Dynamic calculation for lathe body system when some design parameters and external forces were changed values using CosMosDesign software of Finite element methods

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Abstract. The article introduces briefly dynamic calculation results (no-damped natural vibrations and external force vibrations) for lathe body system when some design parameters and external forces were changed values by using CosMosDesign software of Finite element methods. The results of calculation and analysis shall base to research and to increase qualities to come mechanical manufacture centers.

1. Forward

For structures with complex shape, finding an exact solution using analytical methods is almost impossible. However, the finite element method (FEM) can basically solve the problem.

In the past, while calculating and designing metal cutting machines, including machine bed system – a very important assembly, the stiffness, strength are mostly calculated basing on experience or by other traditional methods of low accuracy [3].

This problem refers to the impact of changing in certain structural dimension parameters such as length, cross-sectional size of the bed on natural vibration properties and variable force to determine internal forces, deformation affecting the accuracy of the machining by using FEM software. The mentioned calculation software here is Cosmos Design.

2. Calculation of natural vibration as changing[1] some structural dimension parameters

Model and information of element meshing are shown below with the material of the machine bed (including the cutting test detail) assuming the entire cast iron, the machine body is placed on a base, considering to be a hard restraint connection with the background (see Figure 1 below) [5].

The problem solves the following [2] matters:
- The bed is remained as designed: The length and cross-section (in Figure 2);
- Increase or decrease the length of the machine bed of 25 %;
- Increase or decrease the height of the machine bed of 25 %.
- Increase or decrease the width of the machine bed of 25 %.

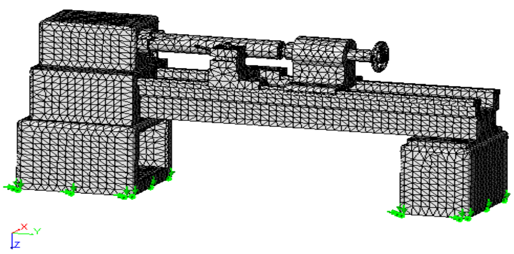

Fig.1. Model meshing body system.

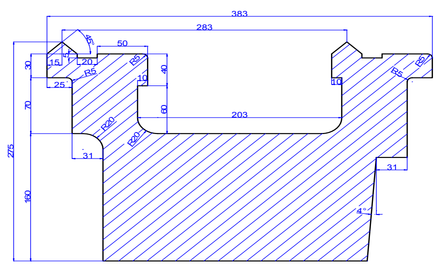

Fig. 2. Structural cross-section of machine bed

[1] Note: When you increase or decrease the number, just change them. The other parameters are unchanged.

After running the software the result as follows:

2.1. As designing

- The value of 10 natural frequencies and periods as shown in Table 1 below:

Table 1 - Value of 10 natural frequencies, Hz (period respectively), s

Mode List		
Mode Number	Frequency, in Hertz	Time period in Seconds
1	113.91	0.019554
2	189.26	0.010478
3	199.23	0.0097166
4	229.43	0.0082067
5	280.46	0.0067102
6	373.75	0.0052953
7	406.44	0.0051012
8	425	0.0043735
9	440.8	0.0042142
10	480.43	0.0037438

- Two natural vibration types corresponding to two first natural frequencies are presented in the Figure 3 and Figure 4.

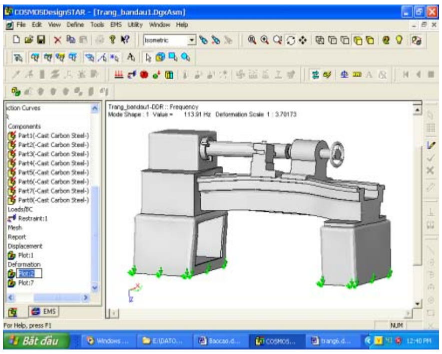

Fig. 3. Natural vibration type corresponding to first natural frequency

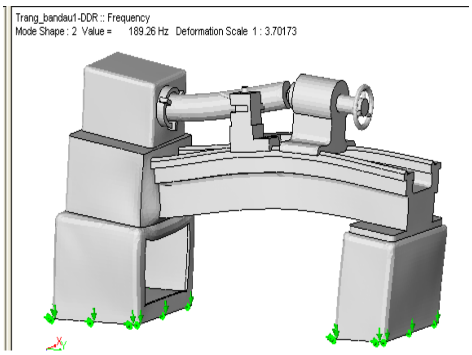

Fig. 4. Natural vibration type corresponding to second natural frequency

With a similar calculation, we have also the following results:

