MACHINE TOOL VIRTUAL MODEL

J. Vesely, M. Sulitka
Research Center of Manufacturing Technology
CTU in Prague, Czech Republic

e-mail: j.vesely@rcmt.cvut.cz

Machine tool development for high speed and highly precise cutting demands employment of advanced simulation techniques, which can enable already in the machine tool design phase to model the feed drive dynamic properties and feed drive control. The paper describes the current state of feed drive complex models development in the RCMT. Influence of an appropriate approach to machine frame structure modelling on the feed drive dynamic properties prediction is discussed together with the influence of the ball screw feed drive mechanical structure parameters. Shown are examples of complex models application in the tasks of machine tool development and optimization. Vision of the machine tool virtual models development aims at the possibility to simulate the cutting process with the employment of the Hardware in the Loop systems and cutting process model.

Keywords
Machine Tools, Virtual prototyping, Feed drive optimization, Feed drive dynamic properties, Feed drive complex modelling

1 Introduction

Approach to machine drive modelling has undergone significant progress in the last decade. The paper aims to outline the new procedures and trends in mechatronic machine tool modelling, giving the opportunity to predict in a better way the feed drive dynamic properties and controller setting. The phenomenon of machine tool mechanical structure and feed drive control interaction has been described in many technical texts. Relevant capture of these interactions and next improvement and extension of the machine tool virtual models is subject of research conducted in the RCMT.

2. Model choice

2.1 Test bed

Test bed that substitutes for machine tool drive with flexible mechanical structure was constructed enabling a better view on to the problem of interaction of feed drive control with mechanical structure dynamic properties. The test bed consists of two servomotors that are connected through the composition of flexible steel rod and inertial mass (see Figure 1).

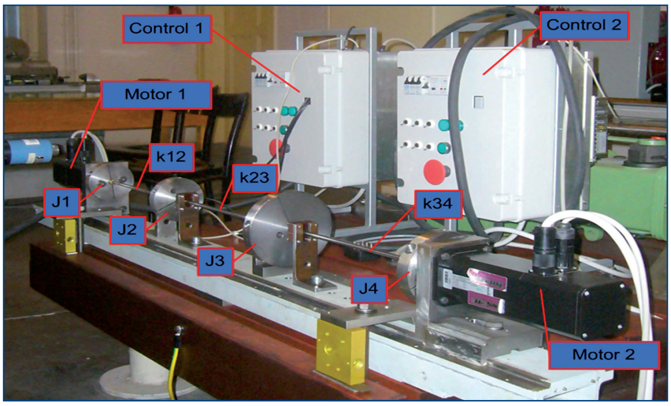

Figure 1. *Test bed.*

As a result we get mechanical structure that is very close to the ideal four-mass system (see Figure 2).

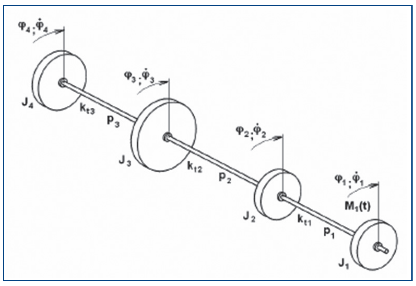

Figure 2. *Mechanical structure of the test bed.*

The correspondence of the discrete model of four masses and three springs is validated by means of the frequency characteristics correlation (transfer function between acceleration and moment on the mass 1 – see Figure 3) between this model and the real test bed behaviour.

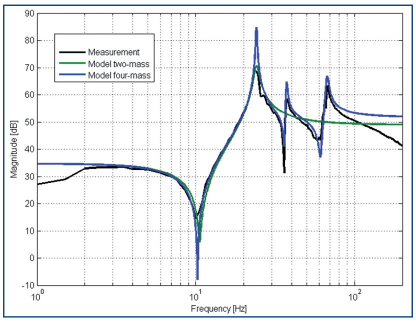

Figure 3. *Transfer function between acceleration and moment on the mass 1.*

Control is carried out by means of the dSpace system (DSP) and the cascade control equivalent to commercial products in machine tool drives is created. Advantage of the DSP use is in the possibility to measure and analyse an extensive amount of signals.

For research of the mechanics and control link, test bed provides an environment with well known properties. Its mechanical structure can be very easily described and thus context of the control and mechanical interaction can be well revealed. On one hand, mechanical structure with such discrete configuration represents a simplification in the comparison with machine tool real structures. On the other hand, because of its multi-mass configuration that generates eigenfrequencies and antiresonance frequencies in the range of 10 – 80 Hz, conclusions obtained by means of this system can be generalized for the real feed drives.

