DEVELOPMENT OF A MACHINING CENTER DRIVE WITH INCREASED EFFICIENCY OF GEARING

Oleg Krol[1], Vladimir Sokolov[1], José Machado[2]*, Maksym Yakunin[3]*
[1]Department of Machinery Engineering and Applied Mechanics, Volodymyr Dahl East Ukrainian National University, 59-a Central pr., Severodonetsk, 93400, Ukraine line of affiliation
[2]MEtRICs Research Center, Campus of Azurém, University of Minho, 4800-058 Guimarães, Portugal
[3] Kharkov Applied College of Mechanical Engineering, 61086, Kharkov, Ukraine,
E-mails: jmachado@dem.uminho.pt , krolos@snu.edu.ua

Abstract - The design of the main movement drive for a multioperational machine such as a machining center with a modified design of a gear gearbox is considered. A 3D modelling of the machining center drive mechanism with a portal layout was performed in the integrated computer-aided design system KOMPAS-3D. Functional extended range of the C3D Modeler geometric kernel when constructing complex geometry of machine housing parts is proposed to use. The procedure for constructing gear rims with high accuracy and geometrically correct tooth surfaces in the CAD environment was applied. At the stage of forming the shape of the tooth of a multivariate choice of the tooth axial shape, it is possible to implement. A new idea to increase the efficiency of the cylindrical gear transmission of the machine speed box based on the modified tooth profile of the gears with inclined-arched teeth is put forward. An analytical apparatus for calculating the design and choosing the radius of the gear-shaping cutting head with known basic transmission parameters have been formed. A comparative analysis for the calculated values of the axial overlap coefficient in an involute helical spur gear and in a spur gear with inclined-arched teeth is carried out with the same engagement parameters, other conditions being equal. Numerical calculations for the modified version of the engagement with inclined-arched teeth compared with the involute helical gear in terms of load capacity have been shown.

Keywords: Machining center Drive, 3D modelling, Inclined-arched profile, Gear-cutting head, Load capacity.

1. Introduction

In modern mechanical engineering, multi-operation machines (machining centers (MC)) are increasingly used, which are characterized by a particularly high concentration of processing [1, 2]. The prospects for using such machines correspond to the development of small and medium-sized enterprises in terms of ensuring their competitiveness [3]. They produce rough, semi-finish and finish machining of complex housing workpieces: milling planes, alters, grooves, windows; drilling, countersinking and reaming, etc.

To carry out these operations, the machine is equipped with a sufficient supply of metal-cutting tools. The presence of high-capacity tool magazines, rotary and globe tables, with dividing, for automatic change of workpieces form their high technological efficiency. A feature of MC kinematic schemes is the complete autonomy of kinematic chains for each of the displacement coordinates. The ultimate simplification of these chains, including a significant simplification of gear

reducers, and the possibility of the widest unification of the main structural elements within both one machine and various MC sizes [4].

The main (basic) types of layouts regulate the main parameters and dimensions of machines: the width (diameter) of the working surface and the length of the table, the inner taper of the spindle, the dimensions of the T-shaped slots, the number of controlled coordinates, the discreteness of the displacement assignment.

Based on the aggregate-modular principle, multi-purpose machines of various layouts are created [5, 6]. A portal layout with a movable portal for parts with a width of the machined surface $H < 800$ mm and a length of the machined surface $L < 5000$ mm has become widespread. It is the most economical in terms of occupied area and provides a thermosymmetrical arrangement of the spindle head. At the same time, this design has increased rigidity and accuracy.

The main drive occupies a special position in the MS structure and can include an electric motor (which in most cases is mounted separately from the drive), a frequency converter, and 2-3 mechanical gears to increase the speed and torque range. This configuration option provides high-performance execution of various operations when changing the rotational frequency in a wide range of speeds and torques.

The reliability of MS operation is also limited by the performance of the shaping main movement drive of the spindle unit [7, 8]. In this regard, the problem of improving the designs and manufacturing technology of the MS-drive elements is relevant in the context of the program for the creation and modernization of modern machining centers.

