Modeling of vertical spindle head for machining center

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Abstract. The design of the vertical spindle head, which is part of the modular technological equipment of horizontal drilling-milling and boring machining centers, is considered. Three-dimensional modeling of parts and assembly units of the spindle head has been performed in the environment of integrated CAD system KOMPAS-3D. The effectiveness of full-featured specialized applications in the design process of the structure main components in the KOMPAS environment is noted. Built-in “Shafts and mechanical transmissions” applications for creating a vertical spindle head model the design procedures in the shortest possible time are provide. An assessment of the forming spindle assembly dynamic functioning quality on which external loading acts, with probabilistic nature is made. The spectral analysis method is used to evaluate the dynamic characteristics of the structure under consideration.

1. Introduction

In the range of metal-cutting equipment, the share of high-speed machines and machining centers with advanced technological capabilities is constantly increasing. In contrast to specialized machines, the design of machining centers (MC) is formed both by a set of specific aggregates and by the principle of their construction – without “hard” kinematic connections between their aggregates [1, 2]. Many machining centers are equipped with additional modular equipment, which is associated with the principle of technological operations concentration at one workplace. So, a drilling-milling and boring type machining center SF68VF4 model with a horizontal spindle is complemented by additional equipment in the form: disk package of cutters, slotting, angular and vertical spindle heads [3, 4]. A replaceable vertical spindle head provides various technological operations for processing spatial surfaces of complex shape, milling flat drilling surfaces and drilling holes with an exact coordinate location, etc.

A study of the functioning effectiveness of the forming components for metal-cutting equipment according to the criteria of rigidity and vibration resistance is carried out in various integrated Computer Aided Design (CAD) systems. It is necessary to connect 3D modeling systems of the forming components with calculation modules for evaluating structures for rigidity and vibration resistance, which have the greatest impact on the accuracy and quality of products. This is especially important for modern machine tools designed for finishing and precision machining of high accuracy parts. The share of such machines in the fleet of machine-building enterprises is increasing, which indicates the relevance of methods for assessing vibration resistance. Such a situation from the methodological point of view necessitates the parallel use of three-dimensional modeling tools with parameterization [5, 6] and research methods in the field of machine-forming dynamic quality for the structures units and its tooling.
In [6], the analysis of the dynamic quality of functioning was carried out on the basis of the matrix method of initial parameters by constructing frequency forms of forming units (FU) – the dependences of the output characteristics for the FU elastic units on the tool and workpiece movements with a variable input load. On the basis of the forms obtained, an express procedure for determining the stability limits for the machine elastic links and the cutting process becomes possible. At the same time, such modeling in the time domain of the forming parts process does not allow us to determine the limits of FU structures stability.

The rigidity of the machine main components affects the accuracy of machining parts in a wide range of sizes. A new approach based on dynamic characteristics (including dynamic correspondence) is used in [7]. The authors propose a procedure for plotting lobe stability graphs (the Stability Lobe Diagram, SLD method) to evaluate the stability of the cutting process on heavy lathes. The SLD method [8] is used to analyze the vibration resistance of the machining center spindle as a rotating system with two degrees of freedom. The simulation results determine the critical mass of the imbalance of the rotating disk value depending on the spindle speed.

The dynamics of the milling machine formative nodes is considered in [9]. The authors focused on the influence of the layout, taking into account the configuration of the axes, in particular the axis of high-speed spindle rotation and fast feed of the workpiece. The advantage of this work is the creation of a database system (DB), including a variety of layouts and configurations of the axes. The composition of such structured databases contains separate sections in which the main properties of the forming units are presented: spindles with cutters and tables with workpieces. In such a structure for presenting information, a place has been found, both 3D models of the machine main formative nodes, analytical models, and arrays of experimental data.

The authors of [9] put forward the idea of a modular modeling system based on three-dimensional models, on the basis of which the machining process stability is predicted within the working area of the milling machine. At the same time, it is unclear from the article’s text how information on random components of input signals (cutting forces) is presented in databases and how the analytical processing of random impact components on machine nodes is carried out.