2.2. When the length of the bed increases and decreases of 25 % (Second level headings are typed as part of the succeeding paragraph)

- Value of the first 10 natural frequencies compared to designing is presented in Figure 5 below.

Comment 2.2:

- When the bed length increases, its natural frequency decreases comparing to the original design machine due to the fact that the rigidity of the lathe body decreases;

- When the bed length decreases, its natural frequency increases comparing to the original design machine due to the fact that the rigidity of the lathe body increases.

2.3. When the height of the bed increases and decreases of 25 %

- Value of the first 10 natural frequencies compared to designing are presented in Figure 6 below.

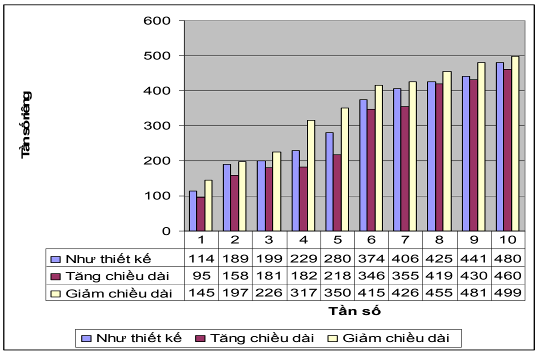

Fig. 5. Changing of natural frequency when the bed length is increased and decreased

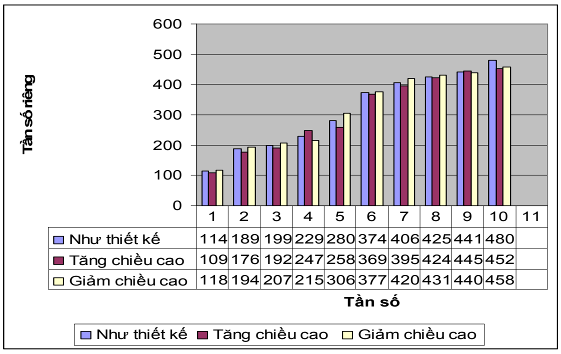

Fig. 6. Changing of natural frequency when the bed height is increased and decreased

Comment 2.3:

- Increase the width of machine bed compared to the original designed width, the natural frequency increases demonstrated that hardness of the lathe body increases;

- Decrease the width of machine bed compared to the original designed width, the natural frequency decreases demonstrated that hardness of the lathe body decreases.

2.4. When the width of the bed cross section increases and decreases of 25 % :

- Value of the first 10 natural frequencies compared to designing are presented in Figure 7 below.

Comment 2.4:

- Increase the width of machine bed cross section compared to the original designed width, the natural frequency increases due to the hardness of the lathe body increases;

- Decrease the width of machine bed cross section compared to the original designed width, the natural frequency decreases due to the hardness of the lathe body decreases.

The first natural vibration types are given below when the cross section width reduces 25 % (see Fig. 8):

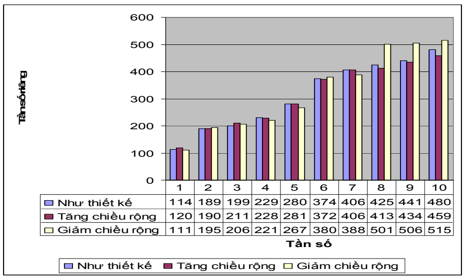

Fig. 7. Changing of natural frequency when the bed cross section height is increased and decreased

