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STRESS AND DISPLACEMENT ANALYSIS OF A MODERN DESIGN LATHE BODY BY THE FINITE ELEMENT METHOD (FEM)

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The Finite element method (FEM) was used in this study for the analysis of the strain and stress of a turning machine body. The final design decisions were made on the basis of stress and displacement field analysis of various design versions related to the structure of the considered machine tool. The results presented in this paper will be helpful for practical static and dynamic strength evaluation as well as for the appropriate design of machine tools using the FEM.

Key words: lathe body analysis, stress, displacement, FEM

Analize naprezanja i pomaka suvremenog postolja tokarilice metodom konačnih elemenata (MKE). Metoda konačnih elemenata (MKE) je primjenjena za analizu deformacija i naprezanja postolja stroja. Konačna odluka o oblikovanju je donešena na osnovi analize polja pomaka za različite varijante oblikovovanja elemenata razmatranih strojeva. Rezultati predstavljeni u ovom radu će biti korisni za procjenu statičke i dinamičke čvrstoće kao i za odgovarajuće oblikovanje strojnih alata korištenjem MKE.

Ključne riječi: analiza postolja tokarilice, naprezanje, pomak, MKE

INTRODUCTION

The bodies of contemporary machine tools are required to be rigid and efficiently damp vibration. Cast iron is the basic material mostly used in machine tool body construction because of its high stiffness (E = 120÷130 GPa) and damping coefficient (β = 0,0085) [1]. Sand casting is the method for making bodies of cast iron. That technology is energy-consuming, but is well mastered and easy to automate. In machines requiring less accuracy man uses welded bodies. The manufacturing cost is lower for small lot production (lack of costly casting models), so it is necessary to apply stress-relief annealing. Welded bodies have higher stiffness (E = 190÷210 GPa), but lower damping coefficient (β = 0,002) in comparison to cast iron ones. Assuming the same rigidity, a welded body is twice lighter than a body of cast iron. From some time now machine tools manufacturers use mineral casting and composite material in body building. Mineral casting consists of a binding agent (methacrylate, epoxy or polyester resin) and a filler (sandstone, basalt, granite or quartz grit), and is of high damping coefficient (β = 0,02÷0,03) and sufficient mechanical properties (E = 30÷50 GPa). Assuming the same rigidity, the weight of a mineral casting body is similar to that of cast iron. Mineral casting is widely used for main bodies of precision machine tools (primarily lathes and grinders). Composite materials are used for moveable bodies, where particular stiffness and weight minimising is a priority [2-9]. The subject of our consideration is the stress and displacement analysis under static and dynamic load of the modern design CTX ALPHA lathe body made of cast iron. Bearing in mind the complexity of the geometrical shape of the lathe body, the finite element method was used as a basic tool in the analysis. The main aim was to evaluate the possibility of reducing the body weight by means of decreasing internal wall thickness.

DETERMINATION OF EXTERNAL LOADS ACTING ON THE LATHE BODY

The external loads acting on the lathe body are the basic forces needed in stress state analysis and strength evaluation of the considered machine tool set. The main external static loads are cutting forces developed in extreme conditions of cutting processes and the gravity forces of the lathe body and the other lathe sub-sets as headstock, slide and tailstock. Dynamic forces resulting from inertia forces developed during machine slide start up and stop in the direction parallel to slideways were also considered. Since cutting forces and inertia forces of the slide are transmitted to the bed indirectly by the headstock, slide and tailstock, appropriate sets that simulate the machine sub-sets with a quite high rigidity were elaborated, in order to determine the inter-reac-

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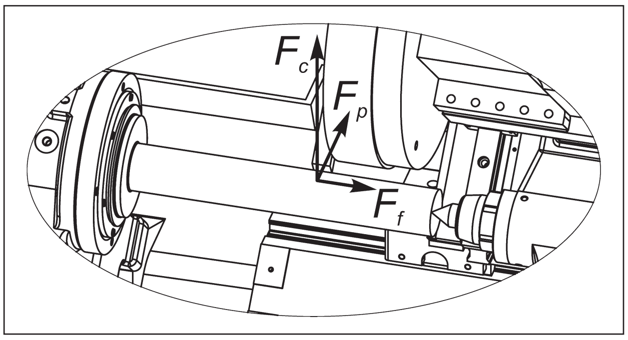

Figure 1 Cutting forces

tions between the sub-sets and the lathe body. The working forces acting on the lathe sub-sets, see Figure 1, were determined for the case of rough turning [10].