2. Literature Review

The work [9] is devoted to the issues of modeling and optimization of machine tools. It is noted that the greatest effect is achieved when considering the machine as a whole as a system that includes mechanical (carrier system, drive), mechatronic and electrical parts, as well as a control system. Particularly noted is the need to analyze the specifics of the mechanical component at the initial stages of project development in the form of complex 3D models. As an example, a study of a CNC machine for waterjet cutting is carried out. The interdisciplinary nature of machine tool research places high demands on the product development process, on 3D and simulation models, and on optimization strategies.

The paper [10] considers the problems of 3D modeling machine tools. The author emphasizes that the designer must first form the 3D entity of the fixture design in his mind and then express it with a 2D diagram to complete the design of the fixture. In the process of iterative design, the virtual structure is repeatedly transformed into three- and two-dimensional models. To implement this process, Unigraphics Software (UG) is used as a set of 3D CAD/CAM/CAE integration of high-quality software. Based on the UG-3D modeling of the CNC machine fixture design, a 3D assembly drawing of the tooling is formed.

It is noted in the article that the application of modern equipment, materials' information, and software allows increasing hard steel processing efficiency to receive with necessary characteristics, which is possible due to the introduction of new programming tools and Industry 4.0 [11].

Algorithms and programs for constructing geometrically accurate 3D models of various mechanical transmissions of machine tool drives are presented in [12]. This allows us to consider the methodology for building three-dimensional models in AutoCAD as an advanced development of accurate samples to control complex parts. Algorithms when writing control programs in machines for the automated production of parts are used. At the same time, there are no recommendations for adapting these programs for the case of design modernization of drive gear elements and variation of the corresponding control programs.

In [13], the issues of a comprehensive study of milling machine parameters are considered. A technique for analyzing the layout scheme of the machine has been developed. At the same time, various configuration options for the rotation axes of a high-speed spindle and displacements of the feed mechanism of angular and rotary tables with workpieces a taken into account. With consideration of this technique, an algorithm for the formation of a database, various layout options, and configurations of coordinate axes were created. A database structure, which includes a number of sections containing data on spindle heads, tooling, and milling machine drives in the form of 3D models is proposed. Along with the constructive component in the form of three-dimensional models, a complex of analytical models and arrays of experimental data have been formed. The authors of [13] introduce a modular modelling system using Boeing-Jeppson Machine Prediction Software, one of the advantages of which is the ability to predict the stability of the cutting process. At the same time, restrictions on the size of the working area for the milling machine are superimposed. In accordance with the given sequence of the modelling process, it is supposed to perform preliminary calculations of various machines. At the same time, information about such an important section of the database as mechanical transmissions of milling machine drives is not provided.

When designing gear drives of milling machines, the development of constructive conditions for the formation of drive stage parameters is carried out. Maximizing the load capacity, and minimizing contact stresses, as a result, increasing the resource and operability of the drive and the machine as a whole are chosen as optimization criteria.

In [14], a variant of the design of a two-stage bevel-helical gearbox is proposed. This variant is optimal in terms of the minimum mass criterion. At the same time, the condition for increasing the efficiency in terms of the contact strength of the wheels is analytically confirmed. The authors of [14] formed dependencies reflecting the influence of the gearbox length on the allowable contact stress and the geometric parameters of the wheels. The software "CAD-9" for the process of designing a bevel-helical and two-stage spur gearbox according to an expended and coaxial scheme was created.

The effective software "Design optimization Gearbox KISSsoft" was proposed in [15] where the tools of the American Gear Manufacturers

Association (AGMA) are implemented. Gearboxes based on spur and bevel gears are considered design objects. At the preliminary design stage, the criterion for minimizing the volume of the internal cavity of the gearbox was chosen, and the level of contact and bending stresses were considered as constraints. In the process of design, rational values of the teeth number of a gear-gear pair and the design parameter of the gear drive module were obtained. This paper [15] also considers a method that extends the concepts described in AGMA 929-A06. It allows the user to calculate not only the surface thickness but also the more general case, i.e. the normal tooth thickness at any location along the surface and profile of the tooth. wheels.

