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Rheological Model Design of Machine-Tool Slides with Plain-Bearing in Dovetail Slides

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***Abstract* – This paper deals with a modeling methodology design of an automatic lathe cross slides with plain-bearing in dovetail slides. A proposed rheological model of a slide assembly is based on partial FEM slide models mutually interconnected by linear springs and dampers. The model respects elastic slide assembly mounting onto the machine and a material damping in individual slides. The model verification is based on a comparison of calculated model characteristics with measurement results of a real machine, i.e. with static stiffness, with experimental modal analysis and with measurement of forced vibrations excited by a shaker and cutting forces during machining.**

***Keywords* – Dynamic Compliance, FEM Model, Linear Spring, Damper.**

I. Introduction

Cross slides of an automatic lathe with plain-bearing in dovetail slides are formed by three main parts: a stationary bed, a bed slide and a traverse slide.

The connection between carriage and the bed slide, resp. between the bed slide and the traverse slide is realized by a dovetail slide. The mutual contact between both parts happens on the dovetail slide surfaces. The surfaces are lubricated by oil, which firstly decreases friction and, secondly, dampens the vibrations between parts. The relative position of these connected parts is determined by coupling of a feed screw and a slide nut.

The modeling of slides with plain-bearing in dovetail slide can be beneficial especially due to possibility to characterize the slides dynamic behavior during machining, including spurious effects such as self-excited vibrations [1], [2].

One way of the modeling of slides with plain-bearing in dovetail slide is based the description of the oil film behavior between the sliding surfaces of the slides [3].

Another option is to create a slide assembly rheological model, which respects the damping between the assembly bodies [4], [5]. Such a model is presented in this contribution.

II. Slide FEM Model with Linear Springs and Dampers

The slide assembly FEM model creation procedure is documented in Figs. 1 to 7.

First, the cross slide main parts were simplified. Small details, such as tiny edge fillet, small slots etc. that have negligible influence on the static stiffness and the dynamic behavior, were removed. Partial FEM models of these parts were created in the next step - see Figs. 1 to 3.The models consist of a combination of tetragonal and interpolation elements with several millions nodes in total.

Every surface of a given slide that is in contact with a surface of the other slide in current assembly position was then split into three identical areas. Sort of “umbrellas” of elements were formed in both edge areas using rigid elements. These “umbrellas” served as a coupling between the slides. Each slide was connected with the other slide using four linear springs on the horizontal surfaces and four linear springs on the oblique surfaces of the dovetail slide. A linear damper was inserted in series with each spring. In order to simulate damping of relative motion of slides in the longitudinal direction two other linear dampers were defined - see Fig. 4. The slide joint detail is in Fig. 5, the slide assembly FEM model is in Fig. 6 (a tool holder was replaced by a prism on the traverse slide top surface). The assembly joint with the machine was realized as elastic using linear springs in the joining screw positions - see Fig. 7. The FEM model also takes account of the axial compliance and damping of nuts and feed screws mounting.

