DETERMINING THE DYNAMIC PROPERTIES OF A CUTTING MACHINE TOOL RAM

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Abstract

This work aims to determine the main dynamic properties of a cast ram in order to create correct modal numerical simulation in FEM software. The dynamic stability of machine tools is directly dependent on natural frequencies and their waveforms and it has large impact on the accuracy of machine tools during the cutting process. Therefore, determining main dynamic properties (vibrations – natural frequencies and their waveforms) is very important. The numerical simulation was done in Siemens Simcenter software with NX Nastran solver. Simulation was carried out as a free state solution. Experimental measurement was done to verify the calculated data. Experimental measurement was performed using the impulse excitation method with a modal sledge hammer. As well as the modal sledge hammer, we used a triaxial accelerometer and a generator module. Subsequently, the experimental modal analysis was done in PULSE19 software. Comparison of results was performed after measuring. Certain agreement was found between the experimentally measured data and data from numerical simulation.

Keywords: casting of ram; FEM; experimental modal analysis; impulse excitation method.

1. Introduction

Not only static forces but also dynamic forces can be applied to all structures, systems (machines). Since the effects of static forces are relatively well mapped nowadays, research into the dynamic properties of structures comes to the fore. Speaking of dynamic properties, it is mainly the natural frequencies and mode shapes of the system [1].

To determine these properties, modal analysis [11] can be used in FEM [12] software. This can be used during design of machine as well as for later validation of dynamic properties. Other possibilities for determination of natural frequencies and mode shapes is experimental modal analysis (modal testing), this can be used only at existing structure.

The aim of this work is create suitable numerical model based on real cast ram (its dynamic properties). The ram is one of the main parts (the most stressed as well) of top-gantry milling machine.

2. Theory

2.1 Modal analysis

Modal analysis is the process of determining dynamic characteristics of a system in forms of natural frequencies, damping factors and mode shapes, and using them to create a mathematical model (FEM). The formulated mathematical model is known as the modal model of the system and its modal data are dynamic characteristics [1], [2].

Second order linear inhomogeneous differential equation (1) with constant coefficients $\mathbf{M}$, $\mathbf{B}$, $\mathbf{K}$ represents mathematical model of linear time invariant system with one degree of freedom. Where $\mathbf{M}$ is total mass matrix, $\mathbf{B}$ is total damping matrix, $\mathbf{K}$ is total stiffness matrix, $\mathbf{U}$ is generalized coordinate (deflection from static equilibrium position), $\mathbf{F}$ is generalized excitation force [3], [4].

$$\mathbf{M} \cdot \ddot{\mathbf{U}}(t) + \mathbf{B} \cdot \dot{\mathbf{U}}(t) + \mathbf{K} \cdot \mathbf{U}(t) = \mathbf{F}(t) \quad (1)$$

The FEM of free undamped oscillation (modal analysis simulation) is based on equation (1) where $\mathbf{F}(t)=0$ (without excitation force) and $\mathbf{B}=0$ (without damping force) – see equation (2) [4].

$$\mathbf{M} \cdot \ddot{\mathbf{U}}(t) + \mathbf{K} \cdot \mathbf{U}(t) = 0 \quad (2)$$

When a harmonic oscillation $\mathbf{U} = \bar{\mathbf{U}}e^{i\Omega t}$ is substituted into equation (2) we get equation (3). Where $\bar{\mathbf{U}}$ is harmonic vibration amplitude column matrix and Ω is angular frequency of oscillation [3].

$$(\mathbf{K} - \Omega^2 \mathbf{M}) \cdot \bar{\mathbf{U}} = 0 \quad (3)$$

The natural frequencies result from the condition of zero determinants (4) [3].

$$\det|\mathbf{K} - \Omega^2 \mathbf{M}| = 0 \quad (4)$$

The values of natural frequencies in FEM are usually given in the form of frequency f [Hz] (5) [3].

$$f = \Omega/2\pi \quad (5)$$

It is not necessary to determine all natural frequencies and mode shapes from equation (3). A small amount of the lowest eigenvalues is sufficient, because there is the highest risk of collision with one of the excitation frequencies of the external load [5].

2.2 Modal testing

Modal testing is a process on a tested structure to obtain a mathematical description of the dynamic behaviour of the structure. FRF (Frequency response function) is created during measuring. FRF is basic of derived model (6).

$$H(\omega) = \frac{\text{output}}{\text{input}} = \frac{\text{movemnet}}{\text{force}} = \frac{\text{response}}{\text{excitation}} \quad (6)$$

There are 3 basic types of frequencies derived functions according to the derived parameter: deflection, speed, acceleration.

An element $\alpha_{jk}(\omega)$ represents a harmonic response at x_j caused by a single harmonic force acting at another place (F_k). λ_r is eigenvalue of the r-mode, Φ_j^r is j-element of custom shape r-vector $\{\Phi\}$ (relative displacement at j-point when oscillating on r-form), ω is the excitation angular frequency and N is number of modes [6], [7].

$$\alpha_{jk}(\omega) = \frac{x_j}{F_k} = \sum_{r=1}^N \frac{\Phi_j^r \cdot \Phi_k^r}{\lambda_r^2 - \omega^2} \quad (7)$$

Equation (7) is basic of experimental modal analysis (modal testing) and shows direct connection between modal properties and response characteristic of the system. This equation confirms that it does not matter if the structure is excited at one point and the response is measured at all points (one column of the FRF matrix is measured) or the structure is excited at all points and the response is measured at one point (one row of the FRF is measured). In both cases after the composition we get the same FRF matrix [6], [7].