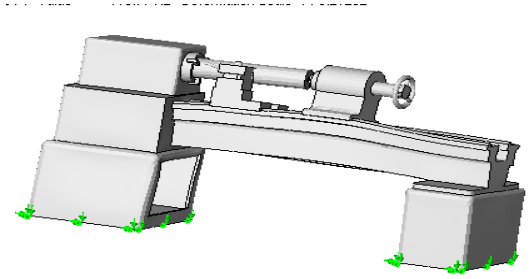

Fig. 8. Natural vibration type with the first natural vibration

3. The system under external force by cutting work

The problem in this article is mentioned in cut testing mode [2]. The cross slide contact with the bed by contacting planes on the sliding ways, so the forced located on bed are not concentrate forces. They are distributed forces on a plane. To make it simpler, we will calculate the counter force on the surface of sliding ways of the bed. After that, the pressure acting on the sliding ways will by found by divide the counter force by contacting area. Forces acting on the bed include the acting of the cross slide and the tailstock assembly [3]. This problem will calculate the system acted by a static load combination. We will not present calculation implementation; just only present the following results:

- Counter force components:

$$F_A = 2849, \text{N}; F_B = 5771 \text{ [N]}; F_C = 658 \text{ [N]}.$$

- The friction force components on slider:

$f_A = 1400 \,.\, 0{,}15 = 210$ [N]; $f_B = 5771 \,.\, 0{,}15 = 865{,}65$ [N]; $f_C = 658 \,.\, 0{,}15 = 98{,}7$ [N].

Based on these values and construction of calculation model using software with the above acting force value, the following results are received (see table 2) for two cases:
+ 100 % of the force as the original calculation - Type 1;
+ Increase the acting force by 150 % original - Type 2.

Calculation and force diagram for the bed system is drawn in Figure 10, in which the link of the machine tool to the foundation are considered to be a hard restraint and force placed on the machine bed is constant [6]. Problem considering the force changed by the time will be mentioned in the other article.

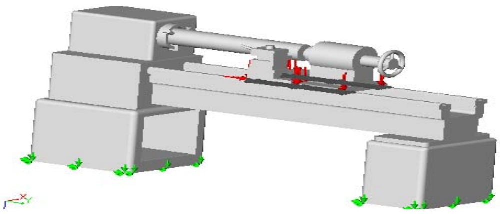

Fig. 9. Calculation and force diagram for the bed system

Maximum and minimum value of stress, displacement and relative deformation after calculation are presented in the table 2 below.

Table 2 - Value of stress, displacement and deformation type 1 and type 2

Type	Stress, MPa												Displacement, μm		Relative deformation,%	
	σ_1		σ_2		σ_3		σ_x		σ_y		σ_z		Max	Min 10^{-27}	Max 10^{-5}	Min 10^{-10}
	Max	Min	Max	Min	Max	Min	Max	Min	Max	Min	Max	Min				
1	9.2	-7.2	5.2	-8.1	2.7	-29	5.2	-1.6	7.8	-2.8	8	-11.5	12	1.0	5.1	4.1
2	14	-10	7.8	-12.	4.1	-44	7.8	-24	11	-43	13	-17.3	18	1.0	7.7	4.8

The chart comparing maximum and minimum stress, displacement and deformation between type 1 and type 2 are presented in the Figure 10 and Figure 11 below.

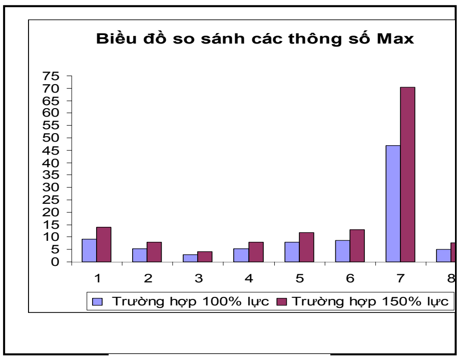

Fig. 10

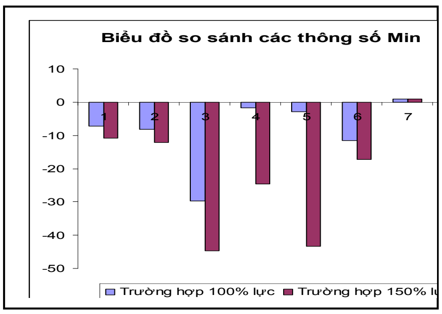

Fig. 11

In Figure 10 and Figure 11:

Column 1 – First principle stress;
Column 2 – Second principle stress;
Column 3 – 3rd principle stress;
Column 4 – Normal stress σ_x ;
Column 5 - Normal stress σ_y ;
Column 6 - Normal stress σ_z ;
Column 7 - Displacement;
Column 8 - Relative deformation.

