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STATIC STRESS PERFORMANCE ANALYSIS USING THE FINITE ELEMENTS METHOD OF THE FUS 25 MILLING MACHINE BEFORE REMANUFACTURING

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Abstract: *The finite element method is a numerical method used for solving the complex issues of engineering. The method comprises of meshing the continuous environment through assemble of finite elements, which interact between them in a finite number of nodes. The interacting forces of the nodes of the model characterize the actions of the interior forces or stresses that are applied upon the contours of the neighboring elements. Remanufacturing is the activity which combines the profitability and the benefits of the sustainable development, thus reducing the landfills and virgin raw material consumption, energy and specialized labor used in production.*

Key words: *finite elements method, remanufacturing, rigidity static, nodes.*

1. INTRODUCTION

Milling is a surface generating process, strongly generating forced vibrations whose frequencies are comparable with the main shaft speed and having relatively high amplitudes. The dynamic aspects occurring within the milling machines can cover the majority of the machines tools.

The purpose of this study, using the finite elements method, is to determine the static rigidity of the elastic structure of the FUS 25 milling machine, before the manufacturing process.

In case the static rigidity of the analyzed structure will be within the admissible limits, the analysis for determining the natural vibrations modes can be continued. Knowing the structure natural frequencies, the maximum speed until the new equipment could operate in precision and safety conditions can be set. For every natural frequency, a speed of the main shaft can be calculated. The static rigidity of the milling machine structure contributes to the determination of some parameters of dynamic nature such as: the equivalent elastic constant and the minimum natural frequency.

The results of the theoretic research concur to obtaining a valid analytic model which is to characterize the future dynamic performance of the remanufactured technological equipment and to offer information that during the remanufacturing projecting stage were less obvious.

2. WORK HYPOTHESES

The processing precision of the remanufactured machine-tools is determined by the rigidity of the tool – machine tool – part system. The components of the machine tool base structure (trunks, upright beams, casings, crossbeams, etc.) have the most important role in this chain. On the other hand, the machine tool structure is formed by the totality of the elements that concur to bear the kinematical chains mechanisms and in which the polygon of the forces, which appear during the working processes, is closed. In case of remanufacturing, the reusable structure elements are in a partially unknown condition, from the point of view of the static and dynamic rigidity.

The effect of the reaction forces and moments on the structure elements as well as of other perturbation factors following the work processes during the previous exploitation cycles must be known before starting the remanufacturing process.

The rigidity and stability to vibrations of the machine-tools base structure elements must be sufficient, as the deformations and the vibrations amplitude level are not to exceed the admitted limits [4].

The study of the structures before remanufacturing allows the verification of the limited conditions hypothesis, according to which the maximum deformation of every element to not exceed an imposed acceptable value, according to the new work regimens in which the remanufactured technologic equipment shall operate. The results obtained following the study of the structure elements static performance allow setup the work regimen limits of the remanufactured machine-tools. The decision regarding the way to perform the remanufacturing of a technological equipement may be grounded on the latter information.

The machine tool static performance is setup by determining the influence of the static loads on the relative positions of its subassemblies, measuring the geometric precision of the equipment loaded with constant load, that's why it is also named static rigidity.

3. ANALYS OF THE STRUCTURAL PERFORMANCE UNDER STATIC LOAD

During the research we took into account that there is not always a direct link between the static precision and dynamic precision of the machine-tools. Under variable stresses, a statically rigid structure element can have dynamically low behavior.

Precision of strains and stresses is significantly dependent on the point location where they are computed. Meshing the structures can be carried out with linear, plane or spatial finite elements. The structures analysis imposes using two systems of reference [2]:

1. One system which is associated to the structure named global system – in which the position is defined and its movements are determined;
2. One system which is associated to each element named local system.

One of the known coordinate systems: Cartesian, cylindrical or spherical, can be chosen as systems of reference.

In nonlinear solid mechanics problems the relationship between applied external influence and resulting displacements is nonlinear. Nonlinearity can be caused by nonlinear material response (physical nonlinearity) or by large displacements and rotations (geometrical nonlinearity).

