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DYNAMIC ANALYSIS OF CNC MILLING MACHINE FRAME

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Abstract: *The aim of this paper is to discuss specific dynamic properties of the main frame of a particular CNC milling machine. Key issues of the preparation of a computational model using the process of FE model including assessment of the loading forces, as well as the results of modal and harmonic response analyses are presented here. Recommendations for setting up the operating conditions are in included in the summary.*

Keywords: CNC, Milling machine frame, dynamic analysis, modal analysis, harmonic analysis.

1. Introduction

Even though there is a continuous development of new technologies for the production of components, it can be stated that 'the machine tool has been, is and no doubt will be, a key factor in industrial and equipment evolution, and as such, improving man's quality of life' (López De Lacalle & Lamikiz, 2009). Total engineering design of a CNC milling machining centre is a complex process and calculation, which should correspond to reality, and can be considered as art (Marek et al., 2014). Modern computational software and sufficient computer power mean that a computational model can be created that approaches reality. FEM models are most often used for these static and dynamic analyses, created based on pre-built CAD models.

2. Analysis of CNC milling machine frame

2.1. CAD and FEM model

From the complete CAD model of CNC milling machine (see Fig. 1), all major structural and connecting parts were used in the preparation of the computational FE model (see Fig. 1).

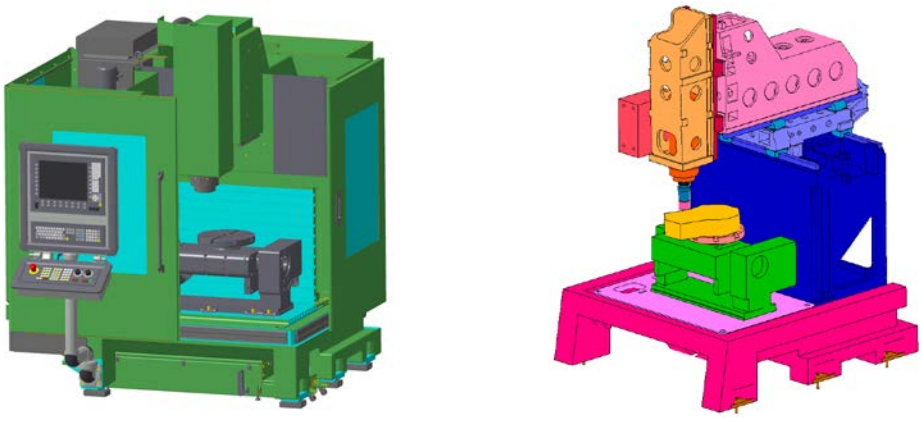

Fig. 1: Complete CAD model (left), simplified FE model for numerical calculation (right)

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Because of the complexity of the frame’s structural design, a FE model was created by combining structured and unstructured mesh (elaborately shaped castings with large changes of thickness were meshed by unstructured elements; rotationally symmetrical parts, plates and linear guides were meshed by structured elements).

Mutual connections of neighboring parts have been accomplished with merging properly selected nodes and by introducing rigid connection (e.g. in case of substitution of screw mountings, see Fig. 2) or by incorporating of linear springs between corresponding pairs of nodes (this was the case of motion screws and the connections between the carriages and the linear guide rails, see Fig. 2).

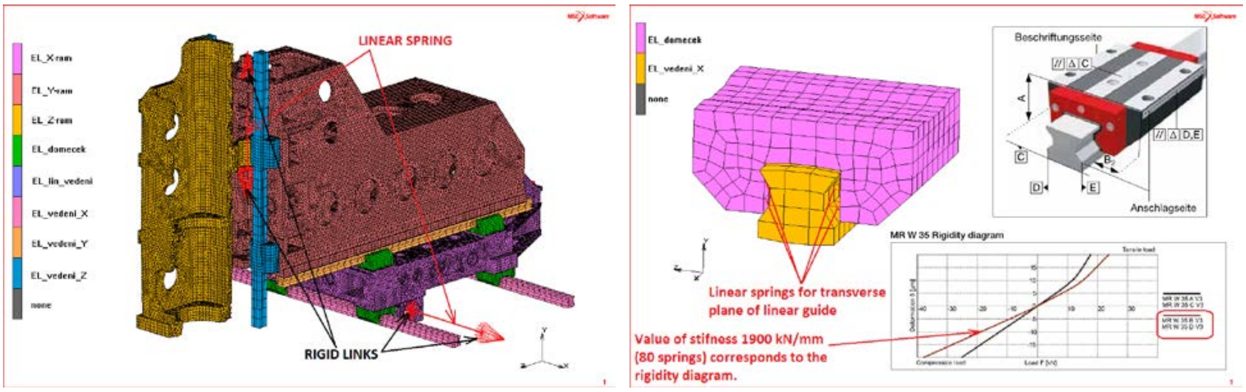

Fig. 2: Replacements of ball screws (right) and definition of linear guide (left)

An overview of the operating parameters and load parameters for tool load according to recommended cutting conditions for steel HB 90-200 are listed in Tab.1. Based on the analysis of the geometry and operating conditions of the load condition (see Tab. 1) it is apparent, that during one revolution of the tool the number of teeth in engagement varies from 2 to 3. From these parameters it case be deduced that is necessary to focus on eigenfrequencies in the range 130-200Hz.

