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METHOD OF DETERMINING DYNAMIC CHARACTERISTICS OF MACHINE WITH BAR SUPPORT SYSTEM

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Abstract
A new methodology has been proposed for theoretical and experimental evaluation of the dynamic characteristics of new configurations of machine tools with working bodies, karouta spatial articulated rod system. Software "ToolsApp" has been developed with a native interface in order to implement the solution of a mathematical model. The latter defines numerical values of the damping coefficient of the system, and particularly the change in the dynamic characteristics of the machine under conditions of cyclic vibration loads.

Keywords: new methodology, dynamic qualities, mathematical model, spatial articulated rod system.

INTRODUCTION

Nowadays, along with the complexity of details forms and designs, requirements for functionality and technological possibilities of machining equipment are increasing. Modern trends in the machine tool industry is aimed at the development of multifunctional procesing centers, equipped with several head movement drives, actuators and auxillary servers-manipulators. The machine tools based on the mechanisms of parallel structures (MPS) not yielded in functionality abovementioned procesing centers, but have less metal consuming due to structural features. The only MPS shortcaming is rigid defiecency comparing with the tradicional lathe. The rigidity of the machine is one of the main parameter that directly affects the accuracy of the processing as well as the life cycle and the overhoul life of the machine. At the preliminary stages of designing the MPS machines necessary to foresee their basic characteristics, including rigidness [3]. There are many licensed software and mathematical modelling methods based on the theoretical calculation the kinematic features of the machine, as well as the structural features of their constituents. However, this simulation is not able to reproduce the real indicators of the machine and manufacturing of the full scale machine for research requires large expenses.

EXPOSITION

The goal of the research was developing the methodology of the mathematical simulating that bases on the results of experimental tests. That can be performed by computer simulation as well as indicators either the real machine or the test bench [1, 2, 7]. The advantage of this approach is that the mathematical simulation uses data (for example the rigidity of the bar support system) in which the errors of the real machine is already taken into account. This approach increases the reliability of the results. The mathematical model itself can be simplified. For example, tridimensional mathematical model of a bar support system comprises six equations [4].

We are propose a combined method for the prediction of the dynamic parameters and characteristics of a lathe support system (fig. 1).

As the results of structural and kinematic analysis of the bar support system with determination of the number and type of kinematic pairs we use the lathe support system (patent no. 27808) for aprobation of the method [3, 4].

The dynamic systems mathematical simulation and dynamic analysis of the spring support system conducted at the following assumptions:

1. The lathe support is characterized by focused mass m and considered as a breaking system with 6-parallel links. All those links have determined parameteres of the stiffness and the dampness.

2. The masses of all the 6-bars can be neglected due to much smaller mass than the mass of the support.

3. The bars have the same working length, $L_i=A_iB_i$, where $i = 1,2 \ldots 6$.

4. The bars position in space is characterized by angles α_{xy}, α_{xz}, α_{yz} in the respective coordinate planes.

5. The bars are connected to the support and the bearings with spherical joints and apllied stretching/ compressing force only.

6. The bars are characterized by the stiffness coefficients Ci and the dampness coefficients hi, where $i=1,2 \ldots 6$, which can be resolved into orthogonal components.

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7. The cutting force acts on the cutting tool *P=P(t)* as a function of time. The components of this force on an orthogonal coordinate axes are P_x, P_y, and P_z, respectively.