There are works that use the multicriteria nature of the designing problem a two-stage gearbox [16], which has found wide application in drives of multi-operational metal-cutting machines. A study to search for a promising design according to the criteria of minimizing the mass and maximum efficiency, which compete with each other was conducted. At the same time, the assignment of significance weights for each criterion is subjective, expert in nature and depends on the type of mechanical gear used in the gearbox.

At the same time, the study in which the effect of non-stationary random processes on the design of mechanical transmissions was analyzed [17]. Thanks to the use of the method of successive approximations, it was established that although the original equation is not a wave one, its solutions are of quasi-wave character in the sense that each subsequent approximation substantially changes the solution only from a certain time moment, all the more, the higher the number of the approximation being made will be.

An analysis of the above works showed that, at their core, variants of gear drives already tested in the practice of machine tool building are considered. It should be noted that in the search for new design options for implementation and technology for their manufacture insufficient attention is paid. Research in this direction can be one of the levers for improving the reliability and efficiency of machine tools.

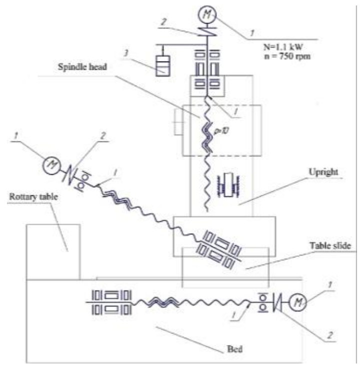
Formulation of the problem. Improving the design of the main movement drive for a multi-operation milling machine with a portal layout using gears of increased efficiency.

3. Three-dimensional Simulation of the Structure for MC Portal Layout with a Two-stage Drive

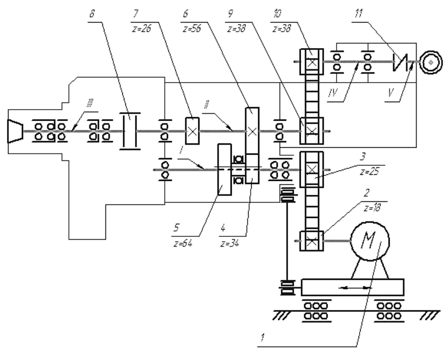
3.1. Design Features of MC Portal Layout

For complex housing parts, the processing of which combines contour milling, boring of shallow holes, trimming ends, etc., and machining centers with a portal layout are widely used. Such an arrangement is characterized by a rational combination of rigidly compact dimensions and the shape of the "Cube" type coordinate space (determined by the ratio of the values of coordinate moves in the coordinate system: X:Y:Z) [18].

In small-scale and medium-scale production, horizontal machining centers are more often used. This type of MC200PF4 (Fig. 1) uses the structural formula $XOYZC_h$ in which the symbols of the formula indicate the movable blocks of the machine, starting from the "table-workpiece" block to the spindle block.



(a) Carrier system



(b) drive of the main movement

Figure 1: Kinematic diagram of MS 200PF4

Such machines with a movable portal (Z axis) provide high-speed processing of housing parts from several sides without part repositioning [19, 20]. This type of machine is called "CNC gantry type machine". CNC gantry-type milling machines can usually complete some complex and complicated parts that are difficult or impossible to be processed by ordinary machine tools, such as various complex curves, curved surface parts and variable lead threads.

As a basic design, consider the model of the MC200PF4 machining center, the features of which include (an enlarged kinematic diagram is shown in Fig. 1, a):

- mounted all components and mechanisms on a rigid box-shaped bed, without fixing on the foundation;

- a turntable, a mechanism for changing satellite tables, and a stand on which the tool magazine is mounted [21, 22];
- frontal cantilever spindle head with hydraulic block [23], mounted on a portal-type rack with vertical (Y-axis) and horizontal (X-axis) movement.