The main question before the mechatronic model is created is how precise the model should be and what the model should cover. We also have to know what kind of simplification we can afford to keep the possibility of obtaining relevant results. At the beginning of this planning we have to define dynamic requirements put on drives (i.e. velocity, acceleration and precision requirements). Now, we will show the process of model choice on the test bed example. For evaluation of the model quality frequency characteristics and time response of the velocity control step function will be employed. Setting of the velocity control is dependent on the system mechanical properties and position control loop setting will emerge from its properties [Soucek 2004]. Tests in the velocity control represent therefore an evidential evaluation of models in relation to reality.

2.2 One-mass model

For examination of the model applicability, let's start with the elementary approach. The model, in which the mechanical structure is described as a sum of its all inertial masses neglecting the flexibility between the masses, represents the kinematic part of the drive. Flow diagram of control shows the Figure 4.

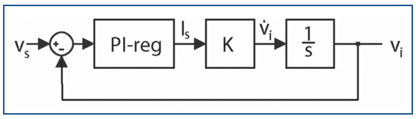

Figure 4. *Flow diagram of control with one-mass mechanics.*

Motor torque constant and inertial mass are covered with the gain K. Setting of PI regulator was performed in a way that the model velocity control passband corresponds with the real behaviour. Comparison between the simulated velocity control loop frequency characteristic and the real measurement is in the Figure 5.

From the comparison of time characteristic in the Figure 6, we can assume that for the mentioned passband of velocity control, the model with one mass mechanical system is relevant. To put it differently, dynamical characteristics of feed drive control is not influenced by the mechanical structure dynamics.

The model mentioned is valid only for low passband of velocity control. Many simulations demonstrated that if the passband is approx. 50% of the first antiresonance mechanical frequency ω_{m1} (corresponds with the first drop in the amplitude characteristic of velocity control loop), the model is in accordance with reality.

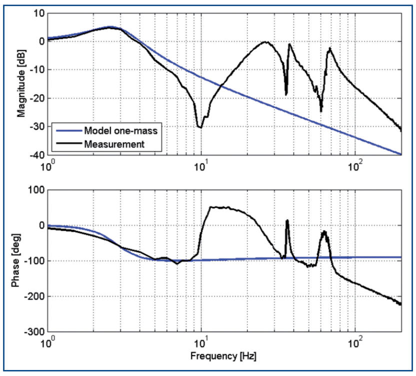

Figure 5. *Frequency characteristic of velocity control loop.*

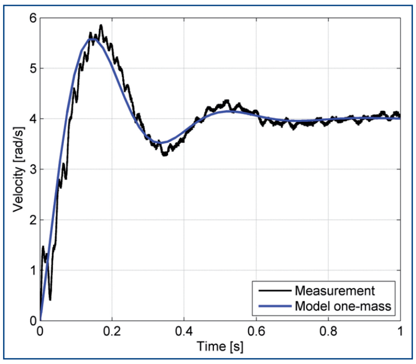

Figure 6. *Step response of velocity.*

2.3 Two-mass model

Applicability of the one-mass mechanical model is limited by the ω_{m1} value. If we want to analyse properties for higher passband (higher gain of PI-regulator), one-mass model fails (see Figure 7, 8), passband different from the reality is simulated.

The need to include relevant dynamic properties of the mechanic structure into the model arises. The simplest way how to fulfil this requirement is to employ two-mass representation of the system, by means of which the correspondence of the ω_{m1} and first eigenfrequency values is provided (see green curve in the graph in Figure 3). Flow diagram of control is then supplemented with transfer function of two-mass system (Figure 9).

We will demonstrate the comparison of simulation and real behaviour on the basis of the velocity control loop with higher passband.