Consider finite element procedure for the solution of elastic-plastic problems. Strains at any point an element are determined using Cauchy relations [5]: *u*

$$\{\varepsilon\} = \{\varepsilon_x \quad \varepsilon_y \quad \gamma_{xy}\} = \left\{\frac{\partial u}{\partial x} \quad \frac{\partial v}{\partial y} \quad \frac{\partial v}{\partial x} + \frac{\partial u}{\partial y}\right\}. \tag{1}$$

With the use of the displacement differentiation matrix:

$$\{\varepsilon\} = [A] \cdot \{q\}. \tag{2}$$

The matrix $[A]$ for interpolating strains using nodal displacements is equal to:

$$[A] = \frac{1}{2\Delta}\begin{bmatrix} b_1 & 0 & b_2 & 0 & b_3 & 0 \\ 0 & c_1 & 0 & c_2 & 0 & c_3 \\ c_1 & b_1 & c_2 & b_2 & c_3 & b_3 \end{bmatrix}. \tag{3}$$

The elasticity matrix [E] has the following appearance for plane problems:

$$[E] = \begin{bmatrix} \lambda + 2\mu & \lambda & 0 \\ \lambda & \lambda + 2\mu & 0 \\ 0 & 0 & \mu \end{bmatrix}, \tag{4}$$

where λ and μ are Lame constants:

$$\lambda = \frac{\upsilon \cdot E}{(1+\upsilon)(1-2\upsilon)}, \text{ for plane strain;} \tag{5}$$

$$\lambda = \frac{\upsilon \cdot E}{1-\upsilon^2}, \text{ for plan stress;} \tag{6}$$

$$\mu = \frac{E}{2(1+\upsilon)}.$$

Here *E* is the elasticity modulus and *v* is the Poisson's ratio. Stresses are calculated with the Hook's law:

$$\{\sigma\} = [E] \cdot \{\varepsilon^e\} = [E] \cdot (\{e\} - \{e^t\}), \tag{7}$$

where $\{e^t\}$ is the vector of free thermal expansion:

$$\{e^t\} = \{\alpha T \quad \alpha T \quad 0\}. \tag{8}$$

For the structural elements analyses we start from the structure modeling of a universal milling machine for tool workshops FUS 25, with he following technical characteristics:

- revolutions domain: 25...2000 rev/min;
- driving domain: 5...400 mm/min;
- fast drive: 1400 mm/min;
- the weight of the machine with fixed table and vertical head: 1 540 kg.

For the static rigidity analysis of the FUS 25 milling machine, using the finite elements, we started from the individualization of its elastic system by physical modeling. The materialized model reproduces the essential properties of the actual system and it is highly accessible for analysis [3].

Starting from the structure shape and according to the stress which solicits it, the types of elements and the discrete network were setup. More discrete types can be carried out for the same structure. When selecting the optimum discrete variant, we took into account that the precision level of the obtained results increased relative to the finite elements number; but even this figure could not have been of any value, as it would have increased the number of unknown degrees of freedom and consequently it would have increased their estimate time. For an efficient correlation of these two aspects, we used the finite elements discreted structure with different dimensions. Thus, for the areas where the movements and tensions had not high variations, we used high dimension finite elements, and in the areas where the values of the movements and tensions differ much from one element to the other, we used a much thicker network of finite elements.

In order to ensure much closer results to the actual ones, it was necessary to progress slightly from the high dimension finite elements to the small dimension finite elements.

Using CAD software, we obtained the geometry of the structure of the FUS 25 milling machine, after which it was meshing, as it is presented in Fig. 1 [4].

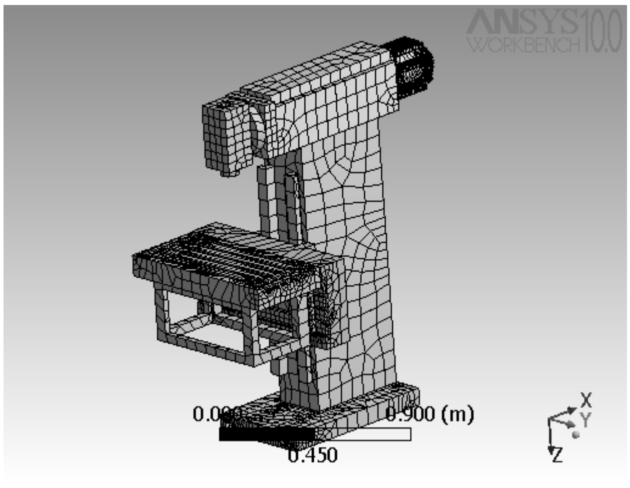

Fig.1. The meshing geometric model.