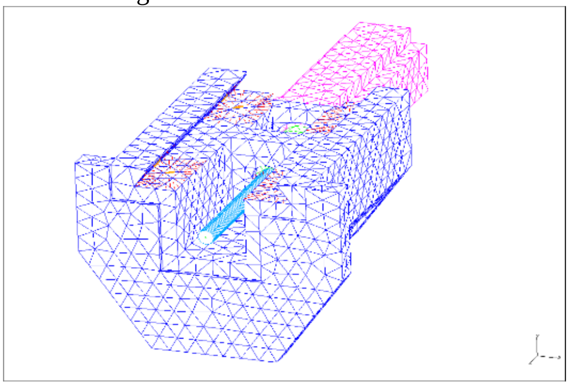

Fig.1. Bed FEM model

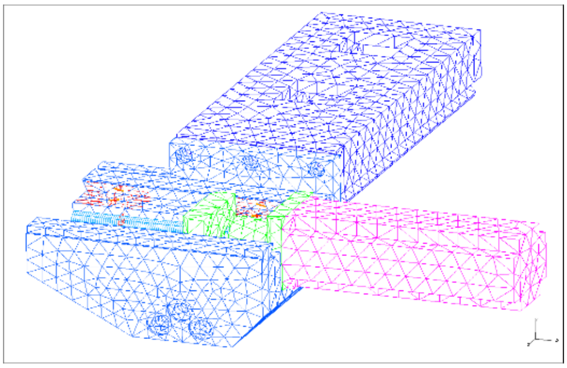

Fig.2. Bed slide FEM model

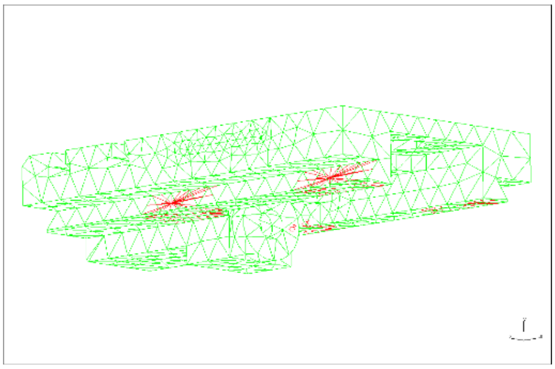

Fig.3. Traverse slide FEM model

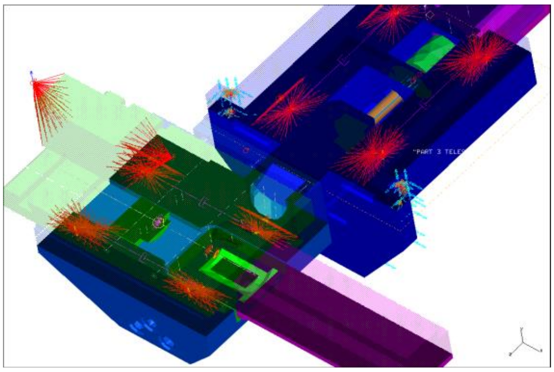

Fig.4. FEM model – slides joint positions using springs and dampers

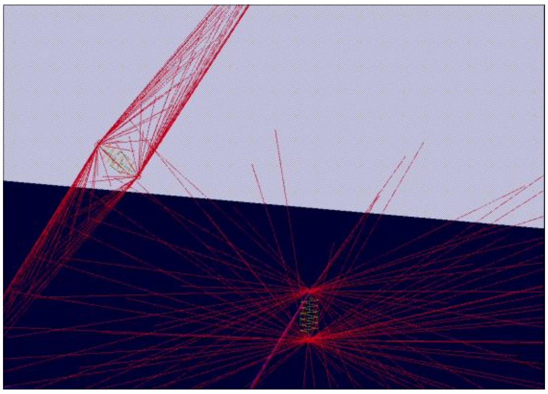

Fig.5. FEM model – slides joint using springs and dampers, detail

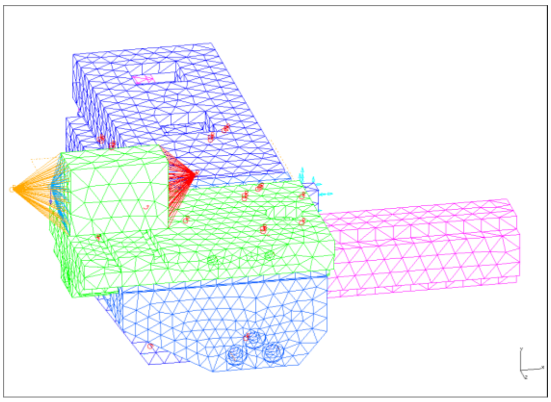

Fig.6. Assembly with tool holder FEM

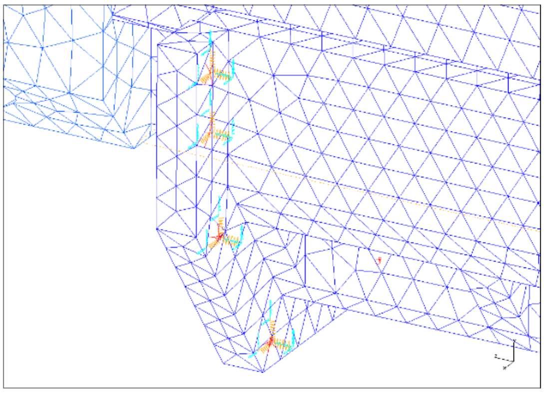

Fig.7. Assembly FEM model - elastic bed mount using springs

III. FEM Model Validation

Slide FEM model validation was realized by comparison of the NX I-DEAS simulation results with the experimental data acquired on a real machine, subsequently in three steps:

Step A: FEM model static stiffness calculation in FEM.

The FEM model deformation calculation caused by static forces and a comparison with measured slide deformation. Determination of approximate (in order of magnitude) spring's stiffness defined in the dovetail slides.

Step B: FEM model modal analysis.

The FEM model modal analysis calculation –e.g. first 30 modes and a comparison of eigenfrequencies and eigenshapes with the experimental modal analysis results. A correction of the spring stiffness in the dovetail slides and in the bed mount in order to obtain the eigenfrequencies for at least first 3 FEM model modes same as in the measurement results.