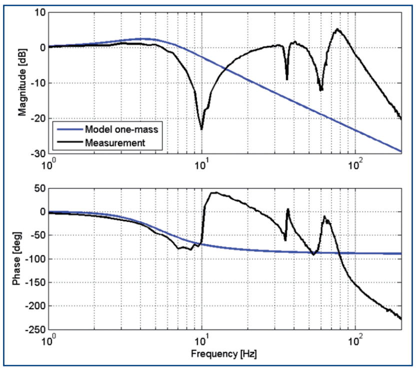

Figure 7. *Frequency characteristic of velocity control loop.*

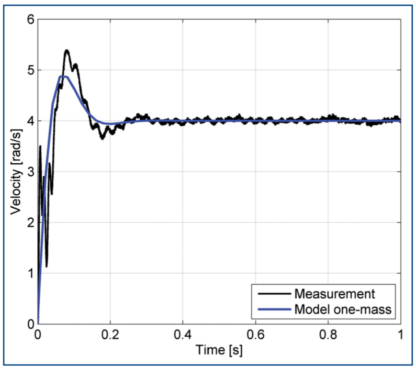

Figure 8. *Step response of velocity.*

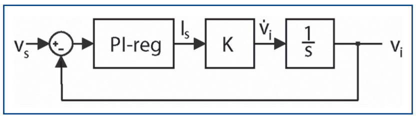

Figure 9. *Flow diagram of control with two-mass mechanics*

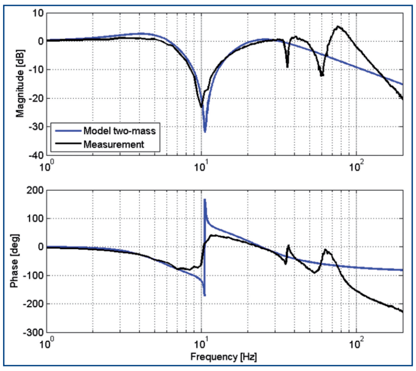

Figure 10. *Frequency characteristic of velocity control loop.*

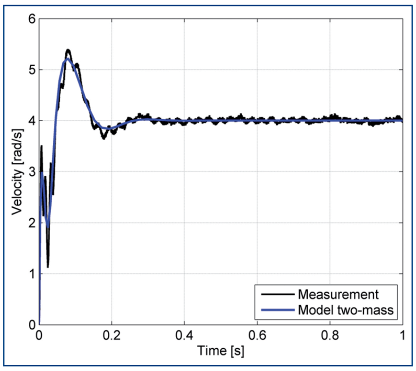

Figure 11. *Velocity step response.*

If we use model with two-mass system we get frequency characteristic of velocity control loop with good correspondence till the ω_{m1} value (Figure 10).

Time response of step velocity confirms the good conformity of the model and real behaviour (Figure 11).

2.4 Complex model

The ω_{m1} that is well captured by means of the two-mass model is often used as a limiting factor for the velocity regulator setting. We will show now that this statement cannot be considered as generally valid even if it is applicable in many other cases.

We will go over to higher setting of the PI velocity regulator. With the machine tools designed for HSC technology the main target is to achieve high setting of the controller parameters. At first we increase the velocity control gain of the test bed. In the velocity loop frequency characteristic and in the time step response, the frequency value of 100 Hz is the limiting one, not the first resonance frequency itself. Simulation with two-mass configuration doesn't reflect this problem and it would permit higher gain of the regulator (for the comparison see Figure 12, 13).

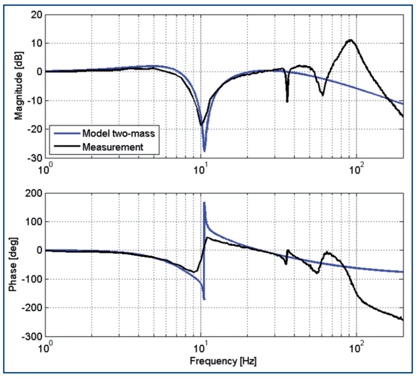

Figure 12. *Frequency characteristic of velocity control loop.*

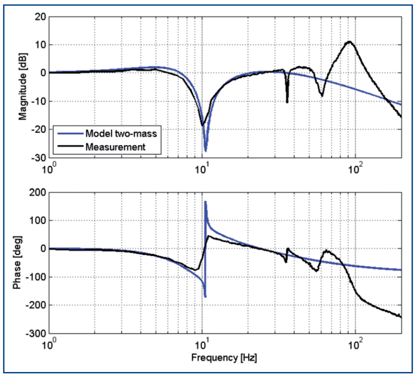

Figure 13. *Step response of velocity.*

Following the detailed analysis we can see, that the origin of the 100 Hz frequency problem comes again from the mechanical structure. This value is neither directly eigenfrequency nor antiresonance frequency of the mechanics. Its origin is however at 70 Hz eigenfrequency. This value is being shifted right to the value of 100 Hz by means of the velocity control loop. We can demonstrate this phenomenon on GMK and it is explained in the literature (e.g. [Soucek 2004]).