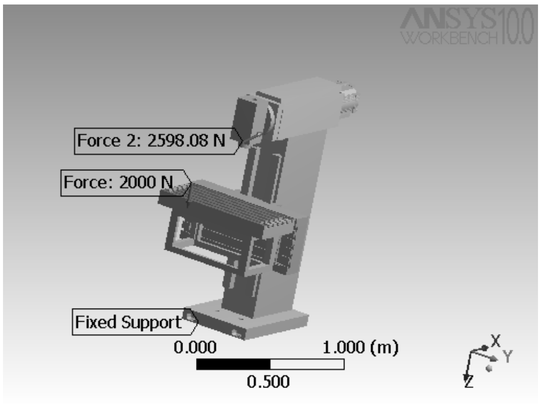

Fig. 2. Mode of fastening and stress.

As it is shown in Fig. 2, the model is prepared for the finite elements analysis, after imposing the stress and support conditions. (the force $F_2 = 2598$ N has its application point on the case of the main shaft, and the force that actions on the machine table has the value of $F =$ 2 000 N). After the analysis has been run by a specialized program, a report that describes these stages and the conclusion which results from the finite elements analysis.

4. RESULTS EXAMINATION

As it is shown in Fig. 3, under the imposed stress, the total deformation of the structure of the milling machine equals 0.169 mm.

Figure 4 shows that the maximum main stress appears in the joining area between the machine trunk and the base plate, and its value is:

$$\sigma_1 = -0.37 \ldots 17.4 \text{ MPa} < \sigma_{\text{adm}} .$$

Figure 5 present the values of the average stress

$$\sigma_2 = -7.2 \ldots 5.6 \text{ MPa} < \sigma_{\text{adm}} .$$

Figure 6 present the values of the minimum stress

$$\sigma_3 = -21.4 \ldots 3.98 \text{ Mpa} < \sigma_{\text{adm}} .$$

Figure 7 presents the deformation of the elastic structure of the milling machine under tearing stress (13.1 MPa).

Figure 8 presents the equivalent stress

$$\sigma_{echiv} = 22.8 \text{ MPa} < \sigma_{\text{admissible}}.$$

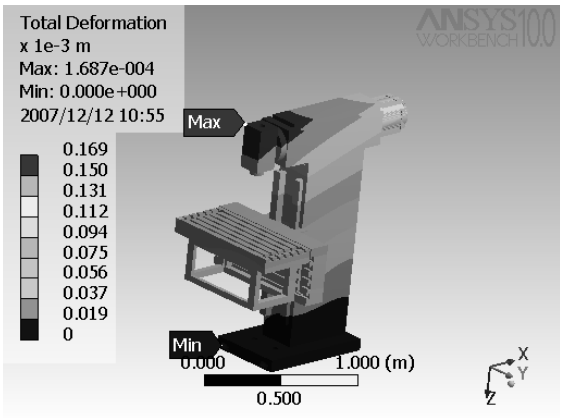

Fig. 3. Total deformation.

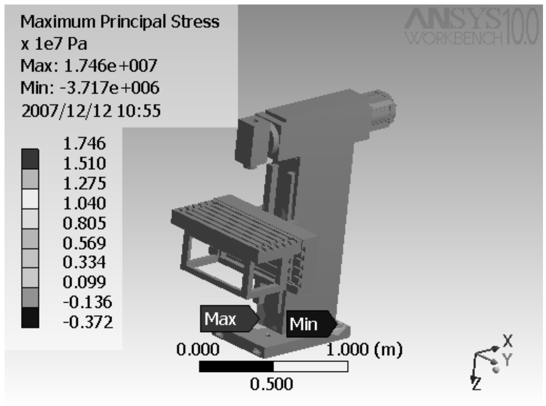

Fig. 4. Maximum Principal Stress.