Improving the process of determining the dynamic quality of the functioning for the vertical spindle head design based on the comprehensive 3D modeling procedure and assessing the dynamic characteristics of the structure under study are development.

2. 3D modeling of a vertical spindle head
The main forming unit – the vertical spindle is mounted in the pinol (it moves in the housing (figure 1) with a maximum stroke of 90 mm), which receives rotation through a conical pair and a cam clutch (figure 2). The vertical spindle head is mounted rotatable 90° in both directions (due to the presence of a T-slot in the adapter plate) [10–12]. This design and layout of the machine spindle node is most effectively used in high-speed technological operations, such as finishing milling and boring. At the same time, machining of products may be accompanied by a high level of vibrations, which affects the quality of products. To research the working capacity and evaluation dynamic quality during machining, three-dimensional models are created and vibration stability based on the spectral analysis apparatus is calculated.

In order to carry out a comprehensive procedure for studying the dynamics of a spindle node and obtain numerical estimates, it is necessary to build solid-state models [13, 14] using CAD COMplex of Automated System (KOMPAS) designed by the company ASCON [15–17]. We use the bottom-up principle, i.e. first, we will build 3D models (figure 1) of the component parts (housing, plate, etc.) with their subsequent integration into the assembly design of the vertical head (figure 2). Three-dimensional modeling of the unit under consideration was carried using specialized system applications, in particular, the application “Shafts and 3D mechanical transmissions” [18].

To achieve a photorealistic image of the structure, the vertical spindle head was rendered in the Artisan Rendering module, which is integrated into CAD KOMPAS. The module’s capabilities make it
possible to obtain a high-quality image of a vertical head (figure 3) by combining material and lighting, a background and a scene based on a three-dimensional model [19–21].

**Figure 1.** 3D models of the vertical spindle head parts: a – housing; b – plate.

**Figure 2.** 3D model of a vertical spindle head: a – general view; b – section.

**Figure 3.** Rendering of a vertical spindle head: a – general view; b – section.
3. Analysis of the elastic system dynamics for MC spindle

Increasing the efficiency of cutting processes is associated with the achievement of the dynamic stability of metal cutting machines [22–24]. An analysis of the balance of flexibility and vibration patterns of the main nodes of the machining centers of the drilling-milling and boring group showed that the most intense vibrations are characteristic of the main forming nodes: spindle – arbor – tool and table – workpiece. Milling as an operation with intermittent cutting is characterized by a large range of force and disturbing influences that occur during processing, including the probabilistic component [25–27].

In practice, the analysis of random processes during mechanical processing is carried out by numerical methods, in particular using the spectral analysis apparatus and the fast Fourier transform method [28-30].

The spindle unit (SU) of a vertical spindle head equipped with a boring arbor is represented as a beam on elastic supports with viscous damping and is divided into sections delimited by spasmodic changes in the moments of inertia of the sections. The SU structural diagram is shown in the figure 4.

![Figure 4. Spindle assembly design.](image)

At the first natural frequency, the unit compliance obtained by the Fourier transform from the displacement (deflection) of the rod in the 0-th section takes on the value:

\[
y_0(j\omega) = H(j\omega) = \frac{1/k}{1 - \left(\frac{\omega}{\omega_n}\right)^2 + jl \frac{\omega}{\omega_n}} = \frac{2.57 \cdot 10^{-5}}{(1 - 6.96 \cdot 10^{-5} \omega^2) + j 3.34 \cdot 10^{-4} \omega},
\]

where \(\omega\) – angular frequency; \(\omega_n\) – self-resonant natural frequency; \(jl \frac{\omega}{\omega_n}\) – imaginary part of the transfer function; \(l = \frac{2\pi}{T}\); \(k\) – stiffness of spindle node; \(m\) – reduced mass.