Tab. 1: Operating parameters and load parameters for the tool

Description	***Value***	**Description**	***Value***
Tool diameter *D [mm]*	63	Cutting resistance of steel k_c *[N.mm-2]*	*1700*
Number of cutting edges *[-]*	6	Spindle power P_c *[kW]*	*8.5*
Cutting rate v_c *[m.min-1]*	330	Spindle torque M_L *[N.m]*	*48.7*
Cutting width a_e *[mm]*	50		
Cutting depth a_p *[mm]*	4	Focused circumferential force F_O *[N]*	*1546*
Spindle speeds *n [min-1]*	1667	Axial force F_A *[N]*	*928*
Feed per rev f_n *[mm]*	0.90	Radial force F_R *[N]*	*1237*

At the first step, modal analysis was conducted in order to assess dynamic properties of the frame (see Fig. 4 and 5). Subsequently, several harmonic response analyses were prepared with amplitudes of the loading forces specified according to operating conditions of individual machine tools and the resulting displacements were evaluated in three specific nodes (see Fig.3). Frequency response diagrams are shown in Fig. 6, deformed shapes of the harmonically excited frames in Fig. 7 and 8.

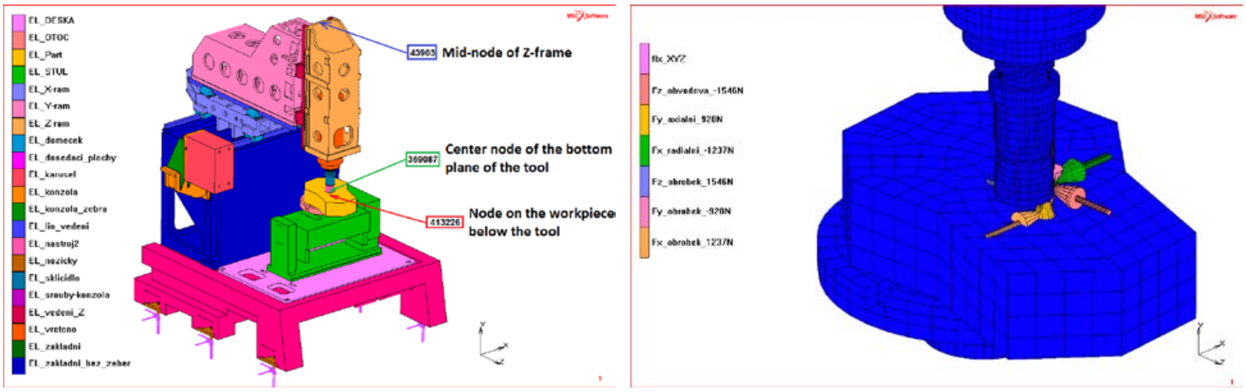

Fig. 3: Location of the nodes (left), Position of load conditions (right)

2.2. Modal analysis results

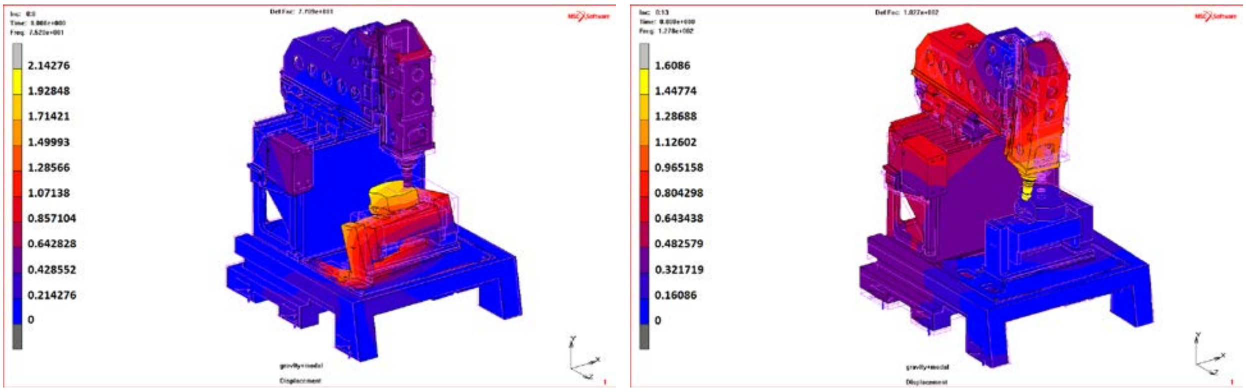

Fig. 4: Displacement of the frame for eigenfrequency: n.8, 75.2Hz (left) and n.13, 127,8Hz (right)