Step C: FEM model forced vibrations calculation.

Calculation of the FEM model response on the excitation forces spectrum (NX I-DEAS, module Response Analysis, Mode Acceleration Method) and a comparisonof the acceleration magnitude with the measured magnitude excited by a shaker. Determination of the viscous damping values of the dampers in the dovetail slides.

IV. Slide FEM Model Comparison with Experiment

The FEM model was validated by the procedure in section III.

The FEM model deformation loaded by a static force F_y = -1 000 N is in Fig. 8. The point of application is on the left side of the tool holder, see also Fig. 19.

The first 3 eigefrequencies of the final FEM model version are compared with the eigenfrequencies obtained by the experimental modal analysis in Table 1. The FEM model 3rd eigenshape is depicted in Fig. 9. The agreement between FEM and measurement results was very good with Modal Assurance Criterion values over 0.9.

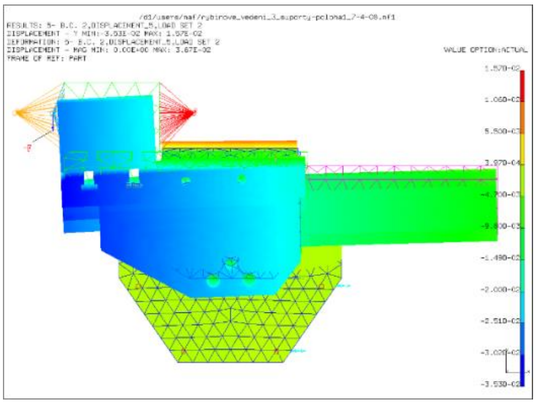

Fig.8. Assembly FEM model loaded by vertical static force F_y=-1000 N and self-weight(point of application is on the left side of the tool holder)

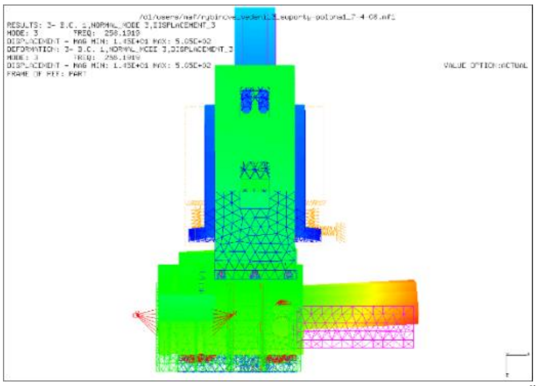

Fig.9. Assembly FEM model – modal analysis – 3rd eigenshape

Table 1: Comparison of the FEM model - version R8 first 3 eigenfrequencies with experiment