Velocity control should also be able to increase damping of this eigenfrequency. However, why this is not the case, the next extension of the model will explain. At first we create full four-mass system. Simulation of velocity loop corresponds better with reality, but 100 Hz is well damped. We add the model of current control loop with filter of the 1st order to the input (as it is in real version). The frequency now significantly rises over the limit of 0 dB in amplitude characteristic of the velocity loop (Figure 14, 15).

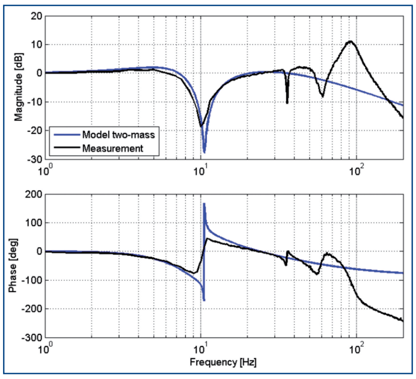

Figure 14. *Frequency characteristic of velocity control loop.*

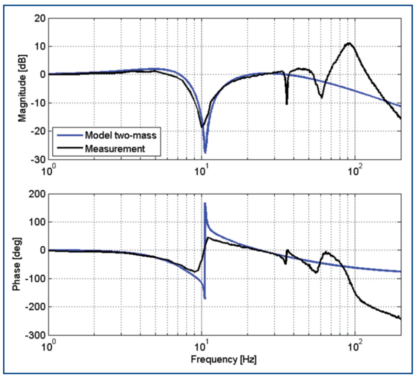

Figure 15. *Step response of velocity.*

What happened in this case is that the current control loop filter significantly distorts the phase of input signal. Force effect that could lead to elimination of resonant oscillation is thus shifted and on the contrary it evokes another eigenfrequency. As a paradox this filter that should reduce oscillations brings about the mentioned problem. To make this filter work effectively, it would have to be positioned lower, to get significant decrease in amplitude. But it would meet lower eigenfrequency and the result would be worse. That is the reason the use of filters for mechanical frequencies is only recommended for isolated higher mechanical eigenfrequencies.

The problem of high mechanical frequencies has been observed in real machines as well. The phase shift evoked by current control itself is usually in the zone of 500 Hz. However, if in even in this frequency range some eigenfrequencies standing close to each other occur, the same problem with filter application, as explained above, arises and the only possible solution is to lower the gain of velocity control. Limitation thus doesn't have to come from the 1st anti-resonance frequency, but it comes from higher eigenfrequencies that interact with delay and with phase shifts in the following control processes.

It is evident, that detailed modelling and maximal effort to approximate real conditions has significant importance. The use of particular simplifications (mainly two-mass approach) can help for initial insight into the system, but generalizing of conclusions can rather be misleading. As we have mentioned at the beginning, the simplification has to be always derived from the final drive control requirements. In the case of machine tools we require maximal dynamics that lead to the use of thoroughly elaborated models.

3. Ball screw feed drive coupled models

As it has been shown, relevant model of the machine tool feed drive dynamic properties is a key component of coupled mechatronical models, which include the interaction of feed drive mechanical structure with servodrive loops. Typical machine structure is characterized by several movable parts, which are joined together by means of linear guiding systems and bearings. Structure of the machine features eigenfrequencies, which have to be included in the feed drive mechanical model.

FE method is nowadays widely used for description of static and dynamic properties of flexible bodies. However, the FEM does not directly provide the possibility to create kinematical models of systems with flexible bodies and therefore various approaches to modelling of such systems are being researched and developed. Common feature of those techniques is that the size of the flexible bodies FE models has to be substantially reduced. In this paper two methods are mentioned with respect to their availability in the commercial software tools.

3.1 Flexible Multi-Body Systems

Kinematical chains of flexible bodies can be created as Multi-Body Systems (MBS) in various software environments, e.g. ADAMS. Flexible bodies are imported into such specialized software in a reduced form, typically as a modal neutral file (*.mnf), the creation of which is commonly provided in FE software tools. Creation of the *.mnf file is based on the Craig-Bampton reduction technique [Sellgren 2003], which allows the user to select certain number of eigenmodes kept in the model and to choose important interface nodes used for joining of the body with another bodies. Resulting reduced model combines both real and modal coordinates. The process of creating the Flex-MBS models is relatively time demanding, however the main unfavourable feature is that with the implementation of the Craig-Bampton technique in commercial FE softwares it is not possible to easily omit the eigenmodes, which are not important for the system. Consequently, the size of the MBS system is relatively big for further elaboration within the feed drive servo-control loop model.