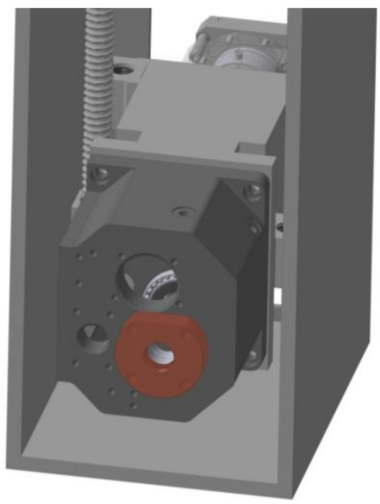
In the machine under consideration, the kinematic connection: spindle – drive output shaft– feedback sensor ROD 700, is implemented in the form of a transmission with a toothed belt 38:38 of the cantilever type [24]. The torque is transmitted to the output hollow shaft of the gearbox from the 4PF112M DC motor through an 18:25 toothed belt drive and a 34:56 spur gear transmission. From the output shaft, the torque is transmitted through the gear coupling to the spindle (Fig. 1, b).

3.2 3D Simulation of the MC Drive

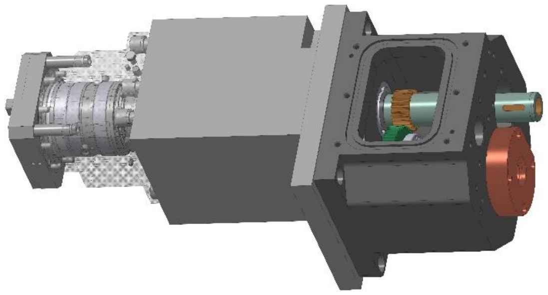
To analyze the design of the MC main servodrive (Fig. 1, b) and carry out its calculation, a complex of solid models was developed using CAD KOMPAS 3D [25, 26].

Traditionally, at the stages of preliminary and technical design, the main features of the MC future design of the being created are worked out at the level of three-dimensional modeling. This applies primarily to the MC basic parts and forming units of the. The KOMPAS-3D system implements the top-down design principle from the assembly unit to the details.

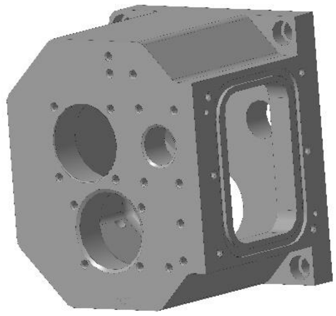
This process effectively uses layout geometry operations, which are designed to control the relative position of components in the main drive (speed gear-box) and act as an assignment for detailed modeling of all its parts. Editing components in the context of an assembly model helps to take into account the overall contours of mating or neighboring parts and create a primary version of the design. On Fig. 2 shows three-dimensional models of a speed gear-box mounted on a moving portal of the machine.



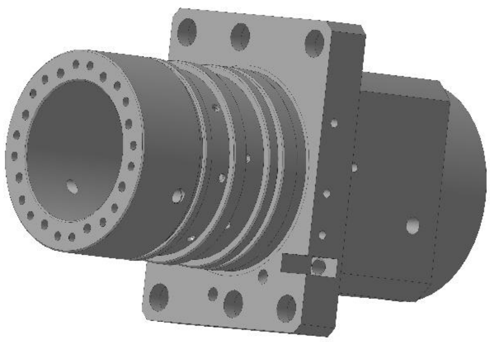
(a) General view of the drive



(b) drive with spindle assembly



(c) drive housing;



(d) spindle assembly housing

Figure 2: 3D models of the main movement drive on the portal

The software and a set of graphic primitives KOMPAS based on the C3D Modeler geometric core allow you to build a geometric model of the housing parts for the main motion drive and calculate the geometric characteristics of the modeled object.

For complex three-dimensional models of casting workpieces (Fig. 2, c; d), characterized by the presence of complex cavities, various holes and rounding in CAD KOMPAS.

A new functionality for selective removal of holes is used. In this case, you can delete the specified holes or holes, the radius of which does not exceed a certain value.

Similarly, you can delete the specified rounding or rounding, the radius of which is less than or equal to the specified value. The same function can be used in direct modeling when you need to make changes to a 3D model with rounding, for example, moving individual elements of this model to a different location in a multivariate design. This can be useful for simplifying a 3D model that is being prepared for further analysis in a CAE system. The software of the system has a geometric core C3D Modeler (calculation of geometric characteristics and construction for a geometric model) as well as a parametric core C3D Solver for the efficient development of a complex drive design of a multioperational machine.