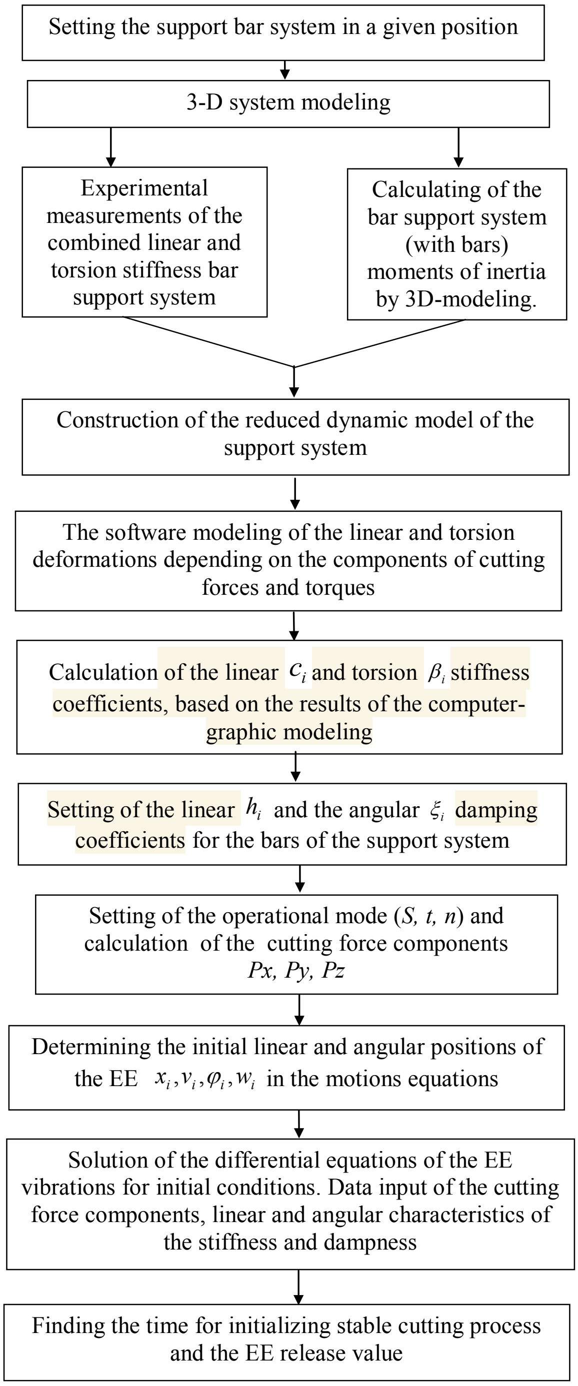


Fig. 1. The dynamic parameters determination sequence of the bar support system [5]

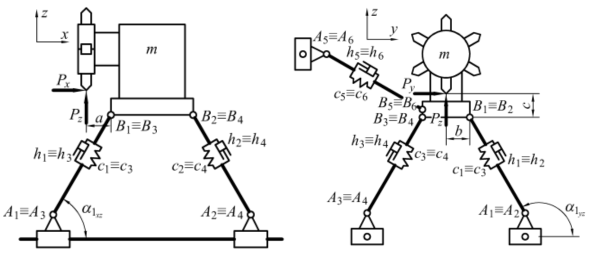
According to the assumptions, the existing dynamic system scheme is reduced to centered mass system with 6-degrees of freedom movement along orthogonal axes and rotations around them. The sequence of experimental and mathematical analysis of dynamic characteristics of the bar support system is proposed (fig. 2).

General solution (1) of a 2nd order linear homogeneous differential equations, provided that $\bar{h}^2 < 4\bar{c}$ has the appearance, ($\ddot{y} + \bar{h}\dot{y} + \bar{c}y = f(t)$ $\bar{h}^2 < 4\bar{c}$),

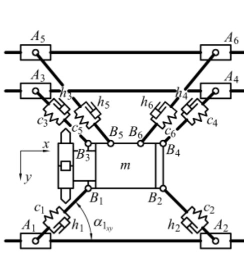
$$y_0(t) = C_1 Y_1(t) + C_2 Y_2(t) \quad , \qquad (2)$$

where

$$Y_1(t) = e^{-\frac{\bar{h}t}{2}} \sin(\omega t), Y_2(t) = e^{-\frac{\bar{h}t}{2}} \cos(\omega t), \omega = \frac{\sqrt{4\bar{c} - \bar{h}^2}}{2} .$$



$$\begin{cases} m \cdot \ddot{x} + h_x \cdot \dot{x} + c_x \cdot x = P_x(t) \\ m \cdot \ddot{y} + h_y \cdot \dot{y} + c_y \cdot y = P_y(t) \\ m \cdot \ddot{z} + h_z \cdot \dot{z} + c_z \cdot z = P_z(t) \end{cases} \qquad (1)$$



$$\begin{cases} \ddot{\varphi}_x \cdot J_x + \xi_x \cdot \dot{\varphi}_x + \beta_x \cdot \varphi_x = M_x(t) \\ \ddot{\varphi}_y \cdot J_y + \xi_y \cdot \dot{\varphi}_y + \beta_y \cdot \varphi_y = M_y(t) \\ \ddot{\varphi}_z \cdot J_z + \xi_z \cdot \dot{\varphi}_z + \beta_z \cdot \varphi_z = M_z(t) \end{cases}$$