Spring stiffness - bed mount on the machine **$K_{4\,X,Y,Z}$ [N/m]**	1.0 E11	1.0 E11	3.5 E8
Spring stiffness - dovetail slide **K_R [N/m]**	1.2 E9		
Damping coefficient **B_{visc}^{I} / B_{visc}^{II}[N/(m/s)]**	1.0 E4/2.0 E4		
$\Omega_{1,2,3}^{exp}$ [Hz]	118	157	260
$\Omega_{1,2,3}^{FEM}$ [Hz]	126.7	143.7	265.1

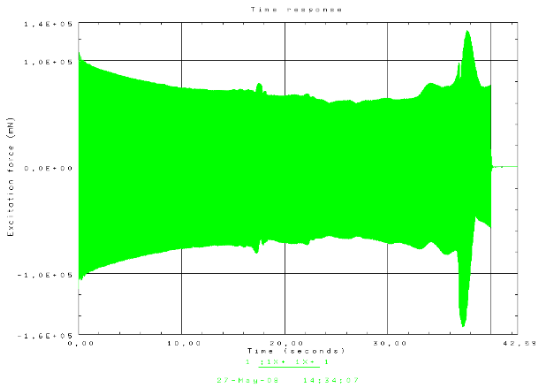

Fig.10. Shaker vertical excitation force F_y

In Fig. 10, the shaker driving force $F_y$time dependency is depicted. This force was applied in the FEM model tool holder at the knife tip positionand in the slides of the real machine as well. The force waveform was developed by a slow sweep of a harmonic force with frequency continually rising from 30 Hz to 430 Hz at the 10 Hz/s sweep rate [6].

The FEM model acceleration magnitudes (yellow) are compared with the measured values (green) for the shaker excitation in Figs. 11 and 12 at the measurement points no. 1 and 2, respectively. The $F_y$vertical excitation force had the time dependency from Fig. 10 and was applied at the knife tip position. The response was evaluated in the y axis direction in the measurement point no. 1, i.e. on the traverse slide close to the tool holder, resp. in point no. 2 on the traverse slide motor drive [4].

A relatively good agreement of the FEM model with the experimental results in studied frequency range to 430 Hz is obvious according to Figs. 11 and 12, concerning both the acceleration magnitudes and the resonant frequency peaks. The difference is caused by the real assembly nonlinearities and a complicated damping character in the dovetail slide [2]. These are not considered in the FEM model.

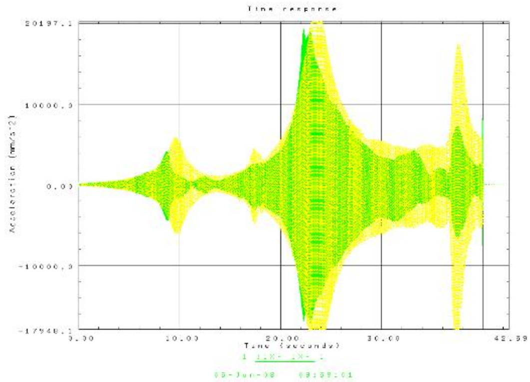

Fig.11. Acceleration magnitudes comparison of the FEM model (yellow) with values measured for the shaker excitation (green) at the measurement point no. 1 (traverse slide close to the tool holder), vertical-y direction

