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MECHATRONIC APPROACH TO THE MACHINE TOOL DESIGN

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Abstract: *In the paper we consider the possibilities of complex approach to the design of machine tools in the light of the newest means which the designer can use on the earlier stages of the design procedure. Same of components is delivered with the controllers in the form of the microprocessors. In that case it is useful to change nominal software to adjust the control law to our purposes. So more we can add additional control loops to improve dynamics of the machine*

1. INTRODUCTION

Machine tools seem to be a very good example for discussing the influence of modern electronics, sensors and sophisticated drives on "mechanical" design. As a matter of fact **mechatronic**, as a symbiosis of mechanics, electronics, informatics..., was born in the field of machine tools. It happened about 50 years ago when computers were introduced to the description of spatial shapes and to the automatic control of machine tools. CAD systems were invented and introduced to industry because of the numerical control, NC, of machine tools.

Looking from another side, numerical control revolutionized the machine tools themselves. NC machine tools are quite different than no-NC machine tools used to be. Traditional machine tools were designed by their producers in practically every detail as a new, original construction, made from new subassemblies. Modern NC machine tools are made of modules which in many cases are produced by specialized enterprises. Even many elements used in the design of machine tools (rolling guideways, ball screws...) are delivered by specialized producers and selected from catalogues.

Design stage time of machine tools can be reduced by using of mechatronic approach to their conceptual analysis and preliminary synthesis. During synthesis and analysis we should consider the complete machine taking into account mechanical structure, drivers, control systems and cutting process. The development of faster and faster machine tools, to reduce machining time and assure the required precision, needs stiff but light mechanical structures and drives with wide bandwidth [1]. It causes that all elements of tool machine form one dynamical system influenced by control systems.

In modern machine tools each drive is a servomechanism with control loop containing: sensors, digital controllers, and actuators. To improve machine precision we are also forced to consider the possibilities of the introduction of vibration and movement control system in the form of feedforward or/and feedback systems [2]. It means that dynamics of mechanical structure is strongly coupled with dynamics of local or global control systems particularly in the range of higher frequencies.

Different criteria are taking into account during designing of machine tools. For example the geometric accuracy of the machning elements is one of the important features of modern tools. The accuracy depends on the compliance of machine and on the cutting forces. Cutting forces can be reduced by the introduction of High Speed Cutting process. There is no simple way to reduce compliance. Of course, we can rise the rigidity of the machine by increase of its mass and dimensions but this is in opposition to the economy and need for high acceleration. Typical way to decrease the compliance is the structural or parametric modification of the machine structure. Such passive approach may be sufficient in the case of slow machne tools were the static stiffness is the most important.

In the case of faster machines beside static stiffness the big role plays the dynamic stiffness of the system: machine-tool-machining element. Presently, we are able to influence machine stiffness in the broad range of frequency by the active means. Moving parts cause that static and dynamic stiffness changes during machine operation. Therefore, to shape the dynamic characteristics the adaptive control methods are often used. Lately, more advanced control methods are applied to the mechanical systems. They are called robust methods since such system is robust to the parameter changes and to external excitations. Robust controller has constant parameters and assures sufficient stability and performance of the closed-loop system in the broad range of parameters of the operation points.

In the paper we consider the possibilities of complex approach to the design of machine tools in the light of the newest means which the designer can use on the earlier stages of the design procedure. Same of components is delivered with the controllers in the form of the microprocessors. In that case it is useful to change nominal software to adjust the control law to our purposes. So more we can add additional control loops to improve dynamics of the machine. Such problems will be considered in our paper.

2. CONFIGURATION OF MACHINE TOOLS

A designer of a new machine tool has much more freedom than a designer of a new car. In the car design there are just a few configurations of the main modules and a designer has a limited choice at the stage of a conceptual design. In the field of machine tools there are many possible configurations of modules which can fulfill: the required parameters of workpieces to be machined, NC movements and additional functions.

The freedom of choice requires responsibility of a machine tool designer. It is very important for the final result what configuration of the machine tool has been chosen at the conceptual stage of design. The know-how of the design team and a heuristic approach may be not enough. A certain method of multi body simulation, BMS, is propose as a tool for the configuration choice based of estimation of the working stiffness of different mechanical structures [3].