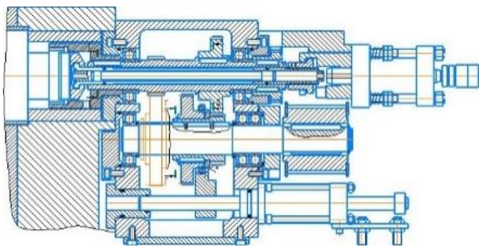
The C3D Modeler toolkit includes a number of new features that make the process of designing complex drive housing parts more productive (Fig. 2, c; d). In the process of creating a three-dimensional model of a housing part, the extrusion operation from the sketch plane is widely used. Its extended functionality consists in the ability to create a thin-walled element of a housing part at any time in the design by changing the parameters during editing (for example, in CAD SolidWorks allows you to make a thin-walled element only at the time of creating the operation).

As a result, it becomes possible to build previously inaccessible combinations of rounding in thin-walled element, which is important when designing molds of varying complexity.

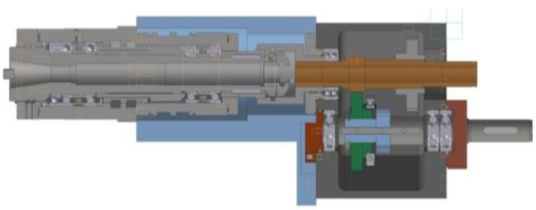
3.3 Construction of the Three-dimensional Models of Gears

When designing and modeling the design parts and assemblies of the MC-drive, a special application "Shafts and mechanical transmissions 3D" with built-in catalogs of materials and standard products were used.

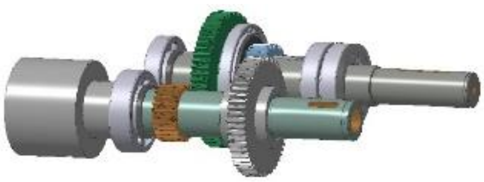
The developed three-dimensional models of the MC-drive elements are presented in Fig. 3.



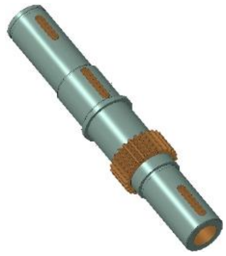
(a) Drawing of the drive



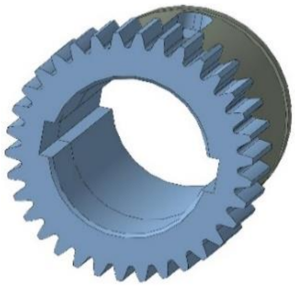
(b) cross-section of the drive



(c) kinematics



(d) pinion shaft



(e) bushing-gear

(f) gear

Figure 3: Three-dimensional models of gears

A feature of the process of designing mechanical transmissions of the MC drive in this application is the automated calculation of the displacement coefficients, which allows you to design an external spur gear with involute teeth (Fig. 3, d; e; f).

A feature of the process of designing mechanical transmissions of the MC drive in this application is the automated calculation of the displacement coefficients, which allows you to design an external spur gear with involute teeth (Fig. 3, d; e; f). When using this application, it becomes possible to design a gear that approaches the optimal variant according to several criteria: contact strength, bending strength, tooth equal strength, wear resistance and smooth operation [31].

It is important that the design calculations of gears according to those created in this application of operational mathematical models are carried out.

Together with the use of optimization methods, makes it possible to select a number of gear options, taking into account the bearings used [32] and satisfying the specified operating conditions. It is possible to design gearing parts in accordance with AGMA, ASA, DIN, etc.

An important advantage of using the C3D Modeler kernel when modeling gears is also the ability to use several closed involute contours in one extrusion operation at once, which, in addition, can intersect each other (in the absence of self-intersection in individual contours). Another kernel option is a separate case of calculating the curve of the intersection of two spheres touching the poles, which is effectively used when constructing a junction curve at the root of the tooth.

The advantage of this application is the geometrically correct 2D profiles of gear rims and generated 3D models. They can be used without any restrictions when creating programs for CNC machines. The generation of 3D models of gear rims is performed on the functionality of the C3D core.