Three cutting forces are shown in Figure 1, namely: cutting force F_c, passive (reaction) force F_p and feed force F_f. These forces are the components of the total cutting force determined on the basis of extreme machining conditions during rough cutting of the shaft with a diameter of Φ 65 mm.

The following assumptions were taken in the analysis: *i*) lathe drive power of 15kW with main drive efficiency ratio 0,8 that results in cutting power P_c=15 000×0,8=12 000 W, *ii*) cutting speed 120 m/min (2 m/s) that gives the cutting force F_c=12 000/2=6 000 N, *iii*) passive force F_p and feed force F_f equal 40% of the cutting force, i.e.: F_p = 2400 N, F_f = 2400 N.

The forces acting on the lathe body were determined according to the diagrams shown in Figure 1 and 2. In order to determine the reactions between the slide and lathe bed the slide set was modelled with bar elements having a high rigidity and loaded by cutting forces and their masses. The support reactions acting on the lathe bed are determined within the regions where the load is transmitted from slide to bed at four support points, i.e. at blocks and at the axis of the ball screw. It was assumed that the blocks will transmit the forces along axes *X* and *Y* (according to the presented coordinate system). However, the ball-screw will transmit the force along the *Z* axis.

A similar approach was applied to determine the load coming from headstock and tailstock. Two bar systems were modelled. The first model was loaded by the cutting force and the masses of specific elements, see Figure 3b. The second model, shown in Figure 3c, was loaded by the force coming from the contact between the tailstock and the machined element. The support reactions were determined at the fixing region between the headstock and the body at 4 points of the tailstock support, and along the cylinder driving tailstock. The load acting on the lathe body was the sum of the determined reactions. The load scheme for the lathe bed is presented in Figure 4.

Based on the presented schemes reactive forces and moments were calculated and used for further analyses.

A dynamic force acting on the lathe body was determined for a case when the slide starts up making a dead

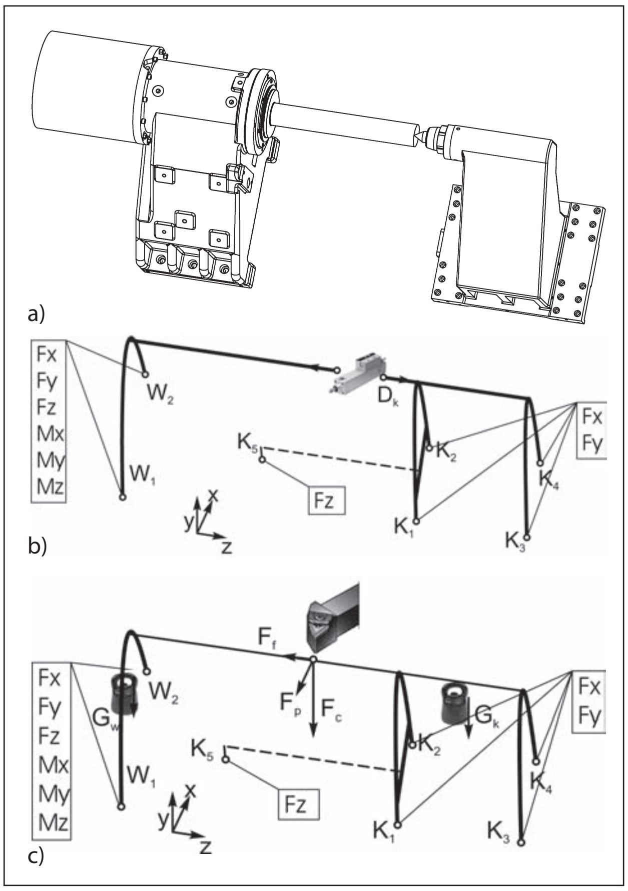

Figure 3 A diagram to determine headstock and tailstock force and moment reactions

