Research of vibration stability for multioperational machine by the D-partitions method

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Abstract. The stability of operation, vibration-free processing of products various types at high-speed machining centers suggests the improvement of research in the field of the dynamics of machine tools and their main forming units. The three-dimensional models of the forming spindle node and the corner table of the machining center for the drilling-milling-boring type in the CAD environment KOMPAS-3D have been developed. An algorithm for analysis the dynamic quality of the machine elastic parts based on the constructed static and frequency formular in the mathematical environment MAPLE is proposed. Programs have been developed and graphs of the main frequency characteristics and the hodograph of the transfer function for spindle node in the specialized Signal Processing module have been obtained. The method of D-partitions using to estimate the vibration stability of the forming units on example of 4 coordinate milling machining centres is introduced. Experiments on two variants of technological adjustments with different dimensions of the tool block (the cantilever part of the spindle assembly) were carried out. The analysis of stability in the plane of the cutting process optimized parameter (specific cutting force) was carried out. The region and the boundary of the stable functioning of the machine and, on this basis, the optimum modes of machining by the criterion of vibration stability are revealed.

1. Introduction

Increasing requirements for the quality of drilling-milling-boring type machines and their technological complexes in connection with the general increase in working speeds of machining in mechanical engineering leads to find ways for improve the basic forming units that have a decisive influence on the productivity and dynamic quality of machine designs.

The introduction of progressive machining rate, minimizing the idle time and auxiliary displacement leads to a significant increase in the speed characteristics of the spindle units for machine equipment. These basic technological characteristics of metal-cutting equipment are limited, as a rule, by their vibration stability, for the evaluation of which knowledge of the dynamic characteristics of multi-operational machines and its basic elements is necessary.

In addition, the increase in the efficiency of the processes implemented on modern technological complexes based on multi-operational equipment is associated with an increase in the dynamic stability of the main assemblies of machine tools. An analysis of the balance of compliance and forms of forming unit's oscillation for multi-operational drilling-milling-boring machines showed that the most intense oscillations are characterized by such forming units, such as "spindle-arbor-tool" and "table-workpiece".

In the study of the structures of the forming units according to the criterion of vibration stability, the need arises to construct methods and models for evaluating their frequency characteristics, as well as building 3D models in advanced computer-aided design systems. Solid modelling is efficiently implemented in integrated CAD KOMPAS-3D, developed by the ASCON group of companies.

In the fundamental work on the machine tools dynamics [1], a system of dynamic quality factors for machines (margin and index of stability, speed of response and deviation of dynamic system parameters under external influences) is introduced for the first time. There is also formed general methodology for theoretical and experimental analysis and evaluation of machines on these factors is given. A postulate on the closure of the machine dynamic system, which is determined by the interaction of the elastic system elements for the “machine-tooling” with the working processes of cutting, friction, is introduced.

The approach proposed by prof. V.A. Kudinov turned out to be productive in various applications of the machine tool industry. Thus, in [2], the dynamics of auxiliary motion mechanisms, in particular, the mechanisms of periodic nodes rotation (Maltese, cam lobe, gear-lever and rocker mechanisms) were considered. In [3], the study of the special diamond boring machines dynamics is carried out and the frequency characteristics of boring bars of diamond boring heads are determined. For these structures, proposed by a model on the basis of which the Amplitude phase-frequency characteristic (Frequency response, FR) of the elastic system “spindle-console” (SpC) with an oscillation damper was built. Based on the results obtained in [4], the dynamic characteristics of the precision boring process are determined. The chip time constant T_p and the specific cutting force (proportionality coefficient) K_p are determined based on the constructed nomograms, which makes it easier to calculate the characteristics of the cutting process.

In high-speed spindle heads, the criterion of vibration stability and the phenomenon of buckling is decisive. Such a problem for a torque shaft transmitting torque was considered in [5]. The author estimates the moment of stability disturb for the deformed and rotating shaft depending on the axial force and torque. Using SolidWorks Simulation CAD software allowed us to obtain the value of the critical load, estimate the conditional time in which a sharp increase in the displacement of the shaft sections and fix the corresponding angular velocity interval. At the same time, the influence of the shaft supports characteristics on the behaviour of the structure under study during loading is not considered.