3.2 Modal decomposition technique as a tool for feed drive mechanical structure modelling

The approach of modal decomposition technique for modelling of kinematical chains assembled from flexible bodies can be utilized thanks to the fact, that in the servo-control loop model, typically prepared in Matlab/Simulink environment, the feed drive mechanical part can easily be represented by means of the State-space description. However, common FEM programs do not work with State-Space models and do not offer a direct possibility of assembling the State-Space matrixes. There are generally two possibilities of how to carry out the transformation – the first one is based on the mass, damping and stiffness matrixes of the FEM system, the other one goes out from equations of motion written in modal coordinates and solution of modal analysis. Commonly, machine tool FEM models contain $10^5 - 10^6$ DOFs, which means, if the transformation had to be based on FEM matrixes, very strong reduction of physical coordinates, leading in its effect to the loss of model accuracy, had to be performed. On the contrary, assembling the State-Space matrixes in modal coordinates provides very efficient possibility of selecting both the eigenmodes important for a certain transfer function and selection of physical coordinates, between which transfer functions are evaluated or to which feed drive mechanical structure is joined. Thanks to it very small size State-Space matrixes, which are easy to handle in Matlab/Simulink, are obtained, without the need of the FEM model reduction in physical coordinates. System of State-Space equations is written as

$$\begin{aligned} \dot{\mathbf{x}} &= \mathbf{A}\cdot\mathbf{x} + \mathbf{B}\cdot\mathbf{u} \\ \mathbf{y} &= \mathbf{C}\cdot\mathbf{x} + \mathbf{D}\cdot\mathbf{u} \end{aligned} \tag{1}$$

where x is the state vector of the system, u the vector of forces and y the output vector. Matrixes A and B are the input matrixes, matrixes C and D are matrixes of the system output. From modal analysis we get the matrix of normalized eigenvectors V and spectral matrix of eigenfrequencies Λ:

$$\mathbf{V} = \begin{bmatrix}\mathbf{v}_1 & \mathbf{v}_2 & \cdots & \mathbf{v}_m\end{bmatrix} = \begin{bmatrix} v_{11} & v_{12} & \cdots & v_{1m} \\ v_{21} & v_{22} & \cdots & v_{2m} \\ \vdots & \vdots & \ddots & \vdots \\ v_{n1} & v_{n2} & \cdots & v_{nm} \end{bmatrix}$$
$$\Lambda = \begin{bmatrix} \omega_1^2 & 0 & \cdots & 0 \\ 0 & \omega_2^2 & \cdots & 0 \\ \vdots & \vdots & \ddots & \vdots \\ 0 & 0 & \cdots & \omega_m^2 \end{bmatrix} \tag{2}$$

Basic equation of motion can then be written in modal coordinates as

$$\mathbf{E}\cdot\ddot{\mathbf{q}} + \mathbf{C}_q\cdot\dot{\mathbf{q}} + \Lambda\cdot\mathbf{q} = \mathbf{V}^T\cdot\mathbf{f} \tag{3}$$

where E is unity matrix, Cq matrix of modal damping, q vector of modal coordinates and f vector of forces. It can be shown, that after substitution the State-Space matrixes get the form

$$\mathbf{A} = \begin{bmatrix} \mathbf{0} & \mathbf{E} \\ -\Lambda & -\mathbf{C}_q \end{bmatrix}, \qquad \mathbf{B} = \begin{bmatrix} \mathbf{0} \\ \mathbf{V}^T \end{bmatrix}$$
$$\mathbf{C} = \begin{bmatrix} \mathbf{V} & \mathbf{0} \end{bmatrix}, \qquad \mathbf{D} = \begin{bmatrix} \mathbf{0} \end{bmatrix} \tag{4}$$

Procedure of the ball screw feed drive models creation is currently based in the RCMT on the principle of separate modelling of the machine frame structure and ball screw feed drive mechanical part.

