3D modelling of angular spindle’s head for machining centre

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Abstract. The unified design of the angular spindle head of the machining centre for drilling-milling and boring type is considered. Three-dimensional modeling of the angular head assembly in CAD KOMPAS-3D is performed using a specialized application “Shafts and 3D mechanical transmissions” including a three-dimensional representation of the machining centre kinematic scheme. The photorealistic visualization module Artisan Rendering, which is integrated into the KOMPAS-3D CAD contour, was used to create presentations and promote this construction to the machine tool industry. A two-stage procedure for analyzing the structure by the stiffness criterion is proposed: at the early stages of the design by the matrix initial parameters method; at the stage of the working project by the finite element method in the APM FEM module. The procedure for determining the stress-strain state of the angular head spindle with end mills realized.

1. Introduction

In the conditions of constant increase in the accuracy of manufacturing the products of machine- and instrument making, fundamental research of limiting stiffness parameters of machine tools is important. Higher requirements for machine tools arise during final processing, which forms the stiffness parameters of the workpiece. Therefore, the characteristics of the rigidity of a metal-cutting machine are the main factors determining the effectiveness of its use. The complex forms of hard-to-reach machined surfaces predetermine the specificity of the structures of the shaping spindle assemblies, including the limited dimensions and considerable lengths of tooling overhangs. Traditional assemblies of machining centers of drilling-milling and boring type cannot realize a full cycle of technology for processing housing parts of this configuration. The need to expand the technological capabilities of metal cutting equipment with the use of special angular spindle heads becomes relevance. Often replaced tool blocks mounted in angular heads (AnH), and the requirements for increasing the accuracy of manufactured products lead to the need for research on the criteria for strength and rigidity of the spindle unit AnH.

In the work on the investigation of the forming unit’s rigidity, the designs of machining centers of medium size are considered [1]. For them, a procedure is proposed for "sounding" the compliance rates of the spindle with the tool within the working area machining, taking into account the standard size of the machine. In this case, traditional technological operations of milling with "open" access to the machined surfaces are considered. The study concerned a fixed version of the main drive adjustment for the machine and did not take into account the influence of the tool blocks on changes in the basic rigidity, which is especially characteristic of machining centers.
In work [2], the problems of processing large-sized parts, in the construction of which there are machined surfaces with limited access, are considered. The author in [3] notes that the influence of the basic principles of design and. As a result, the design procedures for making design decisions are significantly different from the methods of designing machines of small and medium sizes. An important feature of the processing of large parts is the consideration of the gain of any source of error. In this regard, issues of machining accuracy and rigidity of the machine forming units are important and relevant.

Complex use of CAD SOLIDWORKS and CAE ANSYS software to build a spindle 3D model and study it by the finite element method is devoted to work [4]. When the spindle is formalized, a 10-node rod model with three degrees of freedom is used in the node. Analysis of dynamic characteristics at 5 natural frequencies and spindle oscillation modes with allowance for changing load makes it possible to predict the behavior of this workpiece outside the resonance zone and to estimate the rigidity index with the domination of bending stresses.

The stiffness of the machine main units affects the accuracy of machining parts in a wide range of dimensions. A new approach based on dynamic characteristics (including dynamic compliance) is used in [5]. The authors form the graphs of stability lobe diagrams to assess the resistance of the cutting process on heavy lathes. It is used as a traditional single-frequency model as a multi-frequency model based on Chebyshev polynomials. This opens up possibilities for determining the threshold values of an effective cutting process. On the basis of lobe plots, the assessment of the asymmetry of the milling machine design on the increase of dynamic stability in a certain range of changes in stiffness parameters was made: 5.6 ... 6.0 · 10⁶, N/mm presented in [6].

At the same time, it should be noted that the research tasks, including the rigidity criterion, are carried out both at the initial stages of the machine tool construction, and at the final stages of the working project. And at the initial stages of multivariate design, rapid procedures for evaluating rigidity are required, while at the stage of the working project it is necessary to conduct a fairly laborious study by the finite element method. In the above works such an approach is not considered.