In modern manufacturing **monitoring** is more and more popular, frequently as a part of automatic supervision [15]. Automatic supervision attempts to eliminate the influence of disturbances, to guard machine tool or/and workpiece and to optimize production. The term monitoring was adopted from medicine where it is as a kind of automatic, simplified diagnostics. Diagnostics, which is directed only at a chosen features of monitored object, but is active all the time. In hospitals the special sensors are mounted on seriously ill patients and the signals from the sensors are continuously and automatically analyzed from the point of view of selected features (e.g. pulse, blood pressure...). When the value of selected feature is identified as critical the hospital personnel is automatically called to intervene.

Mechatronic modules with their own drive and sensor units are connected by information signals with other mechatronic units and with the control system of the machine tool. Such structure is much more flexible and easier for implementation of "intelligence". At the same time their stiffness may be different than in the case of pure mechanical connections.

An important possibility of "mechatronic machine tools" is reconfiguration. Modules bought from the catalogued items and used in modern machine tools must have unified (and frequently standardized) interfaces. It makes possible changing configuration of the machine tool after some time of its use. The reconfiguration may change the machining parameters and allow the user of the machine tool to be much more flexible and respond to the changing demands of the market quicker and with smaller expenses.

But the best way to reconfigure system is the introduction of proper software to the controllers. It allows the low cost modernization of the machine tools. If we add the control systems of structure parameters we can design the **smart** machine tools.

3. MECHANICAL STRUCTURE MODELING

Classic motion equations of flexible or mass-lumped structure with a finite number of degrees of freedom are as follows:

$$\mathbf{M}\ddot{\mathbf{x}}+\mathbf{C}\dot{\mathbf{x}}+\mathbf{k}\mathbf{x}=\mathbf{f} \quad (1)$$

where $\boldsymbol{x}$ and $\boldsymbol{f}$ are the vectors of generalized displacements (translations and displacements) and forces (point forces and torques) and $\boldsymbol{M}$, $\boldsymbol{K}$ and $\boldsymbol{C}$ are respectively the mass, stiffness, and damping matrices, they are symmetric and semi positive definite. $\boldsymbol{M}$ and $\boldsymbol{K}$ arise from the discretization of the structure, usually with finite elements. A lumped mass system has diagonal mass matrix while the finite element method usually leads to non-diagonal mass matrix. In the complex machine tools the lamped mass system also have non-diagonal mass matrix [3].

The damping matrix $\boldsymbol{C}$ represents the various dissipation mechanisms in the structure, which are usually poorly known. Therefore it is customary to assume hypothesis about Rayleigh damping: $\mathbf{C}=\alpha\mathbf{M}+\beta\mathbf{K}$, where coefficients α, β are selected to fit the structure under consideration.

Let us introduce modal transformation: $\mathbf{x}=\mathbf{\Phi}\mathbf{z}$, where z is the vector of modal coordinates. The transformation leads to decoupled modal equations:

$$\ddot{\mathbf{z}}+2\boldsymbol{\xi}\mathbf{\Omega}\dot{\mathbf{z}}+\mathbf{\Omega}^2=\boldsymbol{\mu}^{-1}\mathbf{\Phi}^T\mathbf{f} \quad (2)$$

where: $\mathbf{\Omega}=diag(\omega_i)$ is the matrix of natural frequencies, $\boldsymbol{\mu}=diag(\mu_i)$ is the matrix of modal masses, and $\boldsymbol{\xi}=diag(\xi_i=\frac{1}{2}[\frac{\alpha}{\omega_i}+\beta\omega_i])$ is the matrix of model damping.

The transfer function between the force ***f*** as an input and ***z*** as an output is in the following form:

$$G(\omega)=\left[-\omega^2\mathbf{M}+j\omega\mathbf{C}+\mathbf{K}\right]^{-1}=\sum_{i=1}^{n}\frac{\phi_i\phi_i^T}{\mu_i\left(\omega_i^2-\omega^2+2j\xi_i\omega_i\omega\right)} \qquad (3)$$

Typical Bode characteristics of the transfer function (3) are shown in the Fig.1.

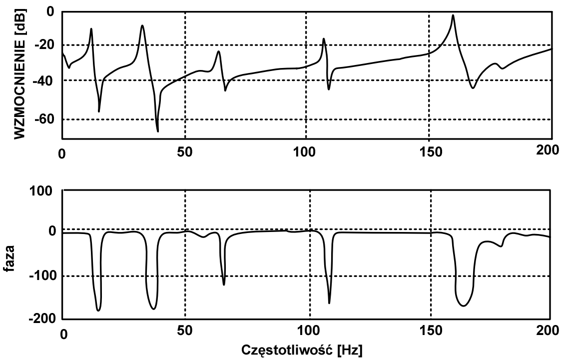

Fig.1. Typical frequency characteristics of a mechanical structure.