3.3 Model of the ball screw feed drive mechanical structure and connection with the machine frame

Ball screw feed drive mechanical structure is described separately from the machine frame FEM model. A discrete model with para-

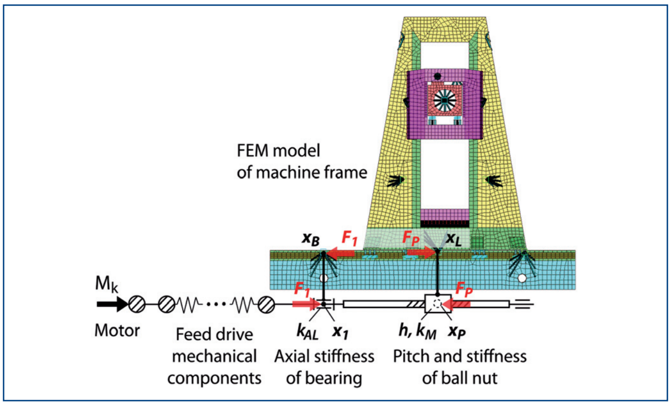

Figure 16. *Discrete model of the ball screw feed drive and connection to the machine frame.*

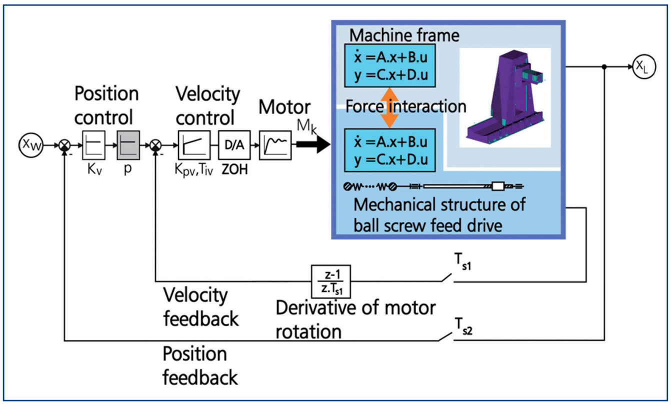

Figure 17. *Complex model of servoregulation with ball screw feed drive model coupled with machine frame.*

meters representing inertial and stiffness characteristics of feed drive mechanical components is created, whereby ball screw itself is modelled as a 1D continuum keeping both axial and rotary DOFs, between which a constraint equation is set up. System of equations of motion of this system is transformed into the State-Space. General scheme of the ball screw feed drive model connected with machine frame gives Figure 16, corresponding symbolic block scheme of the servoregulation model shows Figure 17.

4. Application of machine tool complex models

Machine tool complex models are created for the machine tool producers both in the design phase of a machine tool development and in the tasks of the identification and optimization of already existing feed drives. Typically, the below mentioned tasks are solved by means of complex modelling:

- structural optimization of the machine tool frame with respect to the feed drive dynamic properties
- ball screw feed drive mechanical structure optimization
- simulations of attainable control parameters
- path profile control optimization with respect to tool to workpiece vibration minimization
- testing of alternative control techniques.

Verification of feed drive complex models is performed by means of feed drive diagnostic measurements and modal analysis with motor invoked excitation. On the example of a H50 machine tool of from the production of ZPS-Tajmac Zlin firm it is shown, that feed drive complex models, the creation of which is based on the above mentioned modal decomposition technique and separate modelling of feed drive mechanical, are capable of providing relevant description of the feed drive dynamic properties. At the same time, importance of the machine frame FEM model employment within the feed drive model is demonstrated on the comparison with the model, in which the movable machine column is represented just by means of a lumped mass (see Figure 18). Structure of the column features first eigenmode of the paralelogram shape (Figure 19), which is typical for milling machine tools of the similar conception. This eigenmode significantly interfere with the dynamic properties of the feed drive mechanical structure and therefore it limits the attainable dynamics of the feed drive control.

Velocity loop frequency transfer function (Bode plot) of the investigated motion axis is depicted on the Figure 20. It may be seen, that with the model considering just the lumped mass representation of the movable column, the first important antiresonance frequency is not captured properly (blue curve). On the contrary, feed drive complex model based on the machine frame FEM model (green line) exhibits very good concordance with the real feed drive behaviour not only with respect the first antiresonance frequency determination, but also in the range of higher frequencies.

Separate modelling of the ball screw feed drive mechanical structure and machine frame gives the opportunity to easily perform e.g. the sensitivity analysis of the feed drive first antiresonance frequency to the parameters of the ball screw feed drive components. Findings similar to the example shown on the Figure 21 enable to detect a parameter with the highest impact on the feed drive dynamic properties.