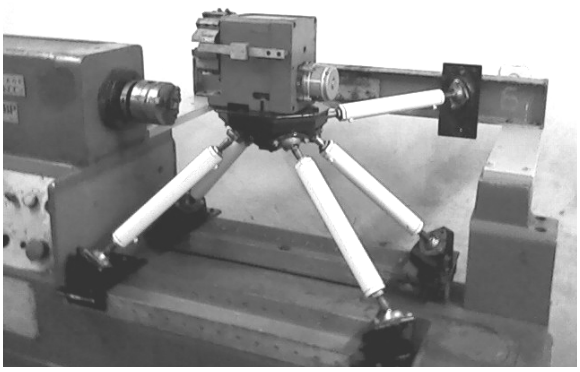


Fig. 2. The test bench for determining of the dynamic characteristics of the machine

According to the method of variation of constant values, instead of C_1 and C_2 are considered auxiliary functions C_1 (t) and C_2 (t), which is determined by the system of differential equations

$$\dot{C}_1(t)Y_1(t) + \dot{C}_2(t)Y_2(t) = 0,$$

$$\dot{C}_1(t)\dot{Y}_1(t) + \dot{C}_2(t)\dot{Y}_2(t) = f(t). \qquad (3)$$

$$\dot{C}_1(t) = -\frac{1}{\omega} f(t) e^{\frac{\bar{h}t}{2}} \cos(\omega t), \dot{C}_2(t) = -\frac{1}{\omega} f(t) e^{\frac{\bar{h}t}{2}} \sin(\omega t).$$

Fluctuations of the end-effector of bar support system corresponds cutting tool damped oscillations with a gradual release of the permanent displacement (relative feed movement) that is equal system elastic deformation $P_i / \bar{c}_i$

$$x_i(t) = Z_1 e^{-\frac{\bar{h}_i t}{2}} \sin(\omega_i t) + Z_2 e^{-\frac{\bar{h}_i t}{2}} \cos(\omega t) + \\ + \frac{e^{-\frac{\bar{h}_i t}{2}}}{\omega_i} \begin{pmatrix} \sin(\omega_i t) \int P_i(t) e^{\frac{\bar{h}_i t}{2}} \cos(\omega_i t) dt - \\ - \cos(\omega_i t) \int P_i(t) e^{\frac{\bar{h}_i t}{2}} \sin(\omega_i t) dt \end{pmatrix}, i = 1,2,3. \qquad (4)$$

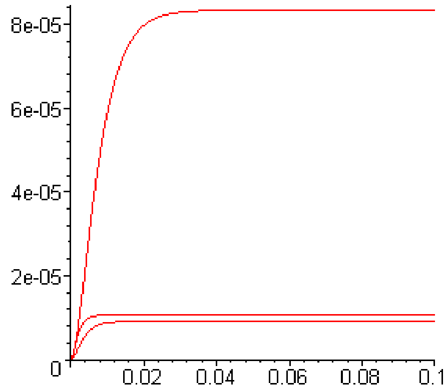
According to the research results of the stiffness and compliance characteristics conducted on the test bench lathe (fig. 2), and based on a priori assumption that the spherical joints, used in the machine, have high dumping characteristic and using mathematical models we find the magnitude of the damping linear and torcion coefficients of the components of the bar support system - the bars with the joints:

$\bar{h}^2 \approx 4\bar{c}$ and $\bar{\xi}^2 \approx 4\bar{\beta}$, then $\bar{h}_1 \approx 2\sqrt{\bar{c}_1} = 633$ s^{-1}; $\bar{h}_2 = 1450$ s^{-1}; $\bar{h}_3 = 2367$ s^{-1}; $\bar{\xi}_1 = 2\sqrt{\bar{\beta}_1} = 307$ (sec·rad·m)$^{-1}$; $\bar{\xi}_2 = 370$ (sec·rad·m)$^{-1}$; $\bar{\xi}_3 = 289$ (sec·rad·m)$^{-1}$.