For control purposes we usually reduce the model to *m* lowest modes:

$$G(\omega)=\sum_{i=0}^{m}\frac{\phi_i^2(k)}{\mu_i\left(\omega_i^2-\omega^2\right)}+R \qquad (4)$$

and residual modes are:

$$R=\sum_{m+1}^{n}\frac{\phi_i\phi_i^T}{\mu_i\omega_i} \qquad (5)$$

Residual modes in the form (5) cause that transfer function (4) is non-proper function. [2]. It leads to big differences between zeros of transfer functions (3) and (4).

For ω=0 we obtain from (3) the modal expansion of the static flexibility matrix:

$$G(0)=K^{-1}=\sum_{i=1}^{n}\frac{\phi_i\phi_i^T}{\mu_i\omega_i}=\Phi(\Phi^T K\Phi)^{-1}\Phi^T \qquad (6)$$

One of the aims of vibration control system of machine tools is to make the static flexible matrix elements as small as possible.

4. DRIVES

Electric and hydraulic drives are used in machine tools. Usully we assume that drives work perfectly. It means that they have amplifications equal one (0 [dB]) in the sufficient bandwith.. It is true in the case of slowly working machine tools. In this case we can omit the dynamics of the drives. We should take into account the dynamcs of servo-drives when the machines work with acceleration reaching 3[g]. The bandwidth of servo-drives depends on many causes: loads, operation point, control law, external disturbances, and so on.

As we show later the bandwidth can be wider in the case of proper choice of control law. A robust control system seems to be a perfect choice in the case of loaded machines. To design the robust controller we should consider all interactions of the control plant with surroundings and take into acount all spectrum of nonlinear plant parameters (which are conected for example with operation points) when we are designing of linear model of the control plant [4].

The interactions and changes of parameters are represented by weigth functions in th form of transfer functions as it is shown in the Fig. 2. The weight functions are chosen in such a way to create the systems parameters which are able to fulfill the assumptions. So, the weight disturbances $W_o(s)$ i $W_i(s)$ describe the contents of the disturbances d_i i d_o or they may be used to model disturbance power spectrum which depends on the nature of signals involved in the practical systems. The weight $W_n(s)$ describes the frequency model of the sensor noise. From the opposite side, the weight $W_e(s)$ reflects the requirements put on the shape of the closed-loop transfer functions, i.e. on the shape of the output sensitivity function. Similarly, the weight $W_u(s)$ describes some restrictions put on the control or actuator signals. The last weight $W_r(s)$ is an optional element used to achieve desired shape of command or to represent a nonunity feedback system in an equivalent unity feedback form [5,6].

In summary, the main step in controller design procedure is to choose the appropriate weight functions and to implement them in practical applications.

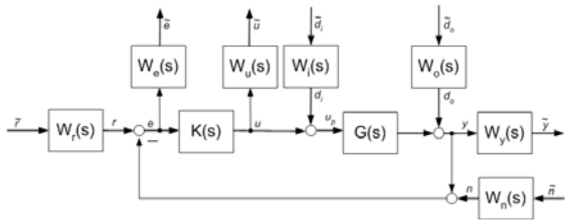

Fig.2. Functional diagram of system for robust analysis purposes.

The way of choice of the weights is shown in Fig. 3 and is described by equations (7). The control error weight $W_e(s)$ is associated with the sensitivity function $S(s)$. The sensitivity function |S| should have the low value for the low frequencies. In this range the control signal and disturbance signal play a meaningful role. So, the sensitivity function must fulfill the requirement $|W_e(s)\cdot S(s)| \leq 1$, where the weight $W_e(s)$ is described by equation (7). The form of the weight is realized in systems with steady-state error. In considered systems we assume the error lower than ε. This is ensured when $|S(0)| \leq \varepsilon$. So, for $|W_e(0)|\geq 1$ the norm is $||W_e(s)S(s)||_\infty \leq 1$. The similar analyses can be used for other weights and functions of the system.

$$W_e(s)=\left(\frac{\frac{1}{\sqrt[k]{M_s}}s+\omega_b}{s+\omega_b\sqrt[k]{\varepsilon}}\right)^k \quad W_u(s)=\left(\frac{s+\frac{\omega_{bc}}{\sqrt[k]{M_u}}}{\sqrt[k]{\varepsilon_1}s+\omega_{bc}}\right)^k \quad (7)$$

where: $k \geq 1$, ω_b – the cut-off frequency for function $S(j\omega)$, ε – the lower restriction of $S(j\omega)$, M_s – the upper restriction of $S(j\omega)$, ω_{bc} – the cut-off frequency for function $R(j\omega)$, ε_1 – the lower restriction of $R(j\omega)$, M_u – the upper restriction of $R(j\omega)$.