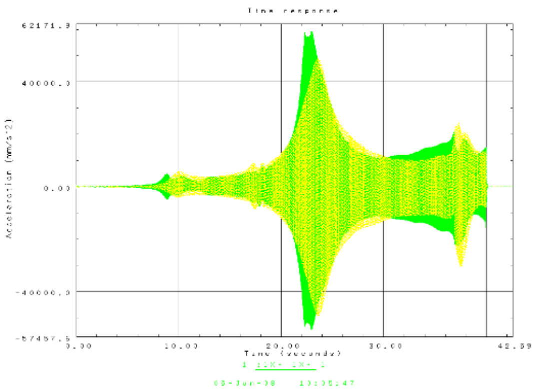

Fig.12. Acceleration magnitudes comparison of the FEM model (yellow) with values measured for the shaker excitation (green) at the measurement point no. 2 (traverse slide motor drive), vertical -y direction

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V. FEM Model of Dynamometer for Cutting Forces Measurement

A dynamometer for the cutting forces measurement during machining was designed and manufactured in order to measure and simulate the slide dynamic response on the cutting force excitation. The dynamometer includes two 3-axial force sensors Kistler 9251A [7]. Consequently, a FEM model of this dynamometer was created - see Fig. 13. The model modal analysis was calculated. The calculated FEM modal properties (Figs. 14 and 15) were compared with the dynamometer experimental modal analysis. A sufficient agreement was confirmed for frequencies up to 1 500 Hz.

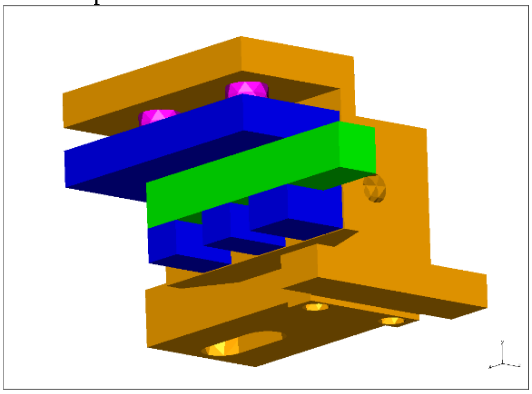


Fig.13. FEM model of dynamometer for cutting forces measurement

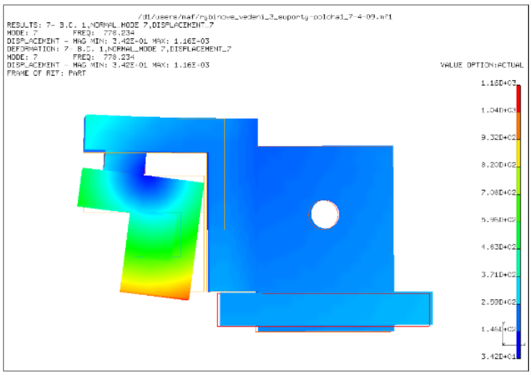


Fig.14. Dynamometer FEM model eigenshape for frequency 778 Hz

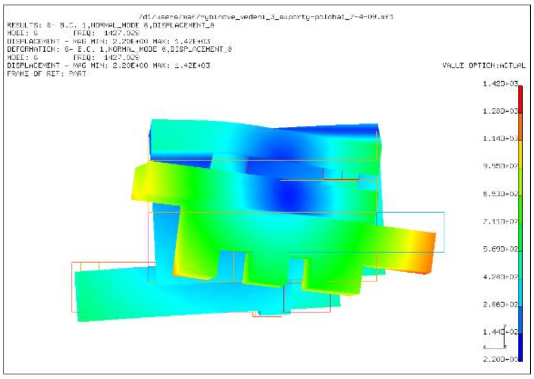


Fig.15. Dynamometer FEM model eigenshape for frequency 1 427 Hz

VI. Slides with Dynamometer FEM Model and Response on Cutting Forces Excitation During Recessing

The tool holder was removed from the former slide FEM model (section II., Fig. 6). It was replaced by the cutting forces measurement dynamometer - see Fig. 16. Previously obtained spring and damper parameters were preserved.

First, modal analysis using this model was carried out, i.e. first 30 eigenfrequencies were calculated. According the results the 4 lowest eigenfrequencies were approximately same as for the FEM model with the tool holder. Additionally, the respective eigenshapes were in agreement [4], [5].