Employing the modal decomposition technique for the description of the machine frame dynamic properties enables to prepare a machine tool FEM model with more movable parts as well. An example of such machine model used for simulations of circular interpolation is given on the Figure 22. Circular interpolation results obtained with the machine tool complex model and by means of measurements are shown on the Figure 23. Very good concordance of the simulated quadrant errors with the measured data may be seen, whereby specifically set model of passive forces has been implemented in the machine tool complex model.

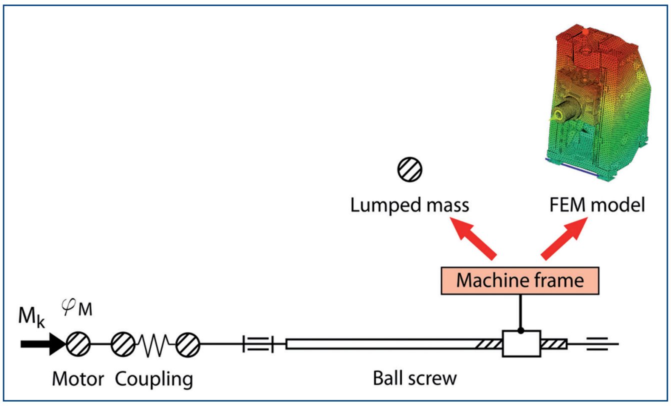

Figure 18. *Representation of machine tool movable column by means of lumped mass simplification or FEM model.*

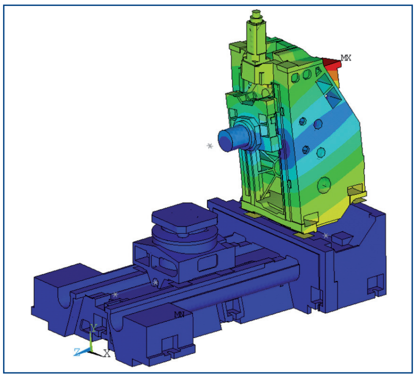

Figure 19. *Movable column of the machine tool X-motion axis and its first eigenmode with paralelogram deformation.*

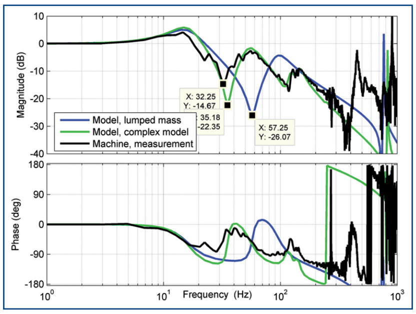

Figure 20. *Bode plot of the velocity loop. Comparison of the model with lumped mass representation of movable column, FEM based complex model and real measurements.*

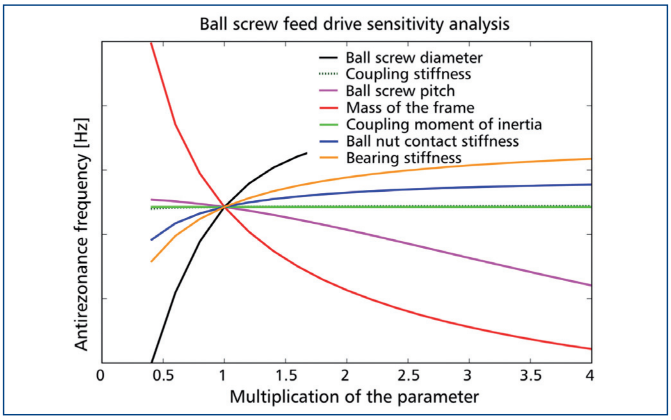

Figure 21. *Sensitivity analysis of the first antiresonance freqeuncy to the parameters of the ball screw feed drive components.*

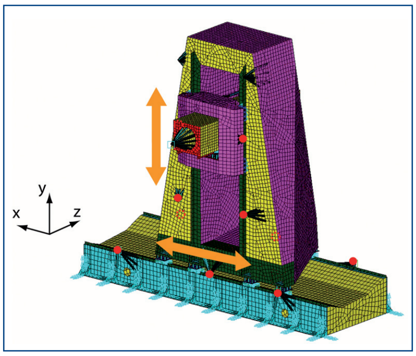

Figure 22. *Machine frame FEM model with two movable axes.*

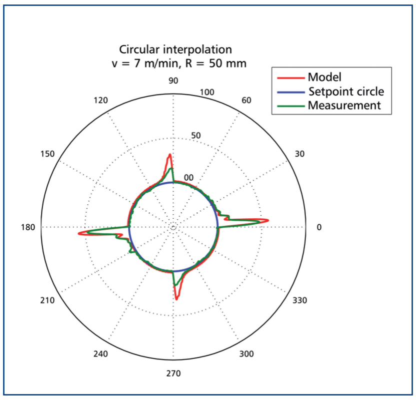

Figure 23. *Circular interpolation. Simulation and comparison with measurements.*