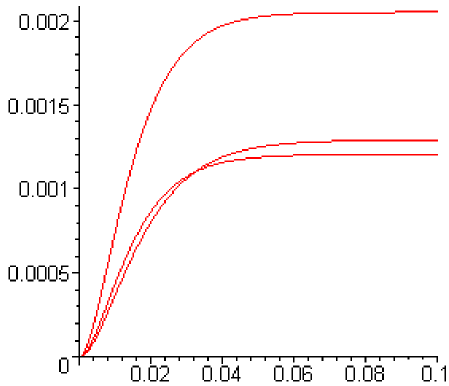
The stiffness has value: c_x=0,04·10^8 N/m; c_y=0,21·10^8 N/m; c_z=0,56·10^8 N/m; β_x=37,78·10^3 N·m/rad; β_y=41,63·10^3 N·m/rad; β_z=31,43·10^3 N·m/rad.

The moments of inertia are determined by 3D modeling licensed software "SolidWorks" They are: J x 1, 61 kg m^2; J_y 1.22 kg m^2; J_z 1,51 kg m^2; mass *m* 40 kg.

The mathematical simulation is determined the release value of the end-effector and the tool during the processing. Substituting the incoming experimental data of the system static stiffness in the static position we recieved the system loads. The loads shown in the following support system compliance (release) graphic. (Fig. 3).



a)



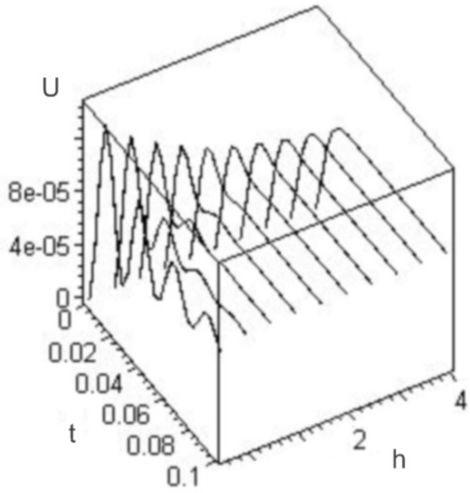
b)

Fig. 3. The value of the end-effector realising and the time to start stable cuting process a: -the linear movement of the end-effector mass center (linear realise); b -the angular movement of the end-effector in the cutting process

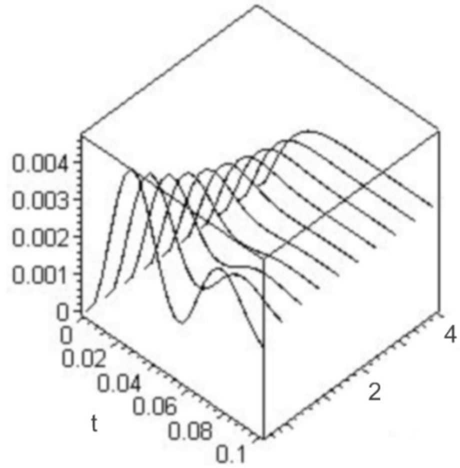
Analyse the oscillation amplitudes of the joints with the differenent stiffness that arise during the processing, (fig.4)

Mathematical model of the support dynamic system has the solution using the "Maple" software, in which the particular code that corresponds to the model, also created a software module "ToolsApp" with an interface (fig. 5) [8].

The program performs the calculations of the components values of the cutting forces of the processing technological modes, as well as the support linear movements by ortogonal axises and their dynamic characteristics. The program calculates the stabilization system time with the variable load. Also it drows spectrograms of the dynamic system parametres.



a)



b)

Fig.4. The time dependence of the system vibration amplitude depending on damping coefficient: a) - linear vibrations; b) - torsional vibrations

For testing the interface of the programs that based on mathematical model of bar support dynamic system, do the calculations with the same input data as was using in the mathematical modeling (testing simulation by "Maple" in manual mode).