A number of papers [6, 7, 8] are devoted to the study of the elastic system dynamics “spindle–arbor–tool” (SAT). In studies of the spindles vibration stability for milling machines, the elastic spindle–arbor and arbor– tool was often not taken into account when constructing dynamic models. However, these compounds form a significant proportion of errors inherent in the machines of this group.

In [9, 10], a comprehensive study of the design dynamics for the milling machining centre of the portal layout, equipped with a rotary table, was performed. 3D-models of the machine, as the basis of the performance research according to the criteria of rigidity and vibration stability are developed. The author proposes a modular simulation system that allows to determine the basic frequency characteristics and estimate (predict) the limiting values of vibration stability. The concept of modifying the position of the spindle axis is introduced by the example of two positions of the spindle head relative to the Y axis (vertical axis of the machine). To estimate the relative dynamic admittance in the tool–workpeace system, the spectral density characteristics G_{xx} and G_{yy} are used to determine the stability limits, in accordance with the approach [11, 12]. The resulting graphs illustrate the main dynamic effect along the Y coordinate at a frequency of 113 Hz, which in turn excites oscillations along the X coordinate (transverse oscillations). The maximum frequency peaks of the spectral density function G_{xx} are observed at the maximum spindle head position along the Y axis. In addition, the predicted phase responses indicate smaller phase shifts for the lower position Y, which is especially characteristic for rotational speeds in the range from 1800 to 2200 min^{-1}. In this range, the stability limit for higher positions amount from 8 to 20 mm, while the limit for lower stability is much higher than 20 mm during the full period of oscillation. At the same time, the issues of determining the optimal cutting parameters (cutting depth), which are on the border of stable operation, are considered

in [9] as a recommendation for determining the range of values of the ship thickness (a_p = 10 ... 13 mm).
The calculation of the stable functioning characteristics and the identification of signs of the real part of the characteristic equation roots in the complex plane is a rather complicated and time-consuming process. Procedure based approach, when bypassing the calculation of the roots themselves, allows determining the distribution of roots in the complex plane relative to the imaginary axis was proposed in [13, 14]. The proposed method of partitioning the space of coefficients is called the method of D-partitions.

Statement of the research task. The aim of this work is to improve the procedure for determining the conditions for the stable functioning of metal-cutting equipment based on the D-partitioning method. To achieve this goal it is necessary to solve the following tasks:

1. Create 3D models of forming nodes that implement the main movement and feed.
2. Develop an algorithm for analysing the dynamic quality of the machine elastic system
3. To form a mathematical model of a two-mass system for elastic parts of the machine based on the static and frequency forms of the spindle node.
4. Apply the D-partitioning method to analyze the dynamics of the machine elastic system and identify the optimal cutting parameters according to the stability criteria.

2. The procedure for building a one-parameter D-partitioning of a two-mass system

As is known [14], the method deals with the characteristic equations of elastic systems with variable coefficients, in which the roots of these equations change. In this case, the roots are located on the imaginary axis only when a point in the space of coefficients is located on the surface, which represents the desired equation. When a point crosses a surface, the roots move from one half-plane of the roots to another, i.e. the surface divides the space of coefficients into domains *D(m)*, each point of which corresponds to a characteristic equation having a certain number of roots in the left and right side of the roots plane.

Thus, for the second-order equation, in the space of coefficients, three domains can be selected: *D(m)* = {*D(2); D(1); D(0)*}. Here *m* is the number of the equation roots in the right half-plane. The last domain *D(0)* is the domain of stability. It is this partition of space into areas with different values of *m* peculiar to the method of D-partitions. The transition through the D-partition boundary corresponds to the transition of the equation roots through the imaginary axis. In a similar way, one can construct D-partitions in the space not of the equation coefficient, but of the system parameters, on which the coefficients of the characteristic equation depend.

In this paper, we study the algorithmic aspects of the D-partitions method and use it in optimization problems of machine systems. Its use is expedient in processes whose functioning is described by equations of the first…third orders. For equations that are higher, instead of the usual three-dimensional space, one has to consider multidimensional space and a hypersurface that divides this space into regions, which greatly complicates the task, and analysing it loses its visibility.