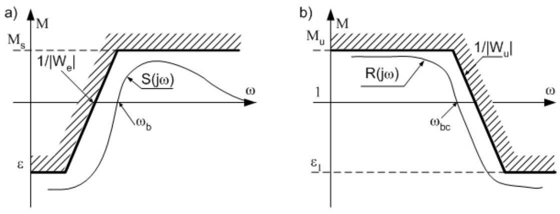

Fig.3. Limitations put on the weights functions.

The nominal model of the plant together with weight functions forms an extended model of the open-loop system. For such model we can obtain the optimal controllers using of commercial software, for example, Matlab-Simulink.

4.1. Electric drive

Modern control theory has recently been widely applied to design the controllers for electric drives [7,8]. The reasons for this are following: 1) the necessity of meeting increasingly stringent requirements on the performance of drive control systems, 2) easy access to modern power semiconductor switching components and microprocessors with which very sophisticated control strategies can be implemented at reasonable cost, 3) modern control theory has been extended and modified so that it is practically applicable to microprocessor-based drive control by taking into account physical constraints such as input delay times and input and state variable constraints.

The object of consideration is DC motor D – 101. At first, analytical model of the open-loop system is derived. Next, the classical PID controller is designed to obtain a reference system. After some theoretical consideration the robust controller was designed. Performance of the systems with both controllers is investigated by computer simulation and in laboratory experiments.

By designing PID controller and optimising its parameters we obtain the template which will be a reference for later simulations and experiments with robust control system. The parameters of PID controller were optimised in Matlab by using of NCD Blockset Toolbox [9]. As a result we obtain the following PID controller parameters: Kp=1 – proportional gain; Ki=0.25 – gain of integration part; Kd=0.01 – gain of differential part.

We assume in our case that the following parameters change during the electric drive operation: k_E - voltage constant of drive; T_m - mechanical time constant; T_e - electrical time constant; in the range ±10% of nominal value. For example, electrical time constant can have values from the set $\{0.252,\ 0.28,\ 0.308\}$. Transfer function of closed-loop system with any nonnominal parameters will be denoted by $G_P(s)$.

Let us introduce relative error in the following form: $e(j\omega)=\frac{G_P(j\omega)-G(j\omega)}{G(j\omega)}$. The choice of the weight is determined by condition that for all frequencies the Bode diagram of weight should be above the plot of relative error. To do this we chose the weight transfer function in the form: $\omega_u(s)=\frac{1.5\cdot 10^{-3}\cdot s}{1.5\cdot 10^{-7}\cdot s+1}$

The next considered weight is connected with the scaling of the controller output signal *u*. For simplicity we assume a scalar weight where its magnitude is as follows: $\omega_I = 7$.

In our case after calculations and simplifications we have obtained the controller described by the following transfer function:

$$K(s)=\frac{49.86\cdot s^2+2600.11\cdot s+9096.58}{s^3+773.35\cdot s^2+4141.89\cdot s+4.14}.$$

The results of computer simulations of closed-loop system for both controllers are shown in the Fig. 4.

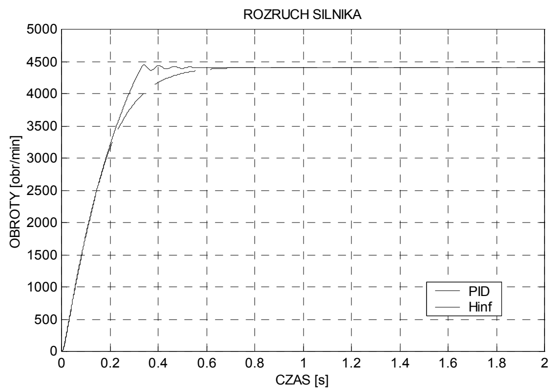

Fig. 4. Time response of closed-loop system with PID and H_∞ *to step change in the reference angular speed (n=0-4400 [rpm]).*

We wanted to confirm the simulation results by experimental investigation of electrical drive D-101. Therefore, the program of lab researches is similar to the program of computer simulations. For both controllers the start phase during which electric drive reaches its operation speed was recorded on the lab stand and transient processes are shown in Fig.5. From these plots we have calculated the performance parameters presented in Table 1.