The slides with dynamometer FEM model response on the excitation forces during recessing was investigated in the following steps. The measured forces time dependencies, $F_x=F_x(t)$, $F_y=F_y(t)$, $F_z=F_z(t)$, which acted on the knife by the workpiece, are depicted in Fig. 17. These forces were defined as the excitation forces in the slides with dynamometer FEM model.

Calculated and measured responses during recessing are compared in Fig. 18. A relatively good agreement is obvious. The model is therefore appropriate for further machine design.

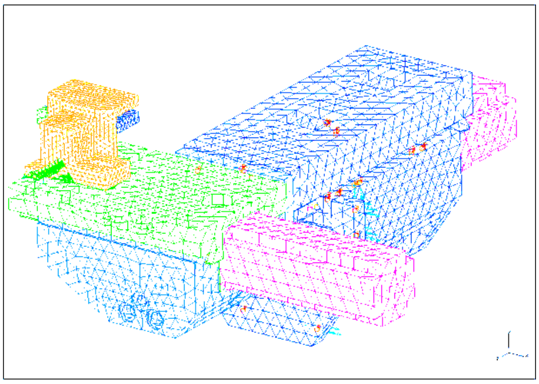
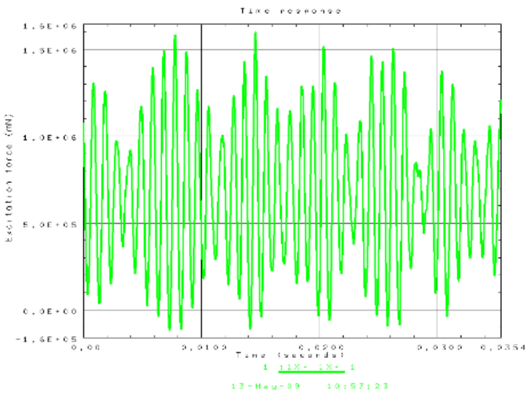


Fig.16. Assembly with dynamometer FEM model



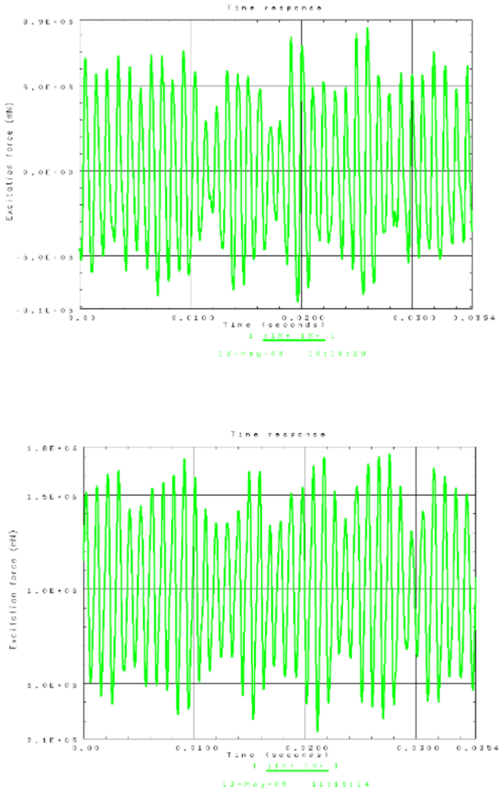


Fig.17. Measured forces $F_x = F_x(t)$, $F_y = F_y(t)$, $F_z = F_z(t)$ during recessing, used as excitation forces in FEM model

