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DYNAMICAL SIMULATION OF A CNC TURNING CENTER

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Abstract: This paper deals with the analysis of a CNC turning centre, more specifically the spindle system. The FEM analysis shows the natural frequencies of the spindle system and the same system was analysed by using the TMM and the influence coefficients method. These analyses pave the way to compare these three methods.

Keywords: *dynamical analysis, turning center, Transfer Matrix Method*

1. INTRODUCTION

Having accurate and productive machine-tools is an essential need. It is very important to be able to dynamically simulate these machines so that accuracy and productivity goals are achieved [1]. In machine-tools, the machines' vibration is a serious problem to be monitored and analysed in order for them to be reduced through mechanical solutions and control systems [2]. The logical approach of doing so is by means of 3D modelling through the help of CAD packages and by FEM modal analysis using special programs (ANSYS, MADYN, etc.). Afterwards, the results are to be verified via mathematical models [3]. For a CNC turning center, it is very important to analyze and simulate the main spindle assembly so the dynamical behavior of the machine-tool is understood from the point of view of mechanical vibrations [4].

2. CNC TURNING CENTRES

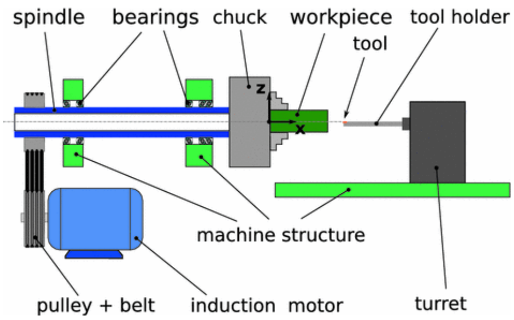

Figure 1
An outline of a metal lathe [6]

As any rotary machine, a CNC turning centre experiences mechanical vibrations for many reasons *(Figure 1)*. For example, the inertia of any moving part in the machine or the unbalance of the rotating parts may cause vibrations since having them perfectly balanced is almost impossible. Another source of vibration in such a machine could be the characteristic natural vibration frequencies (eigenvalues) of the structure and supports of the machine itself [5].

3. The case study and the analysis model

The study and analysis model used is a model of a Haas ST-20T CNC turning centre. Some simplifications should be done so the mathematical model can be constructed. Before starting the simplification procedure, *Figure 2* shows the non-simplified model which was extracted from the machine 3D model *(Figure 2)*.

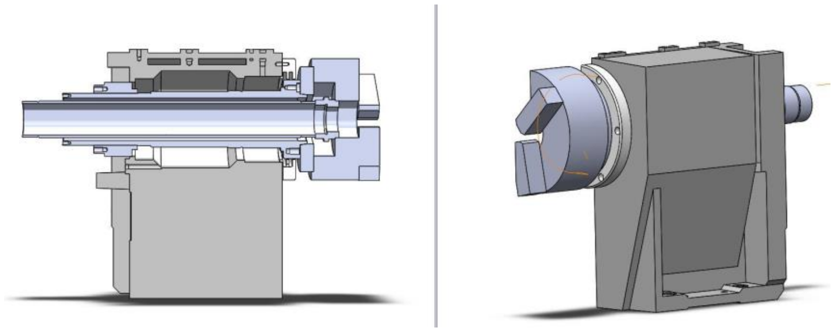

Figure 2
The CAD model of the spindle

4. Results

The previously mentioned turning centre was analysed by three methods: the TMM, the influence coefficients method and the FEM. The results of these three methods will be presented in this chapter so the comparison between them can be performed.

4.1. The TMM

The goal of this analysis is to obtain the values of the natural frequencies of the spindle system. This can be achieved by using the TMM to extract and solve the frequency equation. The system consists of two shaft segments. The overall system transfer matrix relates the state vector at the far left to the state vector at the far right of the shaft. Below is the state vector of station 2 as an example of the matrix form.

$$ {}_{L}\begin{Bmatrix} \boldsymbol{y} \\ \boldsymbol{\varphi} \\ \boldsymbol{M} \\ \boldsymbol{S} \end{Bmatrix}_{2} = \begin{bmatrix} \left(\boldsymbol{l\bar{u}_{21}} + \frac{1}{2}\boldsymbol{\alpha\bar{u}_{31}} + \frac{1}{6}\boldsymbol{\alpha l^2 \bar{u}_{41}}\right) & \boldsymbol{0} & \boldsymbol{0} & \frac{1}{6}\boldsymbol{\alpha l^2} \\ \left(\boldsymbol{\bar{u}_{21}} + \boldsymbol{\alpha\bar{u}_{31}} + \frac{1}{2}\boldsymbol{\alpha l\bar{u}_{41}}\right) & \boldsymbol{0} & \boldsymbol{0} & \frac{1}{2}\boldsymbol{\alpha l} \\ (\boldsymbol{\bar{u}_{31}} + \boldsymbol{l\bar{u}_{41}}) & \boldsymbol{0} & \boldsymbol{0} & \boldsymbol{l} \\ \boldsymbol{\bar{u}_{41}} & \boldsymbol{0} & \boldsymbol{0} & \boldsymbol{1} \end{bmatrix}_{2} \begin{Bmatrix} -\boldsymbol{y_0} \\ \boldsymbol{0} \\ \boldsymbol{0} \\ \boldsymbol{R_a} \end{Bmatrix} \tag{1} $$