It is define the stabilization system time and the total displacement of the center of mass of the bar support system by the load that means the start of the cutting process (fig. 5).

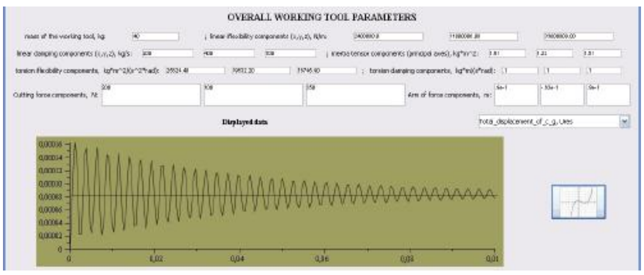


Fig.5. Determination of the stabilization time and the bar support system motions

Also the program capable of research the dynamic errors of the processing non- cylindrical details where periodic shock appears, i.e. details with juts, grooves, and polygonal details. For this the interface provides window where you can specify the cutting force components by the coordinate axes as the numerical values, as well as the analytical dependance.

The dynamic processing scheme those parts looks like (Fig. 6) and characterized by the number impact parameters of the pulse shock load *N*, the period *t* and the shock impact time shock impact τ.

In order to simplify the process of the dependency calculation for complex profile parts offered the separate formula module generator (Fig. 6).

```
n:= 1000;   # Axis speed, rpm
i_c:=3; #   Impact value +1
N_beat:=3;  # impact value for 1 rotation
T_beat:=0.6;  #action time at 1
P:=100;   #cutting force (during processing)
P:=0;     # cutting force (after processing)
I   0:=0;   # start from (either start of the cutting or start of the
motion)
```

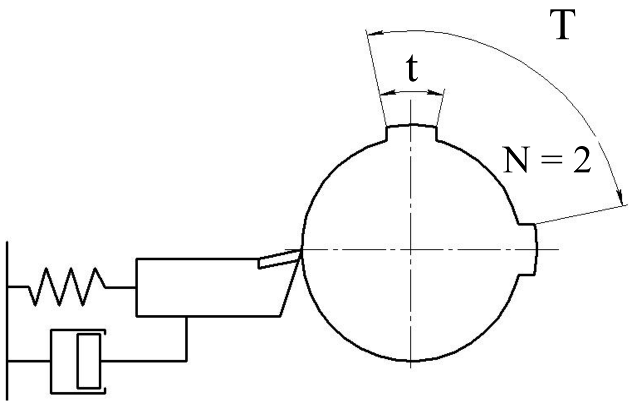
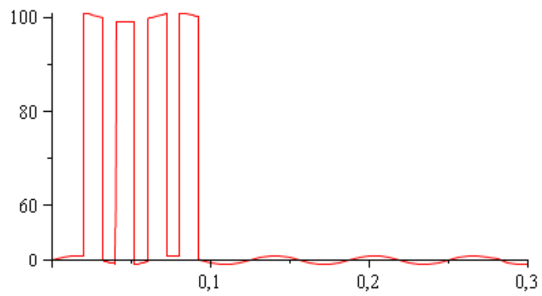


Fig. 6. Dynamic diagram of the complex surfaces processing

$$q := \begin{cases} 100 & 0.02000000000 \le t \textbf{ and } t < 0.03200000000 \\ 0 & 0.03200000000 \le t \textbf{ and } t < 0.04000000000 \\ 100 & 0.04000000000 \le t \textbf{ and } t < 0.05200000000 \\ 0 & 0.05200000000 \le t \textbf{ and } t < 0.06000000000 \\ 100 & 0.06000000000 \le t \textbf{ and } t < 0.07200000000 \\ 0 & 0.07200000000 \le t \textbf{ and } t < 0.08000000000 \\ 100 & 0.08000000000 \le t \textbf{ and } t < 0.09200000000 \\ 0 & 0.09200000000 \le t \textbf{ and } t < 0.1000000000 \end{cases} + \sin(100\,t)$$

a)



b)

Fig.*7. The formulas generator output:*
a) - the cutting efforts change ranges; b)- the diagram of the cutting efforts changes by time

The result of the module calculation is the system of the cutting force change ranges (fig. 7, a) and the diagram its changes by time (fig. 7, b). The program incorporated not only the ability to generate the different modes of detail processing, but also taken into account forced oscillatory processes from the effects of external forces. Those forces superimposed on the active pulsed load bar support system (fig. 7, b).