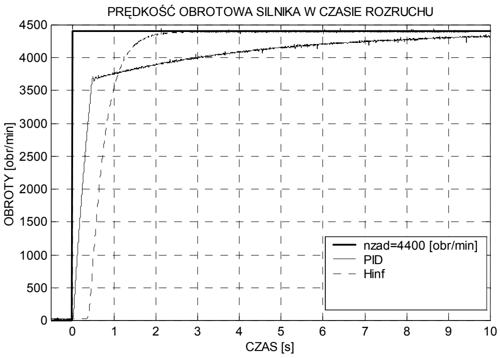

Fig. 5. Start up phase of the electric drive for both controllers recorded in the lab stand (reference angular speed equals 4400 [rpm])

Tab. 1. Performance parameters of experimental control systems.

	Parameters		
Controller	W [J] Control energy	T_n [s] Rise time	t_r [s] Control time
PID	165.07	0.35	5.21
H_∞	168.08	0.79	1.46

On the base of performance parameters collected in the Table 1 we can notice that the system with PID controller has faster response, but it takes longer time to reach the zero offset. It results from too small gain of integration part of controller. The PID controller was designed for linearized model of DC motor while the real motor has a little other dynamic properties. We can see that the robust control system with H_∞ controller is much more insensitive to the plant uncertainty. The systems with both controllers consumed similar amount of electric energy during start-up phase.

We also watched how drive reacts to step changes of reference angular speed. Behaviour of the closed-loop system is presented in Fig.6. There is no big differences between systems with both controllers. So more, there are zero steady-state offsets and smaller overshoots than ones which appeared during computer simulations.

4.2. Electro-hydraulic servo-drive

The frequency characteristics of the system for nonlinear and linear model of servo-drive are presented in the Figure 2 for the following parameters: supply pressures $P_{10\,0} = P_{20\,0} = 21$ [MPa], main-piston and servo-valve displacements $\Delta X_V = \Delta X = 0$, and for three flow rates.

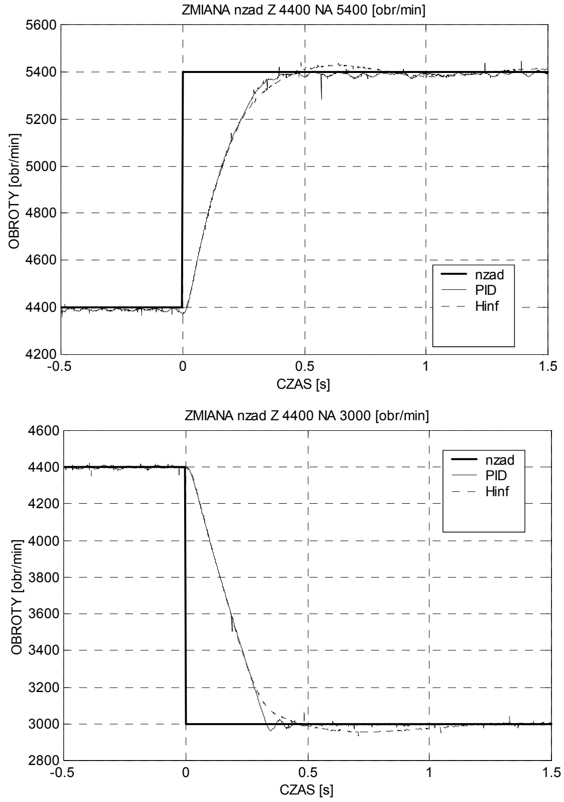

Fig. 6. Recorded control system responses to step changes in the reference angular speed:
a) decreasing from 4400 to 3000 [rpm],
b) increasing from 4400 to 3000 [rpm].

All characteristics of linear model are similar in given the frequency range. In the case of nonlinear model we can see big changes of the slope of characteristics. The biggest differences between models are in the vicinity of frequency 7.5 [Hz].

For the flow rate Q_{3L} the system bandwidths in the case of robust controllers are wider, than for PID controller. The increase of the bandwidths reaches 30 %. System with optimal robust controller has the best cut-off parameter and it is 2,65 [Hz]. The results for gain parameters are similar to the preface case.

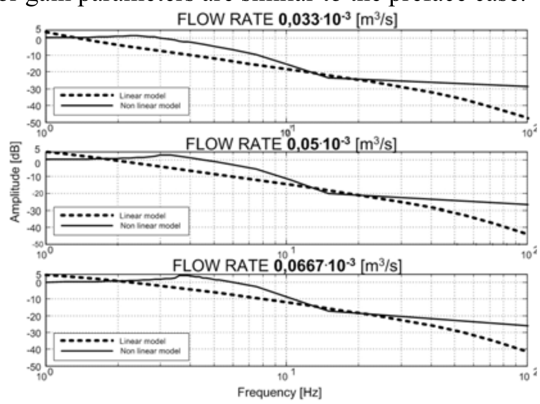

Fig. 7. Amplitude-frequency characteristics of the linear and nonlinear model of servo-drive.