Consider the procedure for constructing one-parameter D-partitions using the example of a two-mass system of elastic links: “spindle-cantilever” (SpC) and “table-workpiece” (TW) of four-axis SF68VF4 drilling-milling-boring machine, taking into account the dynamic characteristics of the cutting process as an inertial link of the first order.

In the technology of the design process, the procedures for constructing 3D models and parametric representations of parts and assembly units becomes important. For the analysis of the design efficiency, the choice of the optimal version of the project and its research on vibration, 3D models of the forming nodes of the SF68VF4 machine (figure 1) were created in the COMPAS-3D CAD system [15, 16, 17].

The forming spindle assembly is built into the two-stage drive of the main movement. In coordination mode with the corner table, it provides processing of parts from all sides, as well as coaxial boring of holes without remounting the workpieces. The spindle node of this multioperational machine (figure 1a) is considered as a beam on two elastic supports, each of which is mounted on duplexed angular contact ball bearings mounted according to the “Tandem-O” scheme with a preload in the form of different-height bushings.

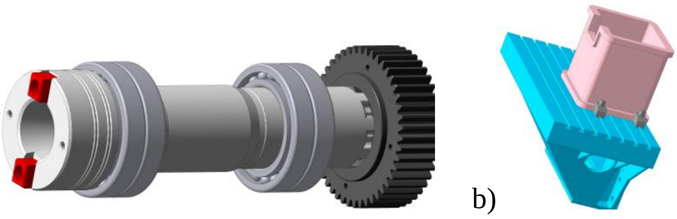

Figure 1. 3D models of forming nodes: a – spindle node; b – node: table-workpiece.

Consider the procedure for constructing a one-parameter D-partition using the example of a two-mass system of elastic links: SpC and TW of the four-axis drilling-milling-boring machine (processing centre) based on the SF68VF4 model taking into account the dynamic characteristics of the cutting process as inertial link of the first order.

The transfer function *W(p)* of an elastic two-mass system in terms of the disturbing effect F_0 in the operator notion after the corresponding transformations [18, 19] takes the form:

$$W(p) = \frac{Z(p)}{F_0} = \frac{A(p)\cdot(T_p p + 1)}{K_p\cdot(A(p) + B(p)) + (T_p p + 1)\cdot A(p)\cdot B(p)}, \quad (1)$$

where *Z(p)* is the output parameter of the system – the displacement of the elastic links; *T(p)* – the chip formation time constant [1]; K_p – proportionality coefficient [1], the value of which depends on the temporary resistance designation σ_0, chip shrinkage factor and parameter of the width of the shear layer during cutting; *p* – Laplace operator; $A(p) = m_1 p^2 + h_1 p + k_1$; $B(p) = m_2 p^2 + h_2 p + k_2$.

In the last two expressions, the reduced masses of the SpC elastic links (m_1) and TW (m_2) are fixed, as well as their damping coefficients (h_1, h_2) and stiffness (k_1, k_2).

Taking into account the rather complex interaction of the two elastic systems of the SpC and TW links in the working mode, a block diagram of the elastic system based on the SF68VF4 machine with the transfer function is constructed as a dependence of the components transfer functions for the individual elastic links:

$$W(p) = \frac{W_1(p)\cdot(1 - W_3(p))}{1 - W_2(p)\cdot W_3(p)},$$

where $W_1(p)$, $W_2(p)$, $W_3(p)$ reflect the transformations: the disturbing force F_0 to the spindle displacement component $W_1(p)$; the resultant displacement of the workpiece under the action of cutting forces in the displacement of the SpC link $W_2(p)$; the resulting displacement of the SpC link in the second component of the displacement of the workpiece $W_3(p)$ – reflecting the dynamics of the TW link.

To determine the components of expression (1), we implement the following procedure.

1. Define the compliance of the elastic links of the SpC and TW for machine SF68VF4 [20–22]. The spindle-cantilever unit (figure 2, a) provides a wide range of technological operations, including the use of high-speed drilling heads. The initial data of this unit are given in the table 1, and the design diagram in figure 2, b.

Table 1. Initial data of spindle node.