The processing of parts with deviations from cylindric form is characterized by cyclical shock load. So it is important to be able to simulate the influence of technological modes on quality of the parts surface, i.e foresay interaction load pulse frequency (depending on angular velocity and the number of stimulus pulses on the part surface), its duration and the system forced periodic oscillations. Next, consider the two cases of the action of changing force by four pulse in one rotation.

Results of the formula generator (an equations system) substituted as input data in the cutting force window in the "ToolsApp" application. The output is the existing force value by time (Fig. 8, a) and spectral diagram that determines the system stabilization time (the stable oscillation of the stable cutting process) (Figure 8, b).

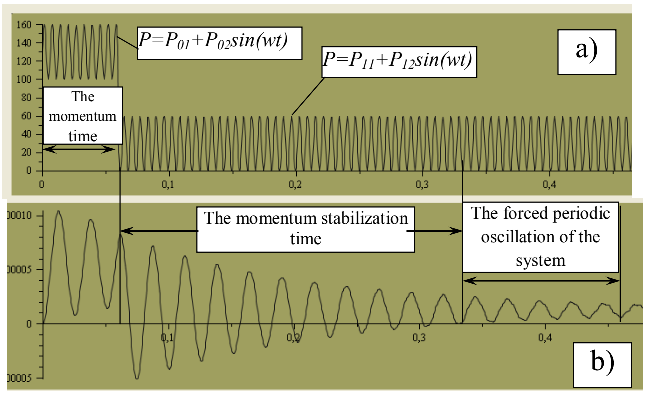
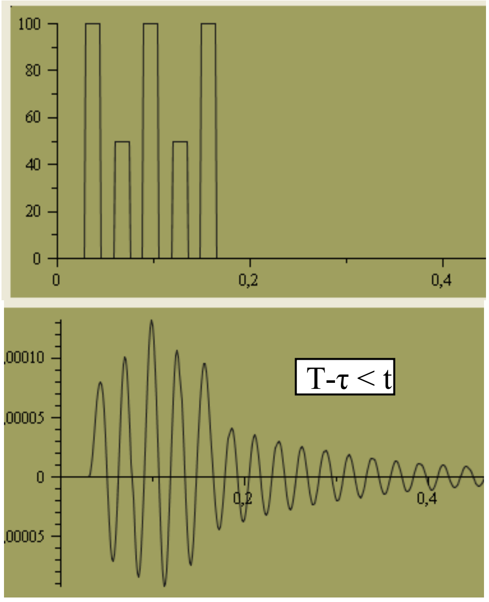


Fig. 8. The simulation results:
a) - the actual force diagram; b) - the oscillatory process diagram.

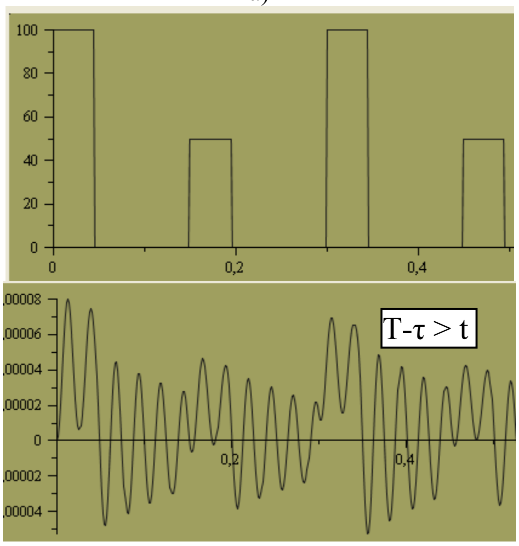
From experemental point of view "ToolsApp" provides the opportunity to select define coefficients of stiffness and damping of the technological systems by trial-and-error method. In this case, test bench lathe have tested by using single-channel vibration analyzer model 795M (Fig. 10). The research of the system stabilization time (as the main indicator of stiffness and dissipative properties of the lathe) conducted by vibration acceleration. [2].