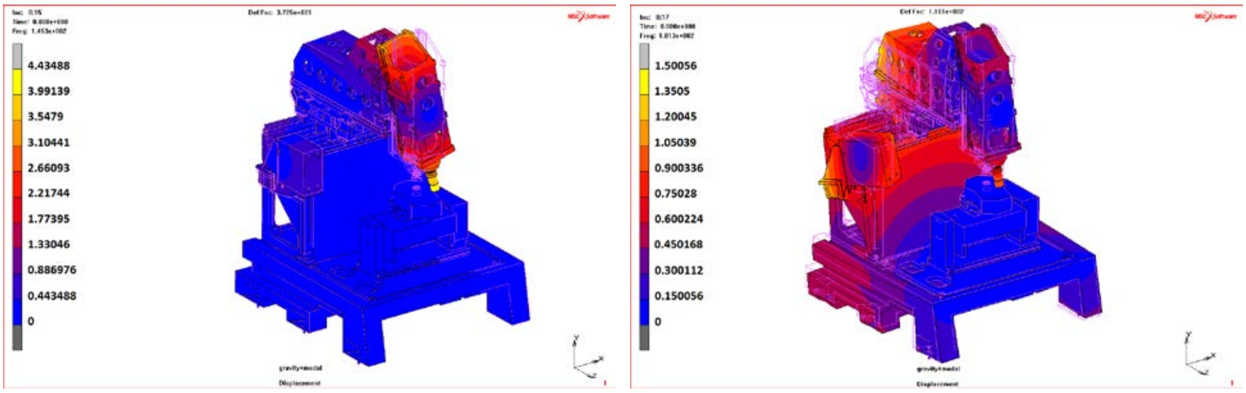

Fig. 5: Displacement of the frame for eigenfrequency: n.15 145.3Hz (left); n.17, 181,3Hz (right)

2.3. Harmonic analysis results

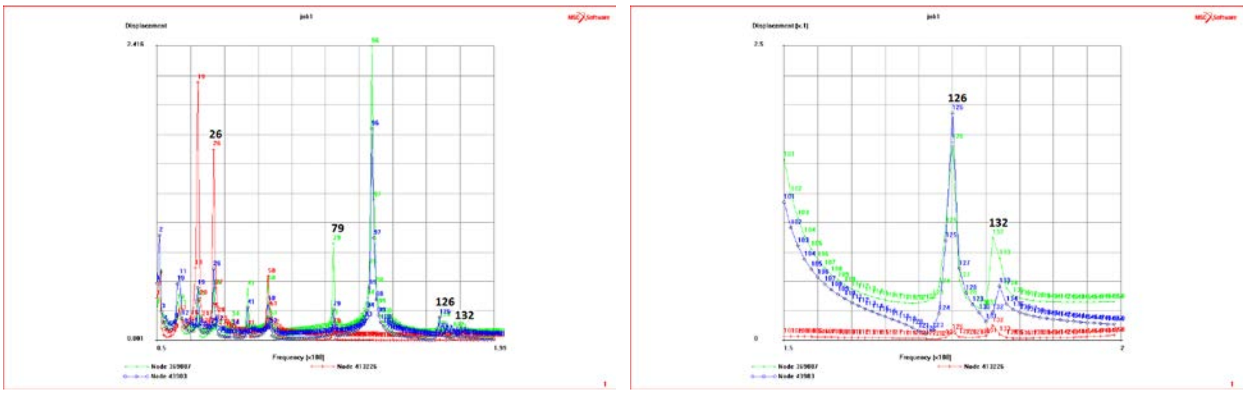

Fig. 6: Diagram of dependence of displacement for selected nodes and the excitation frequency: the entire range (left) and frequency range 150 Hz - 200 Hz (right)