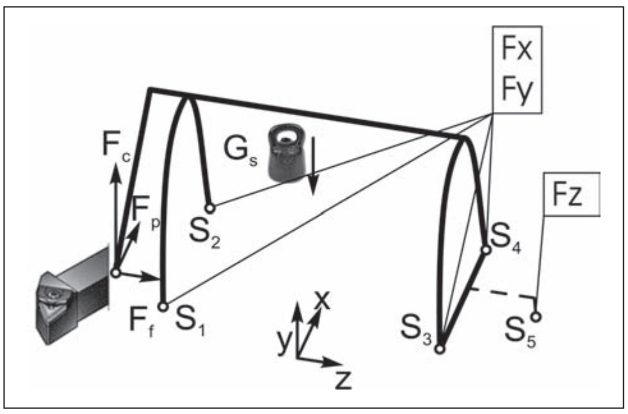

Figure 2 A diagram to determine reaction forces and moments of the slide

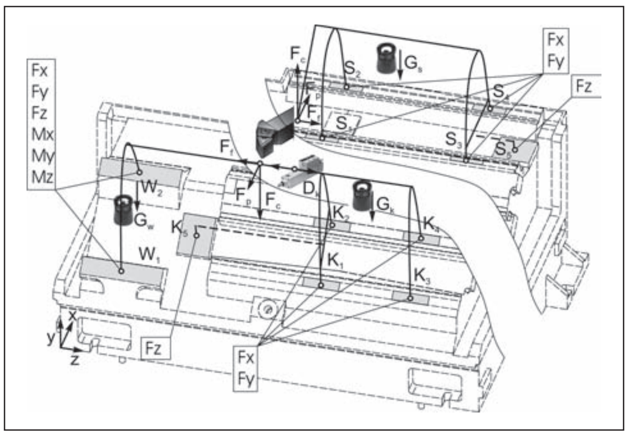

Figure 4 Total loadings acting on the bed

movement. The following data were accepted: *i*) slide stack mass $m = 400$ kg, *ii*) fast movement velocity $v = 30$ m/min, *iii*) acceleration time $t = 0,1$ s.

Acceleration of the slide was determined from the formula: $a = \frac{v}{t} = \frac{30}{60 \cdot 0,1} = 5\text{m/s}.$

Dynamic force acting on the slide is
$F_d = m \cdot a = 400 \cdot 5 = 2\,000\text{N}$.

Following similar steps as we discussed in the case of rough cutting, the slide bar model was loaded by gravity force and the determined dynamic force applied at the gravity centre and acting in an opposite direction to the slide movement. The determined reactive forces acting on the lathe body were used for dynamic analyses.

MATERIAL, BOUNDARY CONDITIONS AND DESIGN OPTIONS

The material of the ALPHA lathe body is cast iron grade (EN-GJL250 according to UNI EN 1 561). The FEM calculations were performed for different cast iron grades: EN-GJL250, 300 and 350. The elastic moduli of the materials are: 104800, 126174 and 131000 MPa.

There were design variations with different thickness of the internal walls: 16 and 18mm and a different number of body supports (legs): 3 and 4.

FINITE ELEMENT MODELLING OF THE LATHE BODY

For the realistic FEM analysis of the considered lathe body we used three dimensional finite elements (solid elements) [12]. We used the tetrahedron finite element with three translational degrees of freedom per node, and hexahedron finite element having eight corner nodes, and three translational degrees of freedom T_x, T_y, T_z per node [11]. For convenience, the element matrices were derived by treating them as an isoparametric element. Once several important mesh generation parameters were introduced, the FEM mesh was generated automatically by the system. The mesh can be also automatically rebuild to obtain a finer mesh subdivision in the regions where stress concentration is developed [12].