4. Result and Discussion

4.1. Theoretical Part

Increasing the load capacity of gears by engagement on parallel axes is solved in various ways – by optimizing the initial contour, changing the longitudinal line of the teeth, using new types of hardening technologies, etc. One of the promising directions for solving this problem is associated with the so-called arched teeth [33]. In these works, a wide range of issues associated with the theory of spur gears (SG) engagement for arched type and their full-scale tests were investigated. These include the parameters of various types of initial contours, the equations of machine and working (length of the part of contact) gears, the reduced curvatures of the conjugating surfaces of the teeth, specific sliding, undercutting, overlap coefficients, etc.

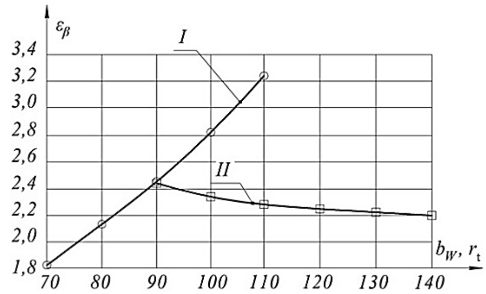
A variety of arched teeth are inclined-arched teeth, [30]. At the SG-arched teeth, in the average end section of the gear (wheel), the angle of inclination of the teeth is $\beta = \beta_m = 0$. At both ends this angles is the same and reaches a maximum value $\beta = \beta_{\max}$. In the SG with inclined-arched teeth in all end sections, including the average, the angle of inclination $\beta > 0$, moreover, from one end section to another the value β is changes.

This article describes one aspect of the SG theoretical study with inclined-arched teeth, associated with the choice of the Gear-Shaping Cutter Head (GSCH) radius with known basic transmission parameters – center distance a_W, width of the gear rims b_W, normal engagement modulus m_n, numbers of pinion (z_1) and wheel (z_2) teeth, average angle of the teeth inclination β_m. These parameters are determined by the criterion of the teeth contact endurance and the SG-geometric calculation according to known methods, [33].

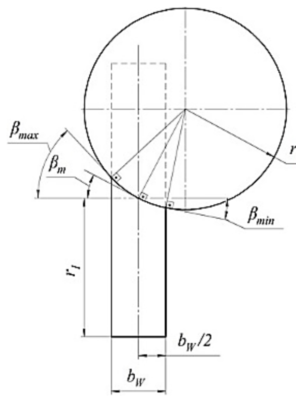
The coefficient of axial overlap in the SG with inclined-arched teeth, [33]:

$$\varepsilon_\beta = \frac{\cos\beta_m}{\pi \cdot m_n} \cdot \left(\sqrt{\begin{array}{l} r_t^2 \cdot \cos^2\beta_m + b_W \times \\ \times\, r_t \cdot \sin\beta_m - 0.25 \cdot b_W^2 \end{array}} - \sqrt{\begin{array}{l} r_t^2 \cdot \cos^2\beta_m - b_W \times \\ \times\, r_t \cdot \sin\beta_m - 0.25 \cdot b_W^2 \end{array}} \right). \quad (1)$$

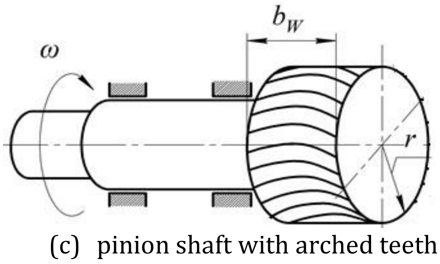
Numerical analysis of expression (1) shows that the parameters b_W and r_t have the opposite effect on the value of ε_β: with an increase b_W, the value $\varepsilon_\beta = \varepsilon_\beta(b_W)$ increases, and with an increase r_t, the value $\varepsilon_\beta = \varepsilon_\beta(r_t)$ decreases. The graphical functions $\varepsilon_\beta = \varepsilon_\beta(b_W)$ and $\varepsilon_\beta = \varepsilon_\beta(r_t)$ for $m_n = 4\ mm$ and $\beta_m = 17^0 49'27''$ are shown in Fig. 4, a.