The most interesting results presented on the Figure 5 are for the flow rate Q_{4L}. The system with H_2 controller has the widest bandwidth and there consists 7,25 [Hz]. It means it is almost twice (100 [%]) wider than in the case of PID system (3,8 [Hz]). The characteristics has the resonant pick for 3,2 [Hz]. It reaches 3 [dB].

Step responses of the closed-loop system with different parameters and controllers are also different (Henzel, 2004). The signals are unsatisfactory for robust controllers (H_2 i H_{inf}). The system was strongly disturbed and the step responses were more oscillatory. Similar tests were carried out for different supply pressures. In this case system with $H_{\infty opt}$ controllers works properly.

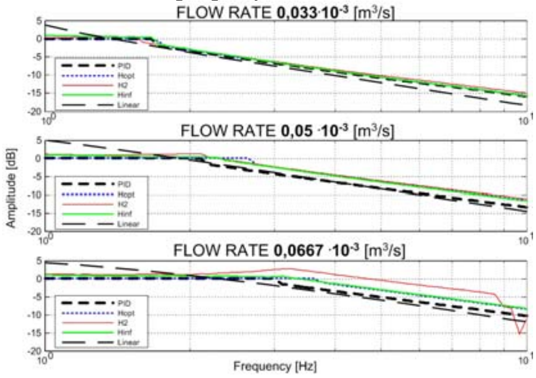

Fig.8. The frequency characteristics.

In summary, the servo-drive with optimal robust controllers is the best in practical applications. Such system has the best dynamic and static parameters.

5. VIBRATION CONTROL SYSTEM

The main problem in structure vibration control is the availability of the actuators with sufficiently high control force and wide bandwidth. For vibration control of rotating parts (rotors) we use usually the magnetic bearings [10] or linear piezoelectric actuators [11]. For non-rotating structure to vibration control we use "structure borne" actuators as: reaction wheels, control moment gyros, proof-mass actuators, piezo strips, etc. [2]. For the further considerations about machine tools vibration control we take into account the piezoelectric strip as an actuator and as a measurement unit [12].

5.1. Piezoelectric vibration control system

Consider cantilever beam as a model of a lathe tool with piezoelectric actuator and sensor as in Fig. 9.

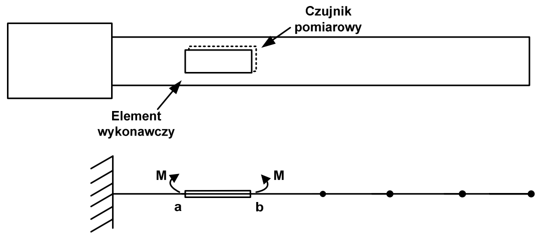

Fig.9. Beam with piezoelektric actuator and sensor.

The beam with actuator and sensor forms an open-loop system. The open-loop system was identified in the fequency domain. The amplitude-fequency characteristics is shown in Fig. 10. We can notice the resonances for frequencies: 10, 60, 180[Hz] and the anti-resonanses for frequencies: 1.3, 20 i 150[Hz]. We adjusted mathematical model (Fig. 11) by proper choice of the damping coefficients ξ. The lead member of the transfer function is connected with dynamics of the amplifier used to supply the piezoelectric actuator.

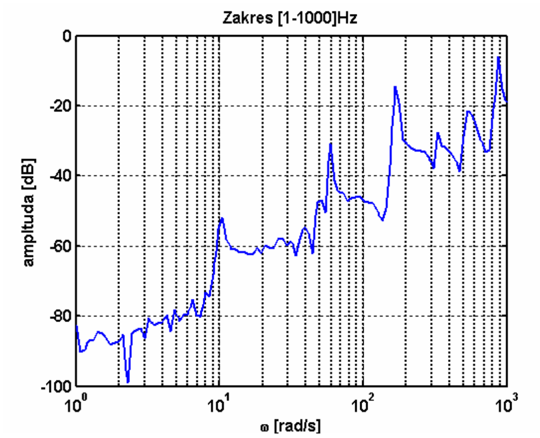

Fig.10.Recorded amplitude-frequency characteristics of the open-loop system.