EVALUATION OF THE FEM RESULTS

A series of results obtained by the finite element analysis using PRO/E system gave an output in the form of the listings or, what is more convenient in the evaluation, in the form of computer plot of contours for displacements, strains and stresses.

The illustration in Figure 5 presents the computer plot of contours of resulting displacements for the selected variant (3rd) of the lathe body. For a better association of specific design options to the applied external loading, the diagrams also present the locations of these loads. Their values are given in Table 1.

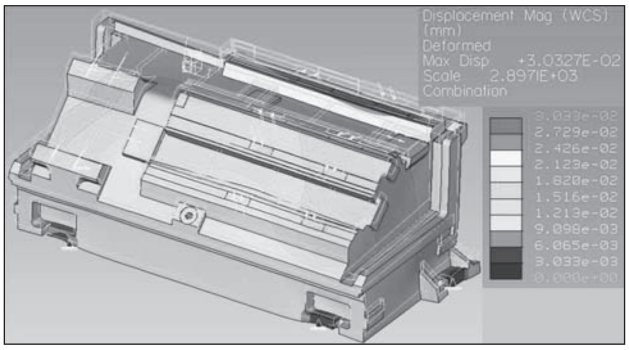

Figure 5 Computer plots of contours of the resulting displacements U [mm] (3rd variant)

Table 1 **List of important FEM results for various versions of numerical calculations**

Item	Version of FEM calculations	Max. effect. stress σ_{ef} / N/mm^2	Max. res. disp. U / mm
1	3 legs, int. walls 18 mm EN-GJL250, cutting forces	28,38	0,046
2	3 legs, int. walls 16 mm EN-GJL250, cutting forces	45,00	0,040
3	4 legs, int. walls 18 mm EN-GJL250, cutting forces	27,22	0,030
4	4 legs, int. walls 16 mm EN-GJL250, cutting forces	42,42	0,048
5	4 legs, int.walls 18 mm EN-GJL300, cutting forces	27,22	0,025
6	4 legs, int. walls 18 mm EN-GJL350, cutting forces	27,84	0,025
7	4 legs, 3rd + 0.1 mm, int. walls 18 mm, EN-GJL250, cutting forces	97,27	0,360
8	4 legs, 3rd - 0.1 mm, int. walls 18 mm, EN-GJL250, cutting forces	111,3	0,405
9	4 legs. 3rd + 0.2 mm, int. walls 18 mm, EN-GJL200, cutting forces	189,40	0,743
10	4 legs. 3rd - 0.2 mm, int.walls 18 mm,EN-GJL250,cutting forces	206,2	0,787
11	4 legs, int. walls 18 mm, EN-GJL250, dynamic load	32,10	0,029

The summarized results of the FEM calculations are presented in table 1. Items 1 - 10 show results of the FEM calculations of the lathe body under static loads caused by cutting forces, and item 11 shows the result of the FEM calculations under dynamic loads.

Variants 3 - 9 of the numerical analysis are related to the general support of the body on four supporting legs. However, we now assume that one of four supports is not located exactly in the plane of the other three supports, but goes out of the plane (the leg is "shorter" or longer") by a gap equal to: $\pm$ 0,1 mm and $\pm$ 0,2 mm. The change in the support (legs) gap has to influence the results because the four leg support of the lathe body is a statically indeterminate system. We can conclude that the increase in the gap value by 0,1 mm results in the increase of maximum effective stress by approx. three times and in the maximum resulting displacements by ten times. However, the increase in the gap value by 0,2 mm results in the increase of maximum effective stress by approx. seven times and in the displacements by twenty times.

