinery, gear cutting and gear grinding machines, measuring and form-generating units are jointed in one integrated network and is controlled according to a Closed-Loop technology for each machining operation aimed at receiving the applicable work-piece. Screen-shots of the design process contain the basic data on the bevel gear pair geometry and tooth (tooth addendum and dedendum factors, pressure angle, basic rack tooth profile shifting etc.); gear tool and gear machine parameters; gear pair and gear theoretical dimensions; optimization of ease-off, at which meshing surface is changed purpose fully; form, dimensions and location of the tooth contact pattern; curve of irregularity of transmitting rotation; analysis of tooth contact under load; stresses and noise level, etc.

Finally, the Closed-Loop technology provides correspondence of tooth contact pattern parameters on bevel gear test fixture to the one computed by the KIMoS program “Analysis of tooth contact”. The KIMoS system and Closed-Loop concept provides transparency and document supported quality of technological processes along the whole technological network; guarantees exact manufacturing of calculated; items permit to exclude the influence of human factor on the final product quality and to provide high-repeatability precision of quality.

7. Conclusion
The results of the authors in the field of automation of gear drives and transmissions have been generalized, including the review of publications on the topic. Quite a few directions in investigation of plastic gears and composite gears, in particular, the advanced two-level calculation method for design and optimization of dispersion-filled composite gears is proposed. The examples of using known standard software systems for design and manufacturing mechanism and machine drives are presented.

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