Size, mm						
l	l_k	l_0	l_1	l_f	***d***	d_1
312	70	74	52	12	60	35

Figure 2 shows the linear and diametric characteristics: *l* – inter support distance; l_k – length of the spindle cantilever part; l_f – length of the flange; l_0 and l_0' – dimensions of the binding of the right end to the place of forces application and the right end of the elastic SpC link; d, d_1, d_f, d_k - respectively, the outer, internal spindle diameter, flange and cantilever diameters; R_f, R_b, m_f, m_b – reactions and moments in the front and rear spindle bearings; A_f, A_b, a_f, a_b – linear angular compliance of the front and rear supports; *R* – unit force applied to the standard point.

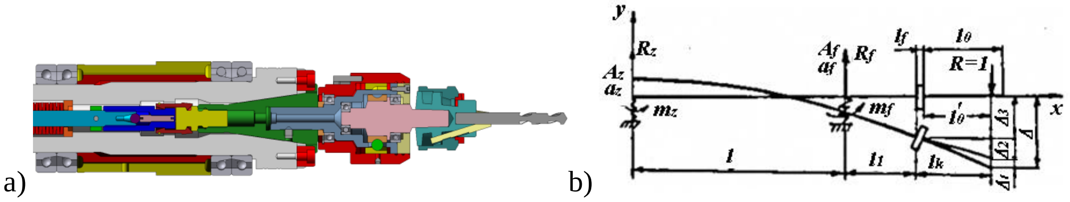

Figure 2. Elastic link Spindle-cantilever: a – 3D model; b – design scheme.

3. Result

To determine the overall compliance Δ using the method of the deformation compatibility, we first calculate the static formulary *(*$\Delta_2 + \Delta_3$*)*, mm/N of the SpC elastic link in mathematical environment MAPLE [3]

$$\Delta_2 + \Delta_3 = \left(1460 + 10{,}64 \cdot l_k + 0{,}037 \cdot l_k^2\right), \tag{2}$$

where Δ_2 and Δ_3 are the spindle compliance terms caused by the linear compliances of the supports Δ_2 and the angular compliance spindle Δ_3 considered as a two-support beam.

Taking into account the deflection of the cantilever $\Delta_1 = 1.27 \cdot 10^{-7}$ mm/N and its size $l_k = 70$ mm, we obtain the value of the spindle node compliance Δ (mm/N), reduced to the cutting place:

$$\Delta = \Delta_1 + \Delta_2 + \Delta_3 = 2.75 \cdot 10^{-5}.. \tag{3}$$

2. Let us estimate the SpC dynamic characteristics using the frequency formular [3], which allows determining the natural frequency ω and the logarithmic decrement of vibrations. The frequency equation with the assumption of the representation of SpC as a weightless beam with one degree of freedom and with three concentrated masses without inertia is:

$$1.13 \cdot 10^{-16} \omega^6 - 14.46 \cdot 10^{-11} \omega^4 + 7.2 \cdot 10^{-5} \omega^2 - 1 = 0 .$$

The dynamic parameters of the elastic link SpC for the given case of consideration are equal: $m_1 = 2.27$ N·c^2/mm; $h_1 = 13.96$ N·c/mm; $k_1 = 3.89 \cdot 10^4$ N/mm. The values of the TW node parameters are determined similarly: $m_2 = 8.65$ N·c^2/mm; $h_2 = 38.51$ N·c/mm; $k_2 = 0.93 \cdot 10^5$ N/mm.

3. Build a graph of frequency characteristics. To calculate the frequency characteristics: {$Re = f(\omega)$; $Im = f(\omega)$; $A = f(\omega)$; $\varphi = f(\omega)$} and the APFC of the elastic link, software was developed in the MatLab system [23–25]. When calculating the aforementioned characteristics, MatLab's possibilities were used to divide the graphic window into 4 sub windows (the "subplot" operator), add textual information to the figures (the "gtext" operator), and print the information in a qualitative form. The calculations were carried out in a wide range of frequency changes with different discreteness of the transfer function definition domain; moreover, the number of simultaneously processed APFC points is more than 8000.