Statement of the research task. In this research, the problem is formulated by a comprehensive study of the modernized angular spindle head design that extends the technological capabilities of metal cutting equipment and increases the degree of technological operations concentration in one workplace. For basic design of the angular head, a new variant of mounting the double support “Duplex-O”, which increases stiffness in the machining is proposed. To study the performance of the projected angular head by the rigidity criterion, a two-step procedure for evaluating the rigidity is proposed:

1) the procedure for determining the compliance of the AnH spindle node at the stage of the preliminary design sketch with use of the initial parameters in the matrix formulation;
2) the procedure for a complex study for the stress-strain state of the spindle node optimal version for the AnH by the finite element method, adopted as the main one, at the stage of the working project.

2. Three-dimensional modeling of the angular spindle head
The spindle unit AnH, as its main forming component, is a closed dynamic system using modular tooling arbors [7]. To study the stress-strain state and the use of finite element methods [8-10], it is necessary to build 3D models of individual parts and spindle assemblies in the integrated CAD KOMPAS-3D with the FEM APM module [8]. At the first stage of the research 3D models of an angular head in integrated CAD KOMPAS-3D are constructed. In the process of construction, KOMPAS resources such as application libraries (especially "Shafts and mechanical transmissions 3D") and a full-featured geometric C3D core [11], were used, which, in addition to the 3D modeler, supports drawing tools, three-dimensional surface and solid modeling, parametric capabilities, as well as conversion facilities in most major data formats [12].

Bellow there is the developed 3D model of the angular spindle head giving an idea of the layout design (figure 1a) and the features of the installation of support nodes (figure 1b).
To create a photorealistic image and analyze the appearance of the AnH future design, the Artisan Rendering module [13] integrated into the KOMPAS-3D system is used. At the same time, it becomes possible to select materials taking into account the color and texture. It is important to create a subsequent feedback in the course of adjusting the workpiece geometry to improve it. In figure 2 shows the rendering of the complete assembly, giving an idea about the design as a whole (figure 2a) and the device of the spindle assembly (figure 2b).

In the composition of the modular equipment, which allows expanding the technological capabilities of machine tools, often include an angular head. With its help, the milling process is performed in a different range of rotation angles (around the vertical axis at 360°). A feature of the angular head is its use for high-speed machining (at spindle speeds up to \( n = 4000 \text{ min}^{-1} \)), the drive power of the main motion is \( P = 7.5 \text{ kW} \); torque on the motor shaft \( T = 74 \text{ N·m} \); torque on spindle angular head \( T_{ah} = 69, \text{ N·m} \) planes, flats and various grooves in hard-to-reach areas of workpiece. This type of head (figure 1) is equipped with a set of end mills with a diameter in the range 3 ... 25 mm.

To evaluate the stress-strain state of the angular head spindle, it is necessary to analyze the kinematic chains realizing the shaping motion during face milling and the power parameters at each stage of the motion transfer. In figure 3 there is a 3D kinematic diagram of a wide-universal machining center for the second standard size.
Figure 3. 3D kinematic scheme of motion transfer to the angular head.

The rotation from the electric motor through the poly V-belt is transmitted through a two-speed gear box to the camshaft, from which the rotation is transferred to the vertical head coupling [14] or to the gear of the horizontal spindle. Through the vertical head, the motion is transmitted to the angled head, in which end mills are mounted.

During the operation of a bevel gears pair in engagement, their teeth are affected by three mutually perpendicular forces: circumferential $P_z$, radial (thrust) $P_y$, and axial $P_x$. The circumferential and thrust forces for a given angular head are determined by the known formulas:

$$ P_x = \frac{T_{wh} \cdot 10^3}{r_m} = 2516, N; \quad P_y = P_x \frac{\sin \alpha \cos \delta - \cos \alpha \sin \beta \sin \delta + f \cos \alpha_s \cos \delta}{\cos \alpha \cos \beta - f \sin \alpha_s} = 910, N, \quad (1) $$

where $r_m$ – the average pitch radius; $\alpha$ – the angle of engagement, usually equal to 20°; $\beta$ – the angle of inclination of the tooth bevel gear; $\delta$ – half of the pitch cone angle, $\delta = 45°$; for a given pair of bevel gears ($z_1 = z_2 = 20$; $m = 2$ mm; $b = 10$ mm); Material Steel 12HN3A, GOST 454371; teeth to cement $h = 0.5...0.9$ mm; hardness 55HRC (229...269 HB); $f = 0.1$ – coefficient of friction; $\alpha_s$ – the angle of engagement in the end section.