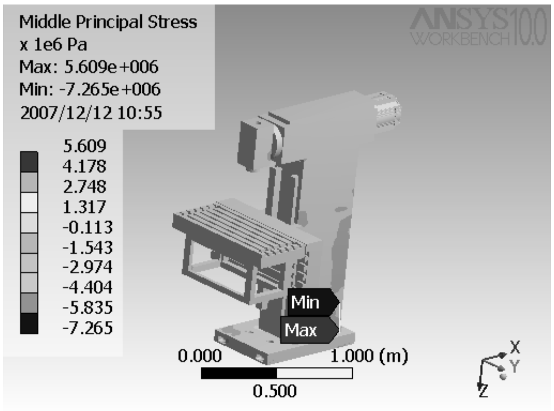

Fig. 5. Middle principal stress.

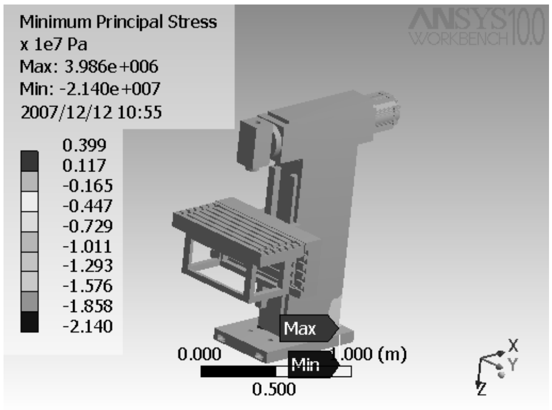

Fig. 6. Minimum principal stress.

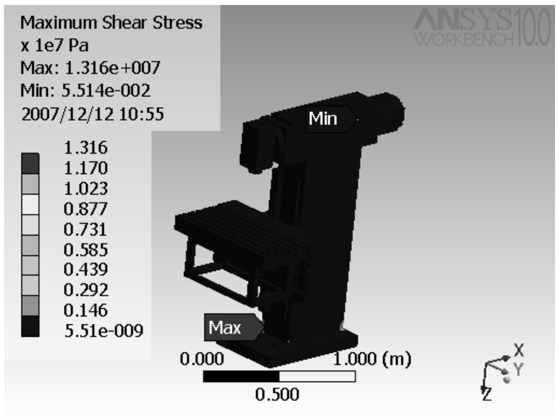

Fig. 7. Maximum shear stress.

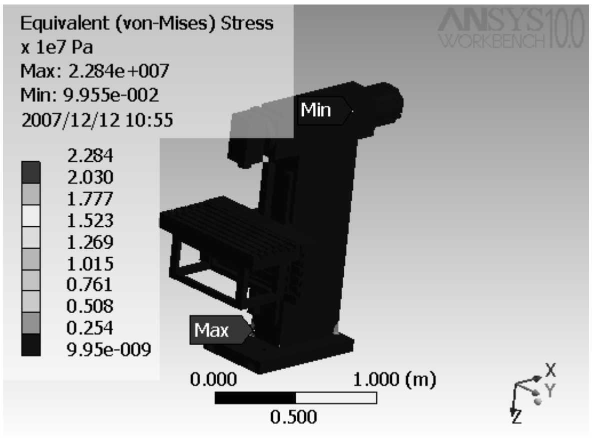

Fig. 8. Equivalent Stress (von-Mises).

From the graphic analysis of deformations, it results that the elastic structure of the milling machine has a static rigidity, which is within admissible limits. In these conditions the following step of the research can begin: the analysis of the elastic structure dynamic performance

5. CONCLUSIONS

As the technological equipment that should be remanufactured have to be brought to the new equipment performance level, it is compulsory that before commencing their disassembling, a static analysis should be fulfilled, for the estimation of the dynamic performance of the new remanufactured equipment.

Referring to the static precision of the milling machine, its behavior was observed from the point of view of the changes of the relative positions of some structure elements under static stress.

The interaction modeling of the structure elements, as component parts of the machine tool, was carried out with some forces, as the force as an abstract expression represents the movement transmission measure, and it can be modeled using vectors.

The static rigidity of the technological equipment to be remanufactured range in the admissible limits, as the equipment proposed for the remanufacturing processes generally have heavy weight structure elements.

The kinematical parameters of the remanufactured machine-tool can be setup, grounded on the results obtained following the studying of the structure elements taken into account for remanufacturing.

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