(a) Influence of the head radius on the axial overlap coefficient (I: $\varepsilon_\beta = \varepsilon_\beta(b_W)$ at $r_t = 110$ mm; II: $\varepsilon_\beta = \varepsilon_\beta(r_t)$ at $b_W = 85$ mm)



(b) the layout of the gear-shaping cutting head



(c) pinion shaft with arched teeth

Figure 4: Gear-shaping cutter head

From Fig. 4, b it follows that the GSCH-radius r_t and its position relative to the wheel being cut cannot be set arbitrarily, since r_t and b_W are interdependent parameters, that is, $r_t = r_t(b_W)$.

Let's introduce an auxiliary variable k that will link the known value b_W and the required value r_t : $k = b_W / r_t$. As a result:

$$\varepsilon_\beta = \frac{b_W \cdot \cos\beta_m}{\pi \cdot m_n} \cdot [\sqrt{\begin{array}{l}(\cos\beta_m / k)^2 + \\ (\sin\beta_m / k) - 0.25\end{array}} - \\ - \sqrt{(\cos\beta_m / k)^2 - (\sin\beta_m / k) - 0.25}]. \quad (2)$$

The angle of GSCH teeth inclination β is a variable value and changes from $\beta_{\min}$ at one end of the wheel to $\beta_{\max}$ the other, $[\beta_{\min} \le \beta \le \beta_{\max}]$. The boundaries of this range β are determined by the dependence:

$$\beta_{\min,\max} = \arccos(\frac{\sqrt{\begin{array}{l} r_t^2 \cdot \cos^2\beta_m \pm \\ \pm b_W \cdot r_t \cdot \sin\beta_m - \\ -0.25 \cdot b_W^2 \end{array}}}{r_t}) = \\ = \arccos\left(\sqrt{\cos^2\beta_m \pm k \cdot \sin\beta_m - 0.25 \cdot k^2}\right). \quad (3)$$

Let's set the limits of change of the auxiliary parameter k corresponding to the found boundaries of the angle range β.

Obviously Fig. 4, b that the extreme values of the angles β for inclined-arched teeth, that is, $\beta_{\min}$ and $\beta_{\max}$, are determined from the conditions: $\beta_{\min} = 0$; $\beta_{\max} = \pi / 2$. Substituting the found values $\beta_{\min}$ and $\beta_{\max}$ into equation (2), we solve it with respect to value k. As a result, the theoretical limits of the coefficient range: $k = [k_{\min};\ k_{\max}]$ for a given average angle of teeth inclination β_m are determined:

$$k_{\min} = 2 \cdot \sin\beta_m; \qquad k_{\max} = 2 \cdot (1 - \sin\beta_m). \quad (4)$$

Naturally, in the practical SG-calculation with inclined-arched teeth should be taken $[k_{\min} < k < k_{\max}]$.

4.2. Experimental Part

For a numerical assessment of the obtained ratios, a calculation was made for the SG with inclined-arched teeth (Fig. 4, c) according to dependences (2)...(4) with the following engagement parameters:

$$b_W = 85 \text{ mm}; \quad m_n = 4 \text{ mm}; \quad \beta_m = 17^0 49' 27''.$$

Calculation results:

$$k_{\min} = 2 \cdot \sin\beta_m = 2 \cdot \sin 17^0 49' 27'' = 0.61;$$

$$k_{\max} = 2 \cdot (1 - \sin\beta_m) = 2 \cdot (1 - \sin 17^0 49' 27'') = 1.39.$$

Theoretical limits of possible GSCH-radii $\left[r_{t(\min)};\ r_{t(\max)}\right]$:

$$r_{t(\min)} = b_W / k_{\max} = 85 / 1.39 \approx 61 \text{ mm};$$

$$r_{t(\max)} = b_W / k_{\min} = 85 / 0.61 \approx 139 \text{ mm}.$$

We accept $r_t = 110$ mm that corresponds to $k = b_W / r_t = 85 / 110 \approx 0.77$.