As the experimental research was carried out the processing of the variable load and the transitional process at the beginning (the tool penetration) and the end of the cutting (the tool output) Use this approach allowed to explore dynamic process in the real conditions (excluding the variable load devices) and to measure and to evaluate several settings sumaltaniously in one processing. (it is not possible by using single-channel vybration analyzer).

The pulse impact load processing is simulating in the stage of the tool output. The detail material – wood. The detail form - disks, that gives the striking impact on the system during the 'breakdown' of remaining allowance. From the spectrograms (fig. 11 a) the stabilization system time after shock load 0.15 seconds.



a)

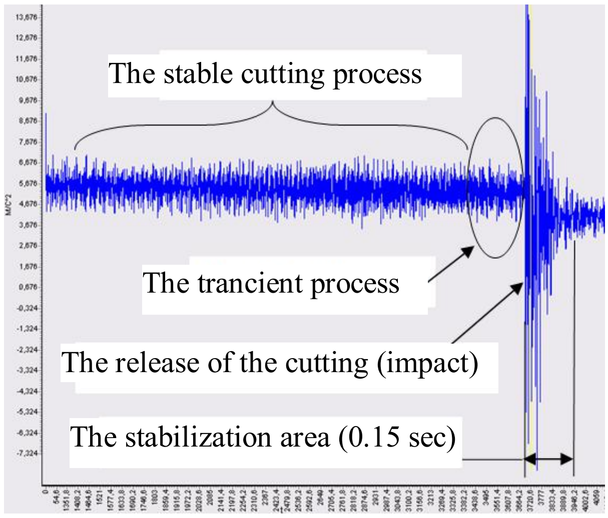


b)

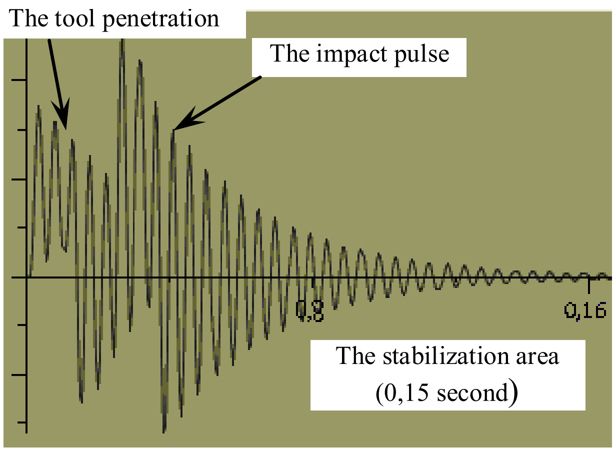
Fig. 9. Simulation results of processing at the different frequencies of the rotating detail:
a) - speed 1000 rpm; b) - speed 550 rpm

Fig.10. The vibration analyzer model 795M

Comparing the experimentally obtained result and simulation results, by selecting damping coefficients *set* 4900 kg/s, 10400 kg/c, 18100 kg/s respectively for the each cutting force ortogonal components *Px, Py, Pz*, according to the classical scheme of the cutting theory.



a)



b)

Fig.11. The vibration acceleration spectral diagram: a) – the research result; b) – the simulation result

Taking into account the peculiarities of the mutual location of the bars in the machine workspace (the presence of the two bars, which are located in the horizontal plane), the rigidity of the spatial systems are not the same in all directions (in the areas of the cutting force components). This might affect the inhomogeneity of the EE release (which is also confirmed by the processing difference). To use the maximum of the stiffness properties is need the study to determine the relative position of parts and tools in the working area of the machine (the cutting forces distribution).

Substituting the force value from the simulation, according to the system stiffness by to the coordinate axes obtain the diagram the end-effector release with the same power value by the abovementioned cases of the tool-detail relative position by three time intervals for stable cutting process (Fig. 12).