To simplify the problem, in the case when the damping forces are proportional to the velocity of the transverse movement, and the sections of the rod between the concentrated masses are considered weightless, the dependence between the Fourier transforms on the parameters at the boundaries of the “0-1” section will be:

\[
y_1(j\omega) = U_1 G_0 y_0 - S_0,
\]

where \(S_0\) – the matrix taking into account the influence of the concentrated force \(F_0(t)\):

\[
S_0 = \begin{bmatrix} 0 & 0 & \frac{F_0(j\omega)T^4}{EI} \\ \end{bmatrix},
\]

where \(F_0(j\omega)\) – the Fourier transform of the force \(F_0(t)\).

The transition matrix \(U_1 G_0\) takes into account the rigidity of the massless portion of the bar \(U_1\) and the concentrated mass of the cantilever (boring bar) \(G_0\):
The value \( y_1(j \omega) \) assumes a complex value:

\[
\begin{align*}
    y_1 &= \frac{2.57 \cdot 10^{-5} - 1.8 \cdot 10^{-9} \omega^2 - 9.46 \cdot 10^{-19} \omega^3 - 1.32 \cdot 10^{-18} \omega^4}{(1 - 7 \cdot 10^{-5} \omega^2)^2 + 2.79 \cdot 10^{-8} \omega^2} - j \frac{4.3 \cdot 10^{-9} \omega + 5.65 \cdot 10^{-15} \omega^2 + 3.16 \cdot 10^{-18} \omega^3 + 3.96 \cdot 10^{-19} \omega^4}{(1 - 7 \cdot 10^{-3} \omega^2)^2 + 2.79 \cdot 10^{-8} \omega^2}.
\end{align*}
\]

4. Discussion

At the front end of the spindle (in the zero section), disturbances from the cutting process act – force \( F_0(t) \). When solving the problems of the technological systems dynamics, the most common law for changing the force \( F(t) \) as an input characteristic is the harmonic law:

\[
F(t) = F_0 \sin(\pi ft).
\]

As is known [11], the frequency response of a system is defined as the Fourier transform \( Y(f) \) of the system reaction (in this case, the displacement \( y(t) \) on the impulse action of the force \( F(t) \)). In terms of the amplitude \( |H(f)| \) and phase \( \varphi(f) \) characteristics, taking into account the fact that the Fourier transform of the pulsed action of the force \( F(t) = \delta(t) \) is equal to unity, is the following expression:

\[
|H(f)| = 2 \frac{l_1}{(1 - (f/f_n)^2)^2 + (2l_1f/f_n)^2}; \quad \varphi(f) = \arctg \frac{2l_1f/f_n}{1 - (f/f_n)}
\]

In the expression (1), the notation:

\[
l_1 = \frac{h}{2\pi km}; \quad f_n = \frac{1}{2\pi \sqrt{km}},
\]

where \( l_1 \) – dimensionless quantity characterizing the damping of oscillations;

\( f_n \) – natural frequency of undamped oscillations (in Hz).

Note that the dimension \( |H(f)| \) coincides with the dimension of compliance, mm/N.

The considered spindle unit of the vertical head [12] (figure 2) has the following parameters of the dynamics elastic link for basic equation: {reduced mass \( m = 25 \text{ N} \); damping coefficient \( h = 13.96 \text{ Ns/mm} \); stiffness \( k = 38900 \text{ N/mm} \)}. Such a constructive variant is characterized by such numerical values of the coefficient \( l_1 \) and the first natural frequency \( f_n \): \( l_1 = 0.02; \quad f_n = 19.86 \text{ Hz} \). It is important to note that the amplitude characteristic \( |H(f)| \) at \( l_1 \leq 1/\sqrt{2} \) has a maximum at a frequency somewhat lower than the natural frequency \( f_n \). The resonance frequency \( f_r \) obtained by minimizing the denominator of the expression \( |H(f)| \) (1) takes the value: \( f_r = 19.84 \text{ Hz} \). For the case of harmonic input action \( x'(t) \) expressed
in units of displacement \( x^*(t) = \frac{F(t)}{k} \), the one-sided spectral density \( G_{xx} \) and the one-sided spectral density of the output signal \( G_{yy} \), as a real even function of \( f \), are determined from the following dependencies:

\[
G_{xx} = \frac{F_0^2 \delta(f - f_0)}{k^2 \cdot 2};
\]

\[
G_{yy} = [H(f)]^2 \cdot G_{xx} = \frac{F_0^2 \delta(f - f_0)}{2[1-(f/f_0)^2]^2 + \left(2l/f_n\right)^2} \times \delta(f - f_0) = 8.12 \cdot 10^4 \cdot \delta(f - f_0),
\]

where \( \delta(f - f_0) \) – delta function localized at \( f = f_0 \);

