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STATIC CHECKING COMPUTATION OF A LATHE TAILSTOCK

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Abstract: *Working accuracy is one of the crucial parameters of every machine and especially of machine tools. This is the reason why checking and computation of the primary parts of all new designed lathes has to be performed to ensure the optimal rigidity, dimensions and weight. This paper describes the computation and results of the strength evaluation, static stiffness assessment and checking of contact normal stress in the guideways of a tailstock of a universal centre lathe. To acquire all of these values, a computation was performed using the finite element method (FEM) in MSC Marc 2011. The computational model was loaded by the maximum forces that were obtained via analytic calculations from the load spectrum of a lathe. The strength evaluations of all individual parts of the lathe were investigated according to two different criteria. Computed values of contact normal stress were compared with maximum pressure loads. Displacements of the centre of the tailstock were found by using this FEM computation. The stiffness was calculated in three mutually orthogonal directions from these displacements and loading forces.*

Keywords: Tailstock, Lathe, Stiffness, Machine tool, Finite element method.

1. Introduction

The demand for higher accuracy NC lathes has increased dramatically with respect to machining accuracy requirements (Mori et al., 2009). The bodies of contemporary machine tools are required to be rigid and to efficiently damp vibration (Staniek et al., 2012). The goal of this computation was to investigate the contact normal stress, evaluate the strength and calculate the static stiffness of the tailstock assembly (Fig. 1a) by using finite element method (FEM). If the tailstock is used for turning operations, its stiffness has a significant influence on machining accuracy. Checking the contact normal stress in the guideways and strength evaluation is important for every designed machine and this complex task can be solved only by using computational software. Modern computational software and sufficient computer power mean that a computational model can be created that approaches reality (Max et al., 2016).