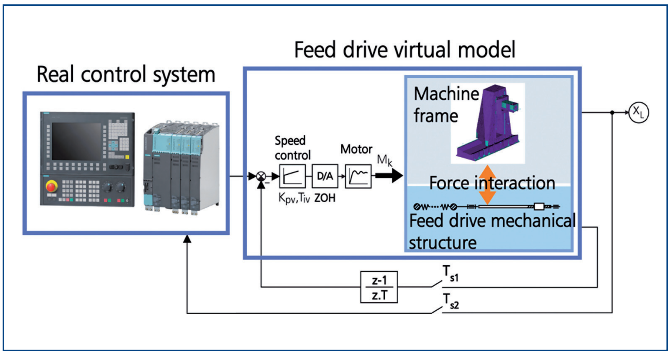

Figure 24. *Scheme of the Hardware in the Loop system.*

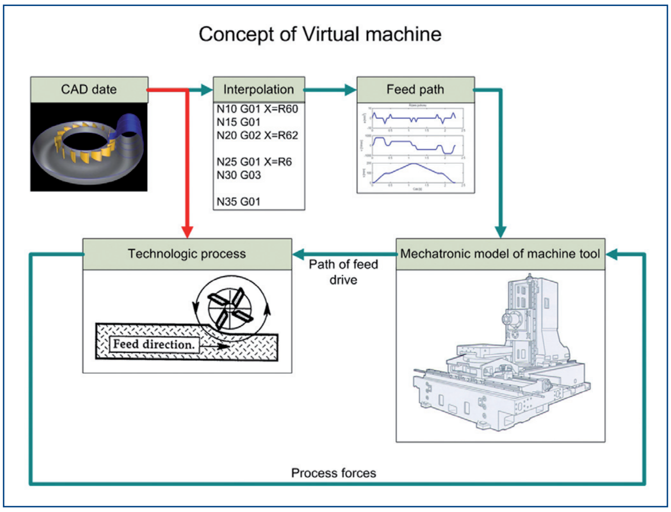

Figure 25. *Concept of virtual machine with process model*

5. Next development of the machine tool virtual models

Complex modelling aims at creation of machine tool virtual models, which will enable to simulate the attainable workpiece precision and surface quality. Development of advanced machine tool virtual models is connected with research on the fields of passive resistance forces modelling, damping of structures, development of machine tool models reflecting the changing dynamic properties of the frame with respect to different mutual positions of the machine tool movable parts, research of cutting process forces and cutting process stability. Currently, the main research interest in the RCMT is focused on the development of Hardware in the Loop (HWinL) system and development of the model for cutting process forces calculation.

The HWinL system couples real NC control with the machine tool virtual model and provides therefore the possibility to test either the machine tool virtual model or the NC control code even before the real machine tool is built. System is being developed in the RCMT in cooperation with the Czech producer of NC control systems MEFI.

Besides the detection of the machine tool dynamic properties analysis by means of virtual modelling, another concern of the next development on this field in the RCMT focuses on cutting process virtual simulations. The existing model, as is presented in the Figure 24, will be extended with a model of technologic process (Figure 25). Be means of such a model, link between the force current, generated by cutting process, and mechatronic model of machine tool will be established. Only then the workpiece surface simulation will be possible and becomes an evaluation factor of the machine tool quality.

6. Conclusions

Presented method of machine tool virtual modelling is becoming sought-after in practise as a tool enabling to perform complex analysis and optimization of machine tool feed drives and machine tool structure. It significantly reduces time of development and reveals real possibilities of the construction during a new machine design in the phase when possible changes are not economically critical. Extending the coupled feed drive models with the cutting process model and implementing this model into the HWinL system, it will be possible to effectively optimize the machine tool design with respect to workpiece, i.e. requirement on its precision, machining time costs minimization.

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Contacts:

Dipl. Ing. Jan Vesely
CTU in Prague, Research Center of Manufacturing Technology
Horska 3, 128 00 Praha 2, Czech Republic
tel.: +420 221 990 958, fax: +420 221 990 999
e-mail: j.vesely@rcmt.cvut.cz