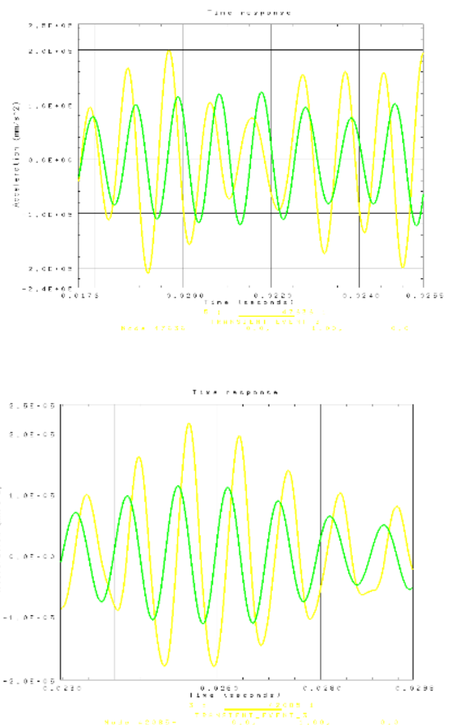


Fig.18. Acceleration magnitudes comparison of the FEM model (yellow) with values measured for excitation by the cutting forces during recessing (green) at the measurement points no. 1 and 2, vertical -y direction

VII. Frequency Dynamic Compliance of Slides FEM Model, Comparison with Experiment

NX I-DEAS works in modal domain during frequency analysis of models with more degrees of freedom [8]. To obtain modal displacements $\gamma(\omega)$, velocities $\dot{\gamma}(\omega)$ and accelerations $\ddot{\gamma}(\omega)$, it applies, in compliance with [9], equations in form

$$\gamma_i(\omega) = \frac{f_\gamma^i(\omega)}{[-\omega^2 m_i + j\omega c_i + k_i]}, \qquad (1)$$

where i indicates i-thmode, f_γ^i i-thmodal force, ω acting frequency, m_i i-thmodal weight, c_i i-th modal damping and k_i i-thmodal stiffness. The individual modal displacement summation gives a total displacement in given direction and the frequency dynamic compliance $\left|\frac{x}{F}\right|$, defined as absolute value of ratio of the total displacement and the acting harmonic excitation force.

The assembly with the tool holder FEM model results were compared with the measurement for frequencies up to 430 Hz. The FEM model with defined acting unit force with variable frequency $F(\omega)$ and measurement point no. 1 is in detail in Fig. 19. The compliance magnitude [mm/N] and phase [deg] of the FEM model is compared with the measurement in Figs. 20 and 21. A relatively good agreement is visible in the plots.

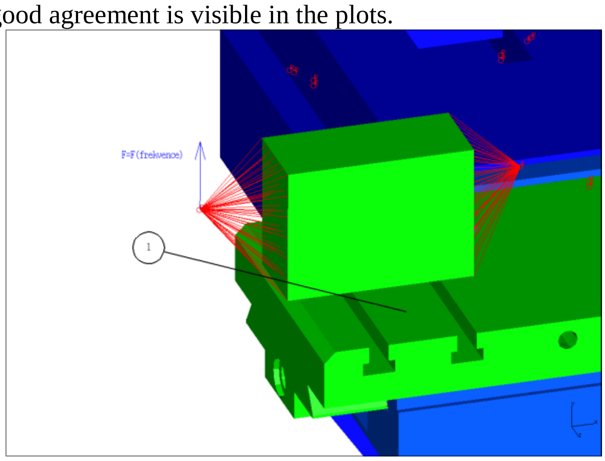
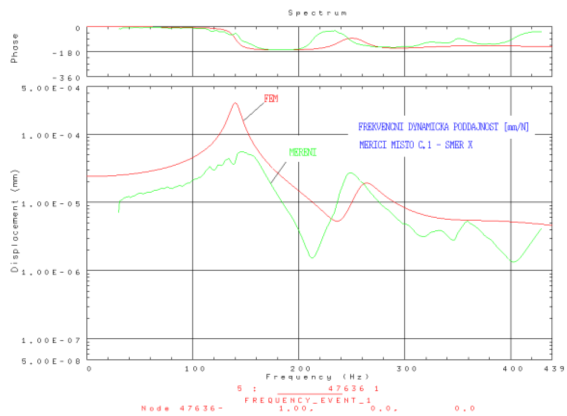