\( |H(f)| \) – the amplitude characteristic of the elastic spindle link, mm/N;

\( f_0 \) – cyclic frequency, Hz;

\( f_n \) – natural frequency of undamped oscillations, Hz;

\( l \) – attenuation coefficient.

The corresponding covariance functions \( R_{xx}(\tau) \) and \( R_{yy}(\tau) \) – even functions of \( \tau \) (time shift) are:

\[
R_{xx}(\tau) = \frac{F_0^2}{k^2 \cdot 2} \delta(f - f_0);
\]

\[
R_{yy}(\tau) = \frac{F_0^2 \pi \frac{l}{f_n} e^{-2\pi f_0 |\tau|}}{2 \cdot 4l} \cdot \cos(2\pi f_n \sqrt{1-l_f^2 |\tau|}) + \frac{l}{\sqrt{1-l_f^2}} \cdot \sin(2\pi f_n \sqrt{1-l_f^2 |\tau|}) \bigg|_{f_0 = f_0}.
\]

The graphs of the functions \( x^*(t) \) and \( R_{yy}(\tau) \), the probability density of the input signal \( p(x) \) and the distribution function \( P(x) \) are presented in figure 5.

**Figure 5.** Frequency characteristics of the elastic system "spindle-arbor-tool": a – input impact; b – covariance function; c – probability density; d – distribution function.

In figure 5a the graph of the harmonic input influence: \( x^*(t) \) is presented: \( x^*(t) = \frac{F(t)}{k} \), where \( k \) is the stiffness of spindle node. The covariance function for parameter \( x^*(t) \) characterizes the degree of variability (the relationship between the values of the changing random process for the cutting force at different points in time) is shown in figure 5b. For nonzero means for the vertical spindle head study considered in this paper, an exponentially cosine covariance function \( R_{yy}(\tau) \) is obtained. Figure 5c and 5d show the probability density and distribution function of the input characteristics, which allow not only to evaluate the probabilistic structure of the process, but to analyze the extreme values.
Conclusion
A complex procedure for studying the functioning dynamics of a vertical spindle head using the specialized application “Shafts and 3D mechanical transmissions” and a spectral analysis apparatus is proposed. Solid-state models of the spindle head design in the KOMPAS-3D system and rendering was performed in the Artisan Rendering module. The frequency characteristics of the spindle-arbor-tool unit elastic system, including one-sided spectral densities, covariance functions of the elastic system input and output signals are estimated. The resonant frequency of oscillations in the low-frequency range, which does not coincide with the first natural frequency is found. This makes it possible, when assigning cutting modes, to exclude such cutting speeds of the spindle and the nearby range falling into the zone of resonant vibrations. The resonance frequency obtained above \( f_r = 19.84 \, \text{Hz} \) makes it possible to determine the critical values of the spindle speed of the SF68VF4 machine. In accordance with the passport data, in stepwise regulation of the main movement drive in the range from \( n_{\text{min}} = 20 \, \text{rpm} \) to \( n_{\text{max}} = 4000 \, \text{rpm} \) for \( f_r = 19.84 \, \text{Hz} \), the critical frequency is \( n_c = 1190 \, \text{rpm} \). With the denominator of the geometric progression of a motor rotation frequencies series by equal to \( \varphi = 1.26 \), the critical frequency is \( 1250 \, \text{min}^{-1} \). In the case of continuous regulation, the frequency range is \( 1125 \ldots 1375 \, \text{min}^{-1} \), which must be avoided during the manufacturing of details on such machines.

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