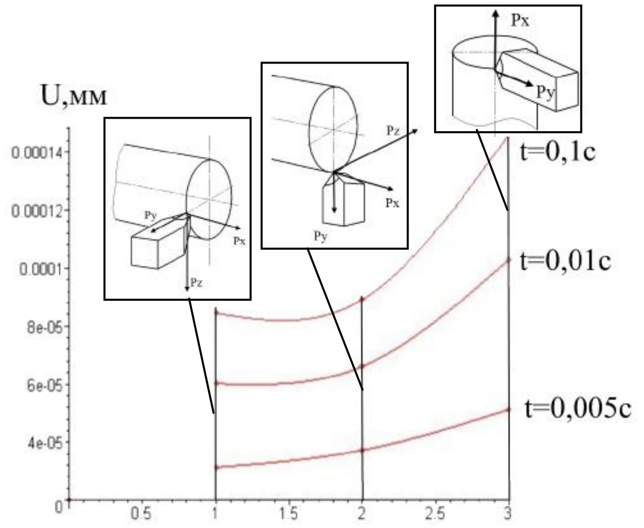


Fig. 12. The dependence of the EE release by the relative position of details and tools in the mashine working space

CONCLUSION

The research method is developed and the teoretical and experimental study of elastic-stress state of the bar support system conducted at different positions of the EE to determine the effects of uneven distribution of stiffness in the workspace.

Designed and produced the test-banch of the lathe to conduct experimental research and theoretical verification.

The bar support system dynamic mathematical model was approved. By the results of that approbation, under the condition of the absorption of the cutting process vibration, the time needed for the support system stabilization and the vibration dissipation (depending on the system damping characteristics) within 0.02-0.08 sec. and the value of the end-effector release - 50-80 microns, depending on the position of EE.

The software "ToolsApp" with its own interface implemented for mathematical simulation was developed. This software calculated the bar support system damping coefficients and the nature of the machine dynamic characteristics under conditions of the cyclic vibration loads.

The comparative analysis of the detail-tool relative position in the lathe working space accomplished. It is established that due to the stiffness of the support spatial frame is not the same in all action directions, the difference in EE release can be 1.3 times depending on the scheme.

REFERENCE

[1] Vnukov, J. N., Kuchugurov M. V., Dyadya S. I., Zinchenko R. N., Gonchar, E. A., Method and device for research regenerative self-oscillation when turning (http://www.zntu.edu.ua/publikaciyi-kafedri-3)

[2] Kravchenko V. M., Sidorov V. A., Sedush V. Y. Technical diagnosis of mechanical equipment. Tutorial. Donetsk 2009. - 498p.

[3] Kuznetsov Y. M., Dmitriev D. A., G. Y. Dinevich Layout of machine tools with parallel mechanisms. / The monograph under the editorship of Yu. M. Kuznetsov. – Kherson: PP Vyshemirskiy, V. S., 2009. – 456 p.

[4] Y. Kuznetsov. Lathe support – hexaglide: rigidity and dynamics. /Y. N. Kuznetsov, D. A. Dmytriev, M. I. Podolsky.// Proceedings of international scientific conference. Gabrovo, 2012, volume 2, P. 69-74.

[5] M. I. Podolsky . Mathematical modeling of the dynamics of the system support lathe type "hexaglide". / M. I. Podolsky, S. A. Rusanov, D. A. Dmytriev// Bulletin of SevNTU: collection / №.139/2013 - Sevastopol, 2013. - P. 176-182.

[6] Strutinsky V. B. Dynamic processes in metal-cutting machines: monograph/ V. B. Strutinsky, V. M. Drozdenko// – Kyiv: Osnova-Print, 2010. – 440 p.

[7] Strutinsky. C. V. Experimental determination of dynamic characteristics and building on their basis of mathematical model of the spatial system drives / C. V. Strutinsky, A. Gurzhiy.// Scientific journal "Technological complexes" No. 1(7), 2013.

[8] Certificate No. 57913 from 29.12.2014 "Computer program "ToolsApp" (Ukraine).