3. Modal analysis

First, was created idealise part (without screw holes, radiuses etc.) of ram. for numerical simulation. Then solution was set with parameters are given in the table 1. Idealise model was meshed with elements CTETRA (see Table 1 – FEM settings).

FEM settings		Solution settings	
Element Type:	CTETRA (10)	Solver:	NX Nastran
Element Size:	40mm	Analysis Type:	Structural
Number of elements:	86 258	Solution Type:	103 Real Eigenvalues
Material:	Iron Cast G40	Eigenvalue Method:	Lanczos

Table 1. Modal Analysis simulation parameters

Simulation was performed in NX Nastran solver with restriction 1 - 1600 Hz (max 40 modes). Boundary conditions represent free state, it mean there are no boundary conditions (Fig. 1). This is commonly known procedure how to get natural frequencies and mode shapes of free state structure [8].

Fig. 1. FEM model (left), 1st mode shape (right)

The calculation was terminated earlier than the 1600Hz limit because 40 modes were previously found. The highest detected frequency is 1158.380 Hz within the specified range. This value is sufficiently high because the excitation frequencies for the ram are assumed to be low. The results of the modal analysis are shown in Table 2.

Mode	Frequency [Hz]
1.	247.243
2.	256.382
3.	280.809
4.	287.823
5.	315.171
6.	394.501
7.	429.799
8.	465.812
9.	479.238
10.	497.127
...	...
40.	1158.380

Table 2. Modal analysis simulation results

Deformation of wider site of the ram represents mode shapes of frequency 247 Hz (1st mode) and frequency 288 Hz (4th mode). The bending mode shapes are evident at frequency 256 Hz (2nd mode) and frequency 281 Hz (3rd mode). Nearly clear torsion is displayed at frequency 315 Hz (5th mode). It is not possible to assign a shape of deformation (bending, torsion) that would correspond to the whole structure of the ram because of the dimension expansion at one end of the ram.

4. Experimental modal analysis (modal testing)

Measurement was done at a cast ram, which hanged on ropes at a crane Fig. 2. Those boundaries were set to get as close as possible to free state (to avoid interference from adjacent structures) [9]. Experimental measurement was performed using modal sledge hammer TYPE 8208-003, triaxial accelerometer TYPE 4524-B, Generator module TYPE 3160 + battery and signal processing was made in PULSE 19 program.

Fig. 2. The cast ram hanging on ropes at crane

Fig. 3. Used Equipment: Modal Sledge Hammer, Triaxial Accelerometer, Generator Module + Battery (from right to left) [10]

A sensor (triaxial accelerometer) was placed at one point and an excitation was done at 63 points at the ram. Impact with modal sledge hammer into each point was repeated three times to eliminate mistakes of measurement. Response values were written in to the nodes of pre-prepared wireframe model (point model in PULSE 19 program). The basic measurement parameters are summarized in the Table 3.

Measurement settings	
Impact spots	63
Count of sensors	1
Type of sledge hammer head	Hard (black)
Boundary	Hung on rope ("free-free state")
Software	PULSE 19
Number of repetitions	3
Restriction	0 – 1600 Hz
Step size	1 Hz

Table 3. Basic conditions of measurement

The frequencies with pronounced amplitude at most measuring points were selected from the measured data. The first ten most significant frequencies are stated in the

Table 4. The results were transferred to the NX software for better graphical representation and direct comparison of measured and calculated values.

Mode	Frequency [Hz]
1.	238
2.	251
3.	314
4.	367
5.	401
6.	435
7.	476
8.	538
9.	673
10.	813

Table 4. Modal test results (1st ten most significant frequencies)

5. Comparison and discussion

Frequencies starts to more significantly differ from third mode (see Table 2 and Table 4). It is caused by that FEM modal analysis simulation shows every natural frequency without respecting intensity of oscillation (amplitude size). While the experimental measurement shows characteristically natural frequencies that appear in response to given excitation (impact with modal sledge hammer). Is necessary deeper research of mode shapes for better comparison between frequencies from experimental measuring and modal analysis simulation. In the Table 5 is five chosen frequencies which visually correspond to each other by mode shape. Respectively, first five natural frequencies which were found in experimental measurement were compared with frequencies from numerical simulation according to their mode shapes.

Modal test results			Modal analysis results	
Mode	Frequency [Hz]	Percent deviation [%]	Mode	Frequency [Hz]
1.	238	3.78	1.	247
2.	251	1.99	2.	256
3.	314	0.03	5.	315
4.	367	7.63	6.	395
5.	401	7.23	7.	430

Table 5. Comparison of values (Modal test/ Modal analysis)

Comparison of modal test mode shapes a modal analysis simulation mode shapes below:

Table 6. Visual comparison of mode shapes

6. Conclusion

We were dealing with problem of determine dynamic properties of cast ram and the accuracy of the numerical model compared to the actual cast ram. In this work a FEM modal analysis simulation was performed on the cast ram of top-gantry milling machine as well as modal test (experimental modal analysis). Strong agreement was found between the experimentally measured data and data from the numerical simulation. Deviation does not exceed 8%. It follows that numerical model can be considered correct. If the results were not compared, the error in the structure of the FEM model could be overlooked, or the defect or difference of the existing machine part could be overlooked.

The next step is correlate results based on mode shape to increase accuracy of compare results from modal analysis and modal test. Then the obtained results can be validated and FEM model could be modified according to measured data. If a good match is found and the model can be modified according to the existing part of the machine, then it gives more accurate data for further calculations (for static and dynamic properties of the whole machine).

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