CONCLUSIONS

The paper presents the strength analysis by the finite element method and the numerical results of the ALPHA lathe body under static and dynamic loads. The final design decisions were made on the basis of stress and displacement field analysis of various design versions related to the structure of the considered machine tool. The main external static loads include cutting forces and gravity forces resulting from masses of the lathe body and the masses of other sub-sets as headstock, slide and tailstock. The FEM calculations were also performed for the lathe body loaded by dynamical forces resulting from inertial forces developed during slide start up and stop. Since cutting forces and inertia forces of the slide are transmitted into the bed indirectly by the headstock, slide and tailstock, appropriate sets that simulate the machine sub-sets with a quite high rigidity were elaborated, in order to determine the inter-reactions between the sub-sets and the lathe body. The calculations were performed for different versions of body support and internal walls thickness.

The casting of the lathe body has a very complicated shape. So the FEM modelling of the body was made by use of 3D “solid” finite elements with 3 (tetrahedron) and 8 (hexahedron) corner nodes. Three translational degrees of freedom of the accepted 3D finite elements were included in each node of the element. The FEM mesh was generated automatically by the system after several important generation parameters had been accepted.

The results of the FEM numerical analysis for various design versions are presented and discussed in section 5 of this paper. The list of important results is given in Table 1.

The FEM analysis allows to conclude that the maximum effective stress under static load in the lathe body is 45 N/mm^2 if the body is supported on three legs and the internal walls thickness is 16 mm. This effective stress can be recognized as an average stress level within the region of allowable stress for the accepted lathe body material, i.e. cast iron grade EN-GJL250. Maximum resulting displacement is 0,048 mm if the lathe body has internal walls of 16 mm and is supported by four legs. These values are noticeably greater if one of the four supporting legs is given a “gap”. This confirms the expected conclusion that the correct levelling of the machine is required.

REFERENCES

[1] Hajkowski M., Forecasting of mechanical properties of the castings made of hypoeutectic silumins, Archives of Mechanical Technology and Automation, 23(2003)1, 41-52.

[2] Epucret. Mineral casting frames for machine tools (www.rampf-gruppe.de, 1.08.2010).

[3] Non-conventional Materials for Machine Tool Structures, M. Rahman, Md. A. Mansur, Md. B. Karim, JSME International Journal, Series C, 44(2001)1.

[4] Vrtanoski, G., Dukovski, V., Design of polimer concrete main spindle housing for cnc lathe, 13th International Scientific Conference of Achievements in Mechanical and Materials Engineering, 16-19th May 2005, pp. 695-698.

[5] Customers demand ever tighter tolerances, HSM today, Specialist journal from Mikron Nidau for the Milling Technology, Issue 14, 02.2005, p.7.

[6] Vibration damping structure for use with a machine tool, United States Patent No 5,765,818.

[7] Szaraniec-Matusiak A., Technological and design analysis machine bodies and technological devices, Archives of Mechanical Technology and Automation, 27(2007)2, 121-129.

[8] Gawdzińska M., Hajkowski J., Głowacki B., Impact of cooling time on the structure and tribological properties of metal matrix composite castings, Archives of Mechanical Technology and Automation, 28(2008)3, 41-48.

[9] Jackowski J., Evaluation of structural compactness of a casting made of saturated metal composite, Archives of Mechanical Technology and Automation, 22(2002)1, 88-99.

[10] Choi1 D., Kwon W. T., and Chu C. N., Real-Time Monitoring of Tool Fracture in Turning using Sensor Fusion, Int. J. Adv. Manuf. Technol, Springer-Verlag London Ltd., 1999, pp. 305–310.

[11] Hinton F., Owen D.R.J., Finite Element Programming, Academic Press, London, 1977, p. 305.

[12] Rao S.S., The Finite Element Method in Engineering, 2-nd Ed., Pergamon Press, 1989, p. 641.

[13] Zienkiewicz O.C., The Finite Element Method, M-c Graw Hill, 3-rd Ed., 1977, p. 773.

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