Extracting the equations from the matrix form then substituting the factors ($\bar{u}$ and α) values and applying the boundary conditions we can get the frequency equation:

$$mI_d l_1^3(3l_1 + 4l_2)\omega_{nf}^4 - 6EI(2ml_1^3 + 6I_d l_1 + 2I_d l_2 + 2ml_1^2 l_2)\omega_{nf}^2 + 36(EI)^2 = 0 \qquad (2)$$

$$\omega_{nf1} = 85946.17\ \frac{rad}{s}\ \ and\ \ \omega_{nf2} = 3850.1\ \frac{rad}{s}$$

4.2. The influence coefficients method

The influence coefficient method deals with rotor dynamics as categorized cases. For each case, the coefficients calculated result in more accurate analysis results. The current case is considered an overhung rotor system and therefore, the frequency values can be obtained via the following equation:

$$\boldsymbol{\Omega = \frac{J_a}{J_p}\omega + \frac{-1+\omega^2\alpha m}{[\beta-(\alpha\beta-\gamma^2)m\omega^2]J_p\omega}} \qquad (3)$$

$$\boldsymbol{\omega_{nf1} = 1577\ \frac{rad}{s}}$$

4.3. FEM analysis

The FEM analysis were carried out ANSYS. The spindle geometry had slight modifications so it became 100% an axisymmetric geometry since having an axisymmetric geometry is the condition for the program to consider the gyroscopic effect *(Figure 3)*. The constraints were defined in the positions of the bearings. The material properties were defined based on having a spindle system made of ASTM A36 steel. The maximum rotation speed was defined based on the manufacturer's data (4,000 rpm). Finally, the model was meshed and the analyses were carried out.

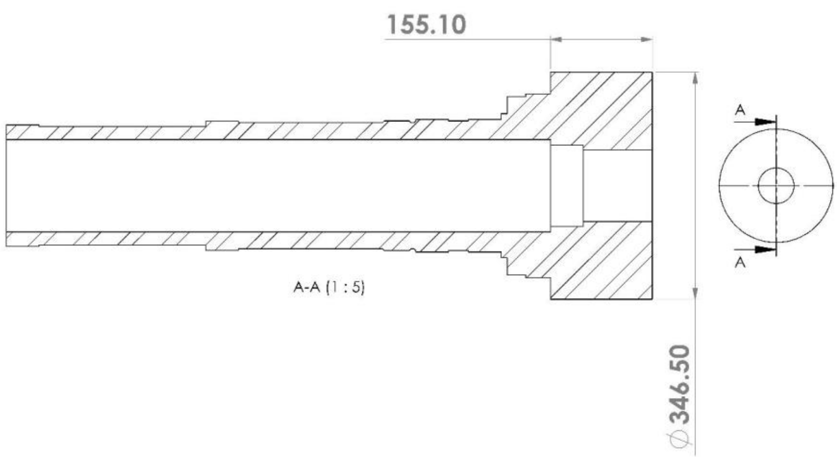

Figure 3
The spindle-chuck model (the analysis model)

For the meshed model, the number of nodes is 12,564 and the number of elements is 7,038 *(Figure 4)*. For the sizing of elements, the adaptive sizing option was used with a minimum edge length of 0.041 mm. The bearings were defined as remote displacements with zero translations in the X, Y and Z directions and free rotations about these directions so they accurately comply with the bearings' role.