The frequency characteristics of the elastic SpC link, including the dependences of the real *Re* and imaginary *Im* components of the transfer function are defined in the MatLab system. The amplitude phase-frequency characteristic (hodograph) of elastic link SpC shows in figure 3. To study the stability of the machine system, one- and two-parameter D-partitions are used.

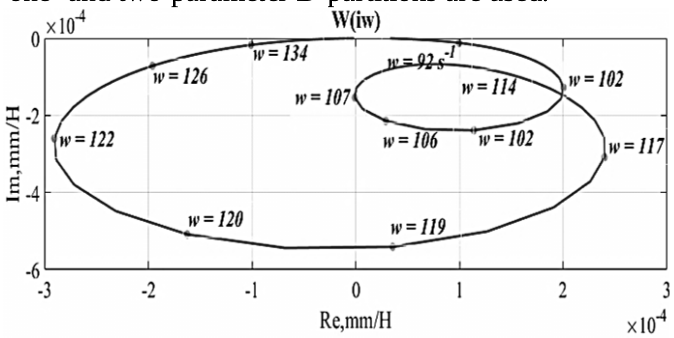

Figure 3. Amplitude phase-frequency characteristic (hodograph) of the SpC elastic link.

One-parameter D-partitioning. As the main characteristic equation, we use the characteristic polynomial of the denominator of the transfer function (1), presented in the following form (4):

$$K_p\left(A(p)+B(p)\right)+A(p)\cdot B(p)\cdot\left(T_p p+1\right)=0. \quad (4)$$

To determine the numerical values of the one-parameter D-partitioning parameters on the K_p plane, we use a transformation of the form (5):

$$K_p=-\frac{A(p)\cdot B(p)\cdot\left(T_p p+1\right)}{A(p)+B(p)}. \quad (5)$$

After the transition to the Fourier transform $p = i\omega$ and the selection of the real and imaginary parts of the expression (3), an algorithm was built and the program was written in MatLab [24, 26, 27]. To solve the problem (it is formulated as the determination of such values of K_p at which the system is stable) we construct the boundary of the D-partition in the plane of the complex parameter K_p, in a wide range of natural frequencies. In this case, only the partition of the real axis will be of interest, i.e. real K_p values.

From the expression (3) it follows that the boundary of the D-partition corresponds to the equation (6):

$$K_p=\frac{-h_1h_2T_pL_4\omega^3-L_5\omega^2+L_6L_1L_2L_3\omega}{L_3^2+L_4^2}-i\frac{-h_1h_2T_pL_3\omega^3-L_7\omega^2+L_8L_1L_2L_4\omega}{L_3^2+L_4^2}, \quad (6)$$

where

$L_1=k_1-m_1\omega^2;L_2=k_1-m_2\omega^2;L_3=(k_1+k_2)-(m_1+m_2)\omega^2;\ L_4=(h_1+h_2)\omega;L_5=L_1h_2L_3T_p+$
$+L_2h_1L_3T_p+h_1h_2h_3;$

$L_6=L_1L_2L_4T_p+L_1h_2L_4+h_1L_2L_4;L_7=L_1h_2L_4T_p+h_1h_2L_4+h_1L_2L_4;L_8=L_1L_2L_3T_p+L_1h_2L_3+h_1L_2L_3.$

Here $\{m_1, h_1, k_1\}$; $\{m_2, h_2, k_2\}$ – the parameters of the dynamics models for the SpC and TW elastic links respectively.

The boundary of the D-partition according to (4) is presented in figure 4

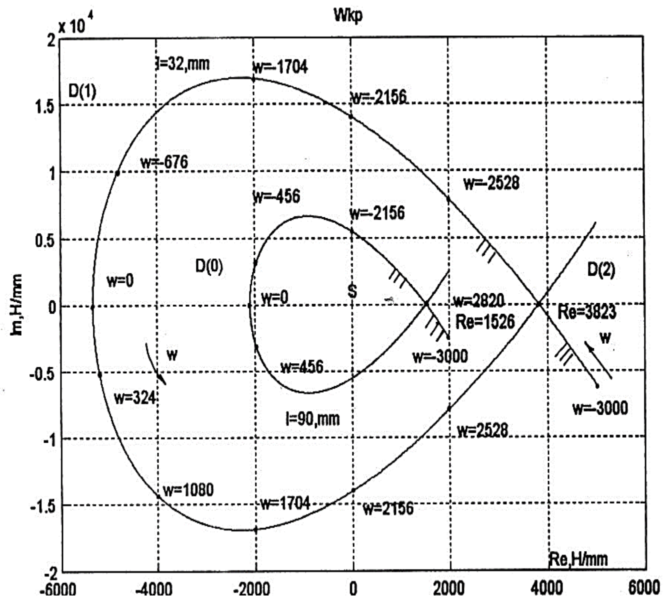

Figure 4. One-parameter D-partition on the K_p plane.