The spindle of the modernized angular head (figure 1) is mounted on two supports: front – bearing 3182108 double row tapered roller bearing with short cylindrical rollers; rear – bearing 246205 – duplexed angular contact ball bearings whose outer rings face each other with wide ends with contact angle 26° [15-17]. The bearing set locks the shaft and housing in both axial directions and provides a stiffer angular fixation of the shaft.

3. Express analysis of the compliance characteristics for the angular head

At the initial stages [18] of the AnH design, when a lot of design alternatives are generated at the layout level, it is necessary to develop methods and procedures for quick evaluation of output technical characteristics. For the express estimation of the compliance of the AnH spindle, we use the approach proposed in [3], which takes into account the variety of tooling standard sizes and a limited number of design variants of the spindle (characterized by the length of the inter-support part $\ell$, the length of the cantilever part $\ell_1$, and the diameter $d$). For this, a static formulary is formed in the view of a two-component analytic dependence. The first component is represented in the elastically deformational model in the view of a static form $\delta = (\Delta_2 + \Delta_3)$, i.e. in the analytical dependence form of the displacement of supports $\Delta_2$ and the own spindle $\Delta_3$ on the length $\ell_1$ of the console (tool milling unit AnH). To construct it, we use a system of four linear equations with boundary conditions [3]. The
second component is an alternating component that can be tuned to different types of manufactured products, represented as an instrumental milling unit with different diameters of end mills. For this AnH construction in the mathematical environment of MAPLE is obtained in the symbolic form of the magnitude of the reactions and moments on the back and front supports \( \{R_z, R_p, m_z, m_p\} \):

\[
R_z = -0.042 + 0.0029 \ell_k; \quad R_p = 0.95 + 0.0029 \ell_k; \quad m_z = -24.44 + 0.0057 \ell_k; \quad m_p = 91.18 + 0.74 \ell_k.
\] (2)

The program for constructing a static formulary for the considered construction in the MAPLE environment is presented in [9]. The calculated reaction values allow determining the deflection \( \delta(x) \) and angle of rotation \( \theta(x) \) at the cantilever end of the spindle \( (x = \ell + \ell_1) \) and form a static form as a function of the length \( \ell_k \) of the conditional console:

\[
\delta = - (1377 + 9.58 \ell_k + 0.062 \ell_k^2) \times 10^{-8}. \] (3)

4. Complex engineering analysis of stress-strain state

At the stage of the working project, after selecting the optimum design variant of the projected product, a comprehensive analysis of the AnH spindle for the machining centre SF68VF4 is required (figure 3). In the environment of the APM FEM module [8, 19], we simulate the design of the AnH by the rigidity criterion. This module is equipped with a finite-element grid generator included in the CAE-library, which implements finite element method (FEM) for engineering solutions. In the process of modeling, fastening is carried out in the front and rear supports [20, 21], as well as the applied loads (figure 4a); the coinciding faces are determined (for the CE-analysis of the assembly); the generation of the grid is realized (figure 4a) with use the MT Frontal method (using multi-core processor); calculation and viewing of results in the form of stress maps and displacements is performed. In the process of FEM, it is possible to evaluate and analyse the decomposition for different values of the viewing depth (figure 4b).

![Figure 4. Procedures of the finite element method: a - finite element grid; b - depth of view.](image)

**Figure 4.** Procedures of the finite element method: a - finite element grid; b - depth of view.

Within the environment of the APM FEM module, all of the above actions have been implemented and obtained for the case of limiting cutting modes:
- fields of equivalent stresses according to Mises (fourth theory of strength), presented in figure 5;
- displacement fields (figure 6) on the set of spindle cross sections;
5. Discussion

The designer often needs to quickly perform a static calculation of a particular adjustment. This can be done using a nomogram consisting of two parts: a static formulary (figure 7a), constructed according to the formulas (2) for three possible values of the diameter and the compliance diagram of the cantilever (figure 7b), constructed for cantilever of constant cross section and different values cantilever diameters $d_k$. The cantilever is considered as a beam built in in the supporting section and loaded at the cutting point by a unit force.