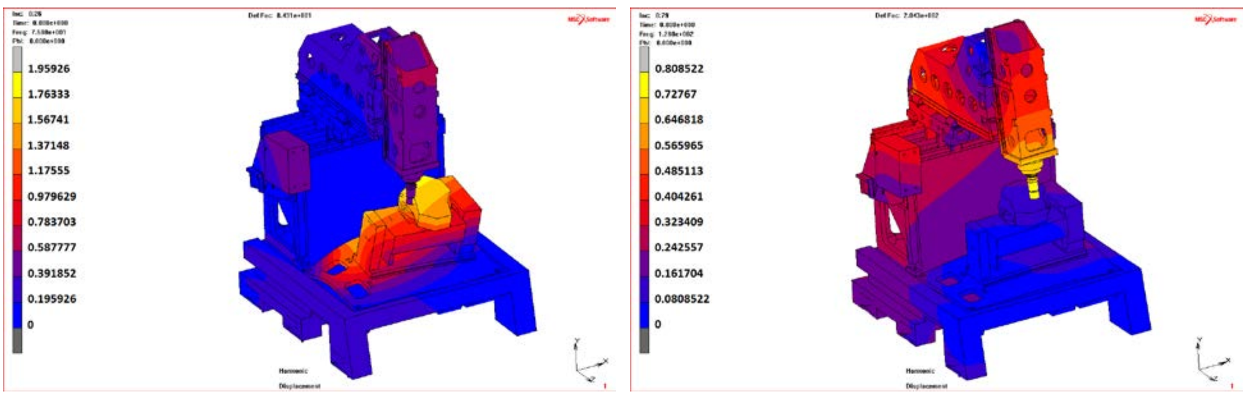

Fig. 7: Displacement caused by harmonic excitation: f=75Hz peak 26 (left); f=128Hz peak 79 (right)

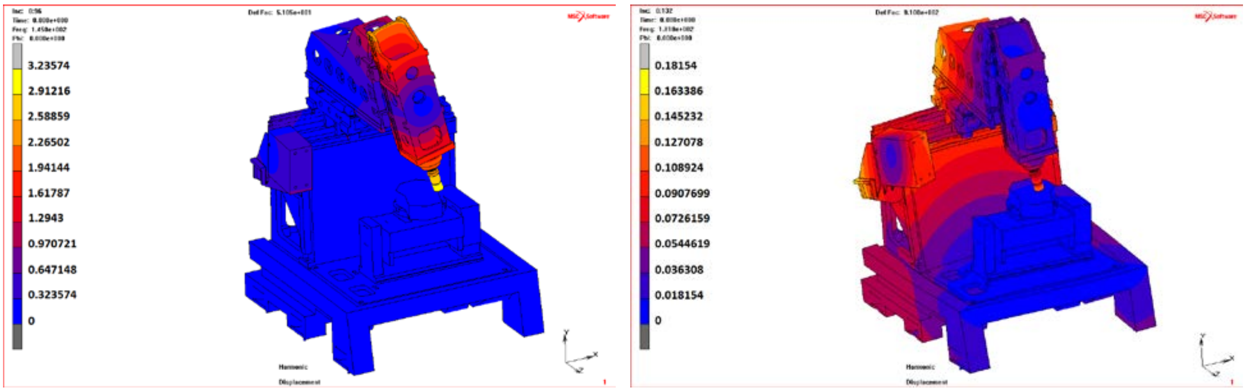

Fig. 8: Displacement caused by harmonic excitation: f=145Hz peak 96 (right); f=181Hz peak 132 (left)

2.4. Summary of results

Modal analysis showed that the eigenmodes occurring in the range of 50 - 200 Hz are reflected in the vibrations of the carousel with a workpiece (see Fig.4) and also in the mutual tilting and rotation of the individual parts of the frame (see Fig.4). Therefore, harmonic analysis was also performed. Comparison of peak frequencies from harmonic response analyses and eigenfrequencies from modal analysis, as well as the displacement amplitudes in specific nodes are shown in the Tab. 2.

Tab. 2: Summary of displacements of selected nodes

Excitation frequency [Hz]	*Displacement (harmonic analysis) [mm]*			*Closest eigenfrequency [Hz]*
	node 369087	*node 413226*	*node 43903*	
75	*0.432*	*1.571*	*0.590*	*75.2*
128	*0.798*	*0.119*	*0.257*	*127.8*
145	*2.416*	*0.011*	*1.741*	*145.3*
181	*0.088*	*0.013*	*0.024*	*184.3*

Based on the results it can be concluded that operating conditions for the tool specified in Tab.1 are not suitable for this particular frame structure of milling machine. It is advisable to choose tools with a higher number of teeth or increase the spindle speed within the recommended cutting conditions at least by 35%. This remedy should shift the main excitation frequencies above 200 Hz, i.e. to the range where eigenmodes do not include a resonance of main parts of the frame.

3. Conclusion

Even though there is a satisfactory agreement between the results of the modal analysis and the harmonic response analysis one should realize that without experimental verification the results of numerical simulations can hardly ever be taken for granted. The major source of the error in the application presented in this paper is probably the specification of the stiffness of the rail guides and the simplified representation of the ball screws and bearings. All contact conditions had to be substituted with either suitable linear springs or by simple node merging - without these simplifications both modal and harmonic response analyses could not have been conducted as they are both linear analyses.

Acknowledgement

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