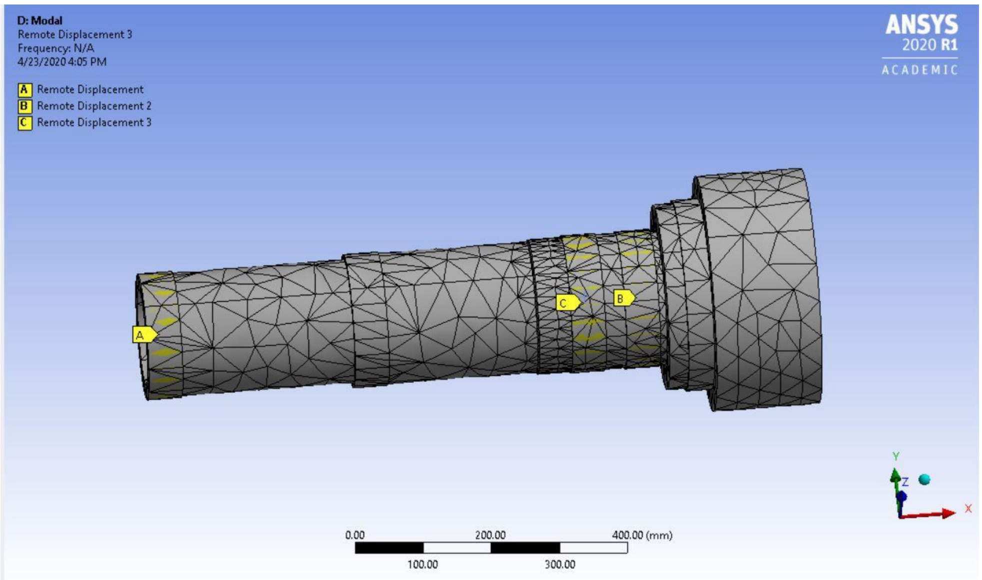

Figure 4
The meshed model with the bearing definition

After selecting the analysis type (modal), importing the geometry, defining the material, defining the geometry fixation conditions, meshing the geometry, activating the rotordynamics properties (Coriolis effect and Campbell diagram), defining the rotational velocity and running the analysis, the following frequency values were obtained (*Table 1* and *Figure 5*).

The results of the FEM analysis are listed in the table below.

Table 1
The values of the frequencies

Mode	**Frequency [Hz]**
1	1.3086e-003
2	611.82
3	612.82
4	1262.6
5	1262.9

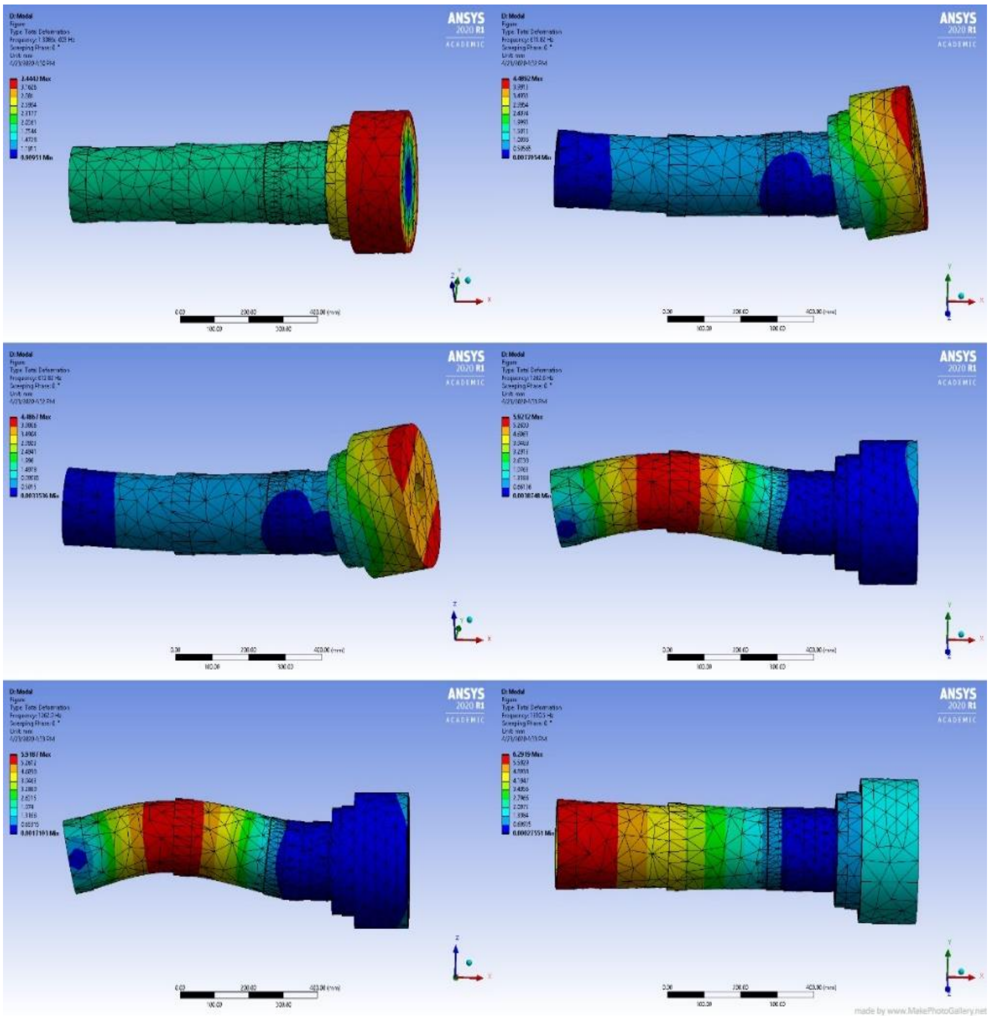

Figure 5
Modes of spindle system deformation

5. CONCLUSION

The TMM analyses were based on the assumptions of having a massless shaft and a lumped-mass whenever having a disk. These assumptions clearly affected the results. The shaft length-diameter ratio clearly affects the analysis using the TMM; the slenderer the shaft, the more accurate the results. The round-off error accumulates gradually through the process of the calculations due to the matrices' multiplication in the TMM. Finally, the results concluded that the influence coefficients method provided less error than the TMM.

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