In order to determine on the nomogram (figure 7) the compliance of the spindle node reduced to the place of cutting, it is sufficient to know the distance to the cutter $l_k$ and the diameter $d_k$ of the cantilever. The compliance at the tool is determined by the sum of the values found on the nomogram (figure 7a and 7 b).

In some cases inaccuracy of determination to general compliance spindle’s node is connected with absence in accounting model some data about springy character bearing support. We shall consider, what influences upon results of the steady-state calculation inaccuracy in determination of linear compliance of one bearing [22-24]. For spindle’s nodes machine tool to models SF68VF4 design value to linear rear support compliance $A_z = 3.99 \times 10^{-6}$, mm/N and linear front support compliance $A_p = 3.93 \times 10^{-6}$, mm/N. We shall take else two values of linear compliance with deflections:

1. $A_{z1} = 2.59 \times 10^{-6}$ and $A_{z2} = 5.39 \times 10^{-6}$, mm/N; 2. $A_{p1} = 2.55 \times 10^{-6}$ and $A_{p2} = 5.305 \times 10^{-6}$, mm/N,
and the angular compliance of one bearing:

1. \( a_{t1} = 0.039 \times 10^{-8} \) and \( a_{t2} = 0.08 \times 10^{-8}, 1/(N\cdot mm) \);
2. \( a_{p1} = 0.039 \times 10^{-8} \) and \( a_{p2} = 0.08 \times 10^{-8}, 1/(N\cdot mm) \)

Dependencies \( \delta \) from \( l_k \) is built on figure 8; from which follows that even significant (before 35 \%) from-bending to linear softness \( A_0 \) one bearing in both sides (over-state or under-state) from nominal of value little influences upon softness of the system spindle – cantilever as a whole. If we consider an approximate 20\% deviation of the linear and angular compliance, then on the basis of the constructed dependences of \( \delta \) on \( l_k \), it can be argued that even a significant (up to 35\%) deviation of the linear compliance \( A_0 \) of one bearing in both directions (overstating or understating) from the nominal value has little effect on the flexibility of the spindle – cantilever system as a whole [25, 26]. At the same time, if one does not take into account the angular rigidity of a single bearing \( (a_0 \to \infty) \), then the angular compliance of the single bearing is taken into account. The estimated value \( \delta \) of the spindle unit increases by 60\%. In addition, as can be seen from the calculations, the change in the linear compliance of \( A_z \) and \( A_p \) of the spindle supports in a sufficiently wide range without a corresponding change in its diameter does not significantly change the compliance of the spindle unit (does not exceed 25\%).

![Compliance analysis](image)

**Figure 8.** Compliance analysis of spindle’s node.

To evaluation the proportion of individual elements deformations for the spindle’s node, it is expedient to analyse the balance of elastic displacements of the components in the working zone of cutting. For the spindle AnH with \( d = 40 \) mm with a cantilever tool, the proportion of spindle and supports deformation is 37\%, the conical connection of the arbor and the end mill is 52\% and the own arbor is 11\%.

6. Conclusion

In the environment of the integrated system KOMPAS-3D, the 3D project was developed for a modified angular spindle head of the machining center using the “Duplex-O” scheme in its support. A comprehensive study of the angular head design on the analytical dependencies basis of its compliance at the initial stages of multivariate design was carried out. The balance of the components compliance for the spindle assembly was revealed: the maximum value for a milling arbor with an end mill – 63\% of the total compliance for a spindle node with supports \((d = 40)\) mm accounts for 37\%. A matrix model of a unified spindle assembly was constructed, taking into account the general properties of two-support structures on angular contact bearings. With the help of such a model, a static form is generated in a symbolic form in the MAPLE software environment. With its help, graphs of
compliance of the spindle node components were constructed and quantitative estimates of the specific displacements were obtained: arbor-cutter – 0.0018 mm/N; spindle-support – 0.00039 mm/H. An assessment of the stress-strain state for the spindle node is given, as well as stress and displacement fields are constructed by the finite-element method in the environment of the APM FEM module. The fields of stresses and displacements for various cross sections of the spindle, loaded with cutting forces during end milling and by the forces arising in the conical transmission of the angular spindle head are obtained.

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