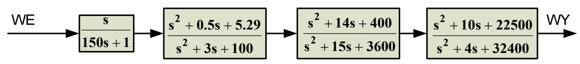

Fig.11. Transfer function of the open-loop system.

The Bode characteristics of the transfer function from Fig. 11. are presented in Fig. 12. We used Matlab and SISO Design Tool [13] to consider the influence of different controllers on the dynamics of closed-loop system. The characteristics of closed-loop system (Fig. 14) with inertial controller are presented in Fig. 13. Comparing Fig. 12 and 13 we can notice that the reduction of the highest peak of amplitude approach 30 [dB].

The second-order filter in the feedback loop assures stronger damping and bigger roll-off (–40dB/dek.) than in the case of inertial controller. But its drawback is connected with increasing the sensitivity of system stability to the parameters of identified model. We can notice it by the comparision of root locus plots for both controllers: inertial (see Fig 17) and filter (see Fig. 18).

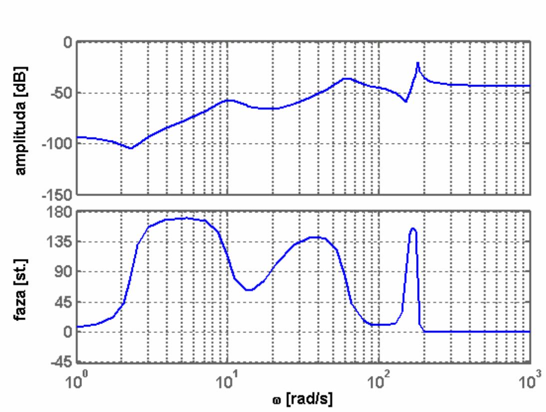

Fig.12. Bode characteristics of the identified model of open-loop system.

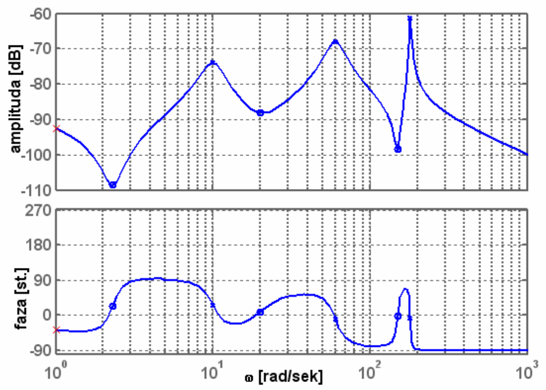

Fig. 13. Bode characteristics of the system with inertial controller.

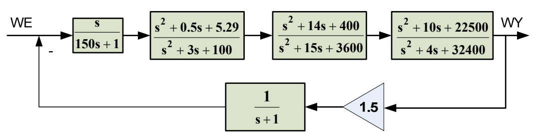

Fig.14. Closed-loop system with inertial controller.

For comparision the the second-order filter was used as a controller (Fig.15). The Bode characteristics of the closed-loop system with such controller are given in Fig. 16.

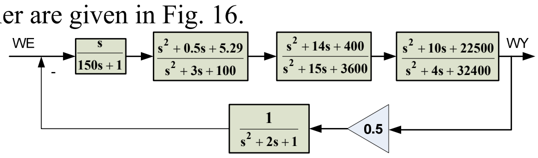

Fig.15. Model of system with second-order filter as the controller.

The closed-loop system with inertial controller is awans stable since branches of plots are in the left half-plane for all controller gains. System with second-order filter becmes unstable for some controller gains. For uncorrect model the system designed as stable one can appear unstable.

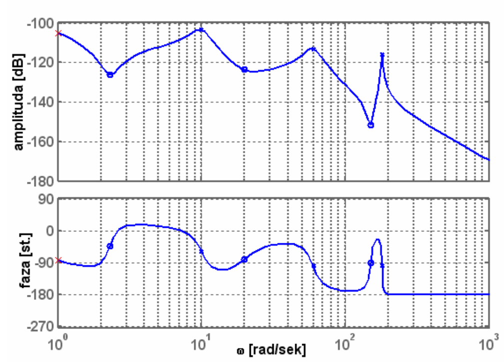

Fig. 16. Bode charakteristicsof the closed-loop system second-order filter.

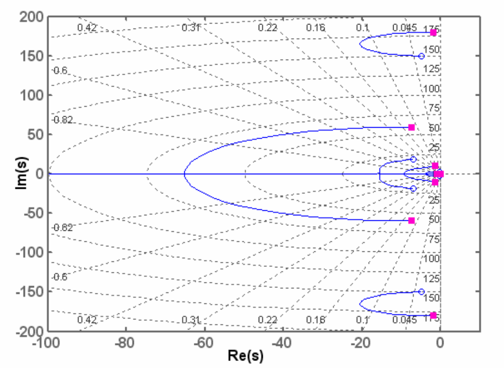

Fig. 17. Root locus for the system with inertial controller.