Axial overlap coefficient, formula (2):

$$\varepsilon_\beta = \frac{85 \cdot \cos 17^0 49' 27''}{3.14 \cdot 4} \cdot [\sqrt{\begin{array}{l}\left(\dfrac{\cos 17^0 49' 27''}{0.77}\right)^2 + \\ + \left(\dfrac{\sin 17^0 49' 27''}{0.77}\right) - 0.25\end{array}} - \\ - \sqrt{\begin{array}{l}(\cos 17^0 49' 27'' / 0.77)^2 - \\ -(\sin 17^0 49' 27'' / 0.77) - 0.25\end{array}}] \approx 2.29.$$

The range of changes in the angle of inclination of the teeth within the width of the gearing $b_W = 85$ mm with $k = 0.77$, formula (3):

$$\beta_{\min,} = \arccos\left(\sqrt{\begin{array}{l}\cos^2 17^0 49' 27'' + 0.77 \times \\ \times \sin 17^0 49' 27'' - 0.25 \cdot 0.77^2\end{array}}\right) \approx \\ \approx 0.08 \text{ rad} \approx 4.5^0;$$

$$\beta_{\max,} = \arccos\left(\sqrt{\begin{array}{l}\cos^2 17^0 49' 27'' - 0.77 \times \\ \times \sin 17^0 49' 27'' - 0.25 \cdot 0.77^2\end{array}}\right) \approx \\ \approx 0.76 \text{ rad} \approx 43.7^0.$$

For comparison, we calculate the value of the axial overlap coefficient in a conventional involute helical gear (hel) with the same gearing parameters b_W, m_n, $\beta_m = \beta = \text{const}$:

$$\varepsilon_{\beta(hel)} \approx \frac{b_W \cdot \sin\beta}{\pi \cdot m_n} = \frac{85 \cdot \sin 17^0 49' 27''}{3.14 \cdot 4} \approx 2.1.$$

As you can see, ceteris paribus, the coefficient of axial overlap of inclined-arched teeth ε_β is almost 11% higher than $\varepsilon_{\beta(hel)}$ in helical gear. This will lead to the fact that a larger number of wheel teeth will be simultaneously engaged in the SG with inclined-arched teeth. As a result, the SG with inclined-arched teeth has a potential advantage over the involute helical gear in terms of load capacity.

5. Conclusions

In this work, the following results were obtained:

1. A set of 3D models of a multi-operational milling machine drive with a gear box of speeds has been developed in the CAD KOMPAS-3D environment. This complex project became the winner of the International Competition "Future Aces of 3D Computer Modeling".

2. Complicated three-dimensional models of housing parts of the carrier system for the portal layout machine have been created. The efficiency of design modeling process is increased due to extended functionality of the three-dimensional operation “Extrude” of its thin-walled element not only at the moment of creation but also at the stages of modification. This is the basis for the multivariate design of machine housing parts variety with increased productivity of the designer.

3. Based on the C3D Modeler geometric kernel, a working profile of a gear wheel formed by the simultaneous extrusion of several closed contours that intersect each other at the algorithmic level is implemented. This gives a computational effect to the construction of the tooth working profile and the junction section in the tooth root. This procedure results in geometrically correct 2D profiles and 3D models generated on this basis.

4. A constructive improvement of the spur gear has been implemented by choosing the shape of the teeth in the form of an inclined-arched profile. The limiting angles of the teeth inclination $[\beta_{\min}; \beta_{\max}]$ and the range of values of the auxiliary coefficient k are determined. According to the boundaries of these parameter values, within which the radius r_t of a standard gear-shaping cutting head is assigned. It is analytically proved and graphically illustrated that the radius of the gear-shaping cutting head r_t and its position relative to the wheel, to be milled cannot be set arbitrarily, since r_t and the width of the gear rim b_W is an interdependent parameter. A comparative calculation of the axial overlap coefficient $\varepsilon_{\beta(hel)}$ in an involute helical gear and ε_{β} in a spur gear (SG) with inclined-arched teeth with the same gearing parameters showed, all other things being equal, the axial overlap coefficient ε_{β} almost 11% more than $\varepsilon_{\beta(hel)}$. This indicates the advantage of the proposed SG-variant in terms of load capacity, and, consequently, in terms of the operability criterion of the drive of the machining center.

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