The safety factors are also as the table 3.

Table 3 - Safety factor for 100 % and 150 % cutting force

Safety factor	100 %	150 %
Maximum tangential stress (Tresca) $\frac{\tau_{m\cdot}}{0.5.\sigma_{Limit}} < 1$	7,84	6,28
Maximum normal stress $\frac{\sigma_1}{\sigma_{Limit}} < 1$	7,54	5,42

The field of stress, displacement and of the machine bed system in case of being acted with 100 % cutting force is presented in Figure 12 and Figure 13.

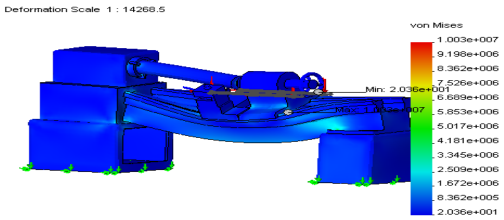

Fig. 12. Stress field

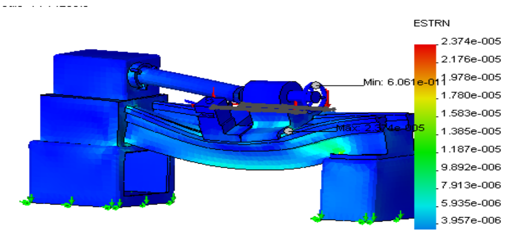

Fig. 13. Displacement field

In which:

$$\sigma_{Vonmiss} = \sqrt{\frac{1}{2}\left[(\sigma_x - \sigma_y)^2 + (\sigma_y - \sigma_z)^2 + (\sigma_z - \sigma_x)^2\right] + 3(\tau_{xy}^2 + \tau_{xz}^2 + \tau_z^2)}$$

4. Conclusion

- Calculation results demonstrate a general picture of stress, displacement and deformation for the whole of machine bed system with positions of point with considerable stress and deformation. The positions with high stress, displacement, deformation and defletion are all concentrated in the midle area of the machine bed.

- With the design machine bed structure as showed in table 1, the first minimum natural frequency also reach to 113.91 Hz (corresponding to the resonance speed about 7,000 rpm). From 2nd to 10th natural frequencies are also of 1.7 to 4.24 time of the 1st natural frequency. Therefore, the machine bed is dynamically stable. The machine bed works below the resonance frequency [1], [7].

- However, at the cutting test regime 100 %, the maximum displacement in the middle of the machine bed can reach about 12,66 µm; at the cutting test regime 150 % is 18,24 µm. This displacement value is also considerable while combine with the displacement of spindle. Through calculation, it is necessary to increase the width and the height of the cross section of the machine bed about 15%. The machine bed will be more stable. It will be able to work at 10,000 to 15,000 rpm without resonance. The cutting accuracy will be higher (the maximum displacement is only about 10 µm, the machine will meet the accuracy level 2 in accordance with TCVN 5882:1995) [4].

5. Reference

[1] Nguyen Van Khang, *Technical vibration*, Science and Technology Publisher, Hanoi 2001.
[2] Design documents of T18 lathe, Hanoi Company of mechanical Engineering, Hanoi 2008.
[3] Nguyen Anh Tuan, Pham Dap, Pham the Truong, Nguyen Huy Hien, *Dynamis of tool machines*, Publisher of Hanoi University of Tecnology, HaNoi 1991.
[4] TCVN 5882:1995, Small purpose general lathe.
[5] Ahmed A. Shabana, *Dynamics of multybody systems*. Second Edition, Cambridge University Press 1998.
[6] Rao S.S, *The finite element method in engineering*, Pegamon Press 1990.
[7] Shabana A.A (1997), *Vibration of discrete and continouous systems*, 2nd ed, New York, Springer Verlag.