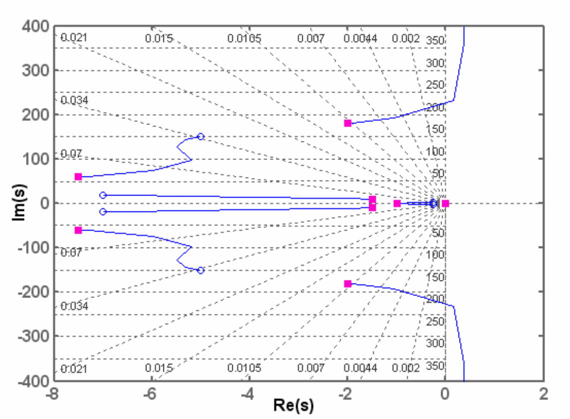

Fig. 18. Root lotus of the system with a second-order filter.

5.2. Experimental results

The inertial controller was verified in experimental way by using of it in the vibration control system of beam from Fig. 9. Durig transient response the control system was lock-in and the results are seen in Fig. 19. It was calculated that logarithmic decrement of damping increased from δ=0.2 to δ=0.51.

The amplitude-frequency characteristics is presented in Fig. 20. We see the high level of the damping in the whole range of frequency. Characteristics is similar to respective characteristics of the simulated closed-loop system from Fig. 13.

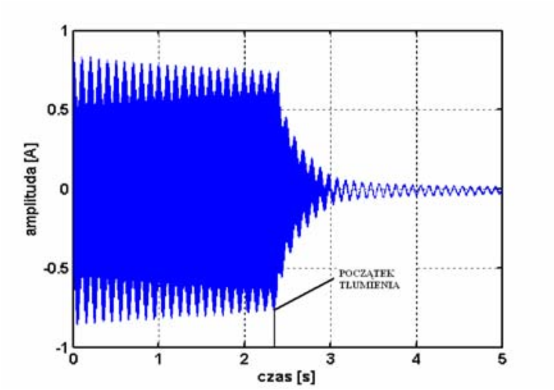

Fig. 19. Impuse response of experimental beam. The control system starts after 2.4 s.

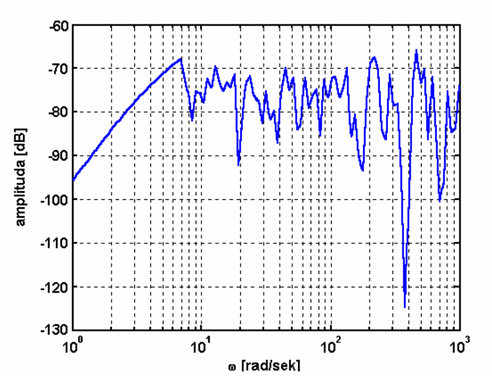

Fig.20. Amplitude-frequency characteristics of the closed-loop system with inertial controller recorded on the lab stand.

6. CONCLUSIONS

Machatronic approach to design gives new possibilites in the improvement of economic and technical properties of machine tools. In the paper we considered influence of the local control loops on the machine dynamics. We omitted dynamics of the main body of the machine tools. The dynamics of complete machines will be considered in the future works. After this early-stage consideration we can come to the following conclusions.

1. It is evident that in the similar way we can use global control loops to improve the dynamics of the system: machine-tool-machining piece. In this case it is desired to measure the localization of the tool.

2. The model-based control systems are better than artificial inteligence based control systems. So more we can resign from adaptive methods since they are sensitive to adjusting of models to actual system. The robust controllers seems to be a good solution to control the vibrations and motions of machine tools.

3. New materials like piezoelectrics or alloys with shape-memory can be used as sensors/actuators in the control loops. Since they can be located in any part of the machine tool as a parameter-distributed elements we can design so called "smart" structures. Smart structure is used to shape the dynamic characteristics of machine.

4. In machine tools with smart structures we can also deform the system to reach desired location of tools and machinig piece. So we can resign from high rigity of machine to reduce mass and energy costs.

5. New tools like electrospindles with fast rotating tools (120000 rev/min) release forces acting during the cutting process. If they are equipped in magnetic bearings there is some possibilites to change the location of the tool. It gives additional degrees of freedom and make the machine tools more flexible to the production requirements.

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