Fig.19.Assembly with tool holderFEM model - detail with defined force $F(\omega)$ and measurement point no. 1 identification



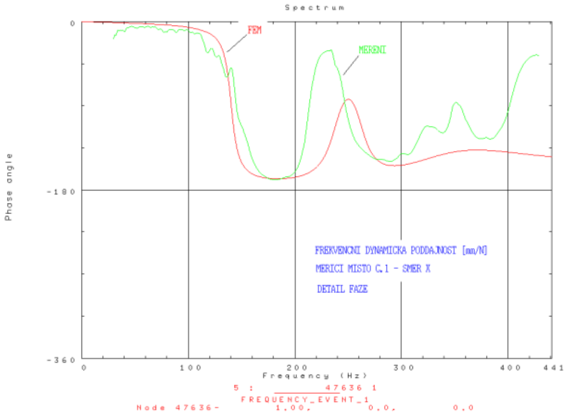


Fig.20. Magnitude [mm/N] and phase [deg] comparison of frequency dynamic compliance of slide FEM model with measurement, at measurement point no. 1 (traverse slide close to the tool holder), horizontalx direction (red = FEM model, green = measurement for shaker excitation)

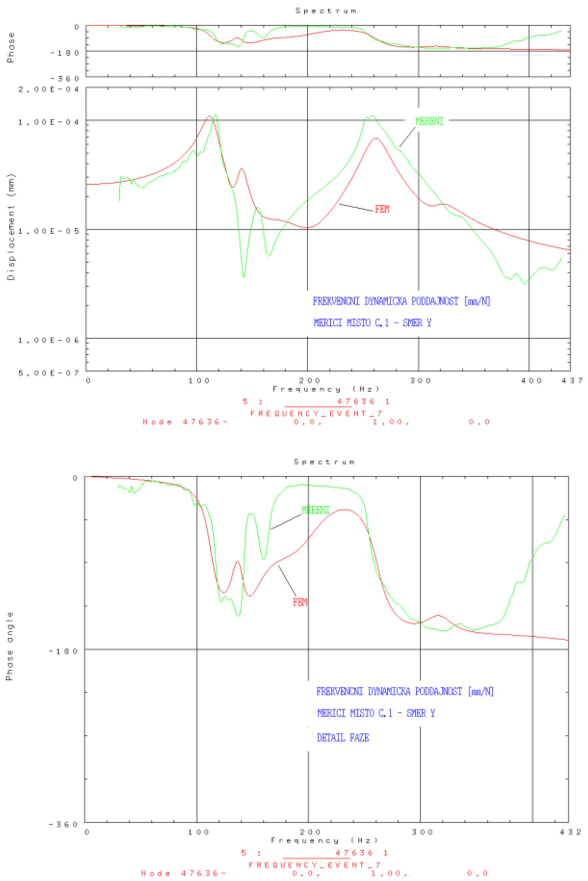


Fig.21. Magnitude [mm/N] and phase [deg] comparison of frequency dynamic compliance of slide FEM model with measurement, at measurement point no. 1 (traverse slide close to the tool holder), vertical y direction (red = FEM model, green = measurement for shaker excitation)

VIII. Conclusion

The rheological model of the machine tool slides with plain-bearing in the dovetail slide is proposed in the paper. Partial FEM models were created for individual assembly bodies. These were joined in the final FEM model using linear springs and dampers. The model respects the elastic assembly mounts to the machine and the material damping in the slides.

The model was verified sequentially in three steps. The results from the NX I-DEAS simulations were compared with the experiments on the real machine, specifically with the static stiffness, the experimental modal analysis and with forced vibrations excited either by a shaker or cutting forces during machining.

Good agreement was proved for the proposed model in terms of transfer characteristics, specifically the magnitude and phase of the frequency dynamic compliance in the measurement range up to 430 Hz.

Acknowledgment

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