Two adjustments were experimentally investigated with a cantilever length l_k = 500 mm (the limit capabilities of the SF68VF4 machine along the programmable displacement along the Z axis) and l_k = 300 mm. With an increase in the length of the cantilever, the cutting stiffness decreases from K_p = 4466 N/mm (with f = 181.4 Hz) to K_p = 2008 N/mm (with f = 181.75 Hz).

The pretender for the stability area is the area S (figure 4), to which the hatching of the D-partition boundary is directed [28–30]. It can be shown that this area is not only a pretender, but also an area of sustainability. Indeed, the point (0, 0) in which K_p = 0, lying in the domain S, belongs to the stability area $D(0)$ and when K_p = 0, the characteristic equation (2) turns into the equation:

$$7.74\cdot 10^{-9}p^{5}+3.0\cdot 10^{-5}p^{4}+0.02p^{3}+33.38p^{2}+86.24p+8.64\cdot 10^{6}=0.$$

All five roots of this equation: {p_1 = –3.59; $p_{2,3}$ = –0.13 + 0.77 i; $p_{4,5}$ = –0.074 + 0.72·i} lie in the left plane of the roots.

Thus, the system is stable if the K_p real values vary within the limits defined by the segment: ω = 0...1142, c^{-1}.

Analysis of the stability region in the K_p parameter plane during milling with an end mill (d_f = 16 mm; L_k = 32 mm) shows that the area is limited to a maximum value K_p^{lim} = 3823 N/mm with f = 346 Hz. The corresponding limit value of the chip width is b_{lim} = 1.8 mm (case of processing carbon structural steel with a shrinkage factor ς = 2.2; σ_b = 750 MPa). With an increase in the length of the cantilever, the magnitude K_p sharply decreases – with L_k = 90 mm, the stability area is limited K_p = 1526 N/mm, the frequency ω = 2820 s^{-1} and the limiting width b_{lim} = 0.7 mm.

The stability analysis of the dynamic SpC – TW system allows designers to determine the optimal cutting depth t_{opt} when analyzing the cutting process. The limit value of stiffness K_p = 3823 N/mm will correspond to the optimal values of the cutting depth t_{opt}:

$$\mathrm{t}_{\mathrm{opt}}=\frac{K_p^{\mathrm{lim}}}{1.4\cdot\varsigma\cdot\sigma_b\cdot\sin 60^{0}}=1.1\ mm.$$

4. Conclusion

In the course of the research the following results were received:

1. A procedure for analysis the dynamic quality of an elastic machine system and determining its stability area based on the D-partitioning method has been proposed. To implement this procedure, the transfer function of the elastic machine system on the basis of the transfer functions for the individual elastic links, taking into account the rather complex interaction of the two Spc and TW systems in the working mode is built.
2. A mathematical model of a two-mass system of elastic links for studying the compliance of a spindle node, equipped with a cantilever instrumental block under arbitrary loading has been created. To do this, a procedure for constructing static and frequency forms of the spindle node is introduced, on the basis of which the natural frequency and the logarithmic decrement of oscillations are determined.
3. A procedure for analysis the stability of machine-forming units using the D-partitioning method has been developed. In the development process, the frequency characteristics and APFC of the spindle node in the mathematical system MatLab were constructed to analysis the dynamics of the machine elastic system. An equation that determines the boundary of a one-parameter D-partition in the plane of the complex parameter K_p in a wide range of natural frequencies is obtained. The optimal cutting parameters were determined according to the criteria of stability, which is especially important for the stable operation of high-speed machining centres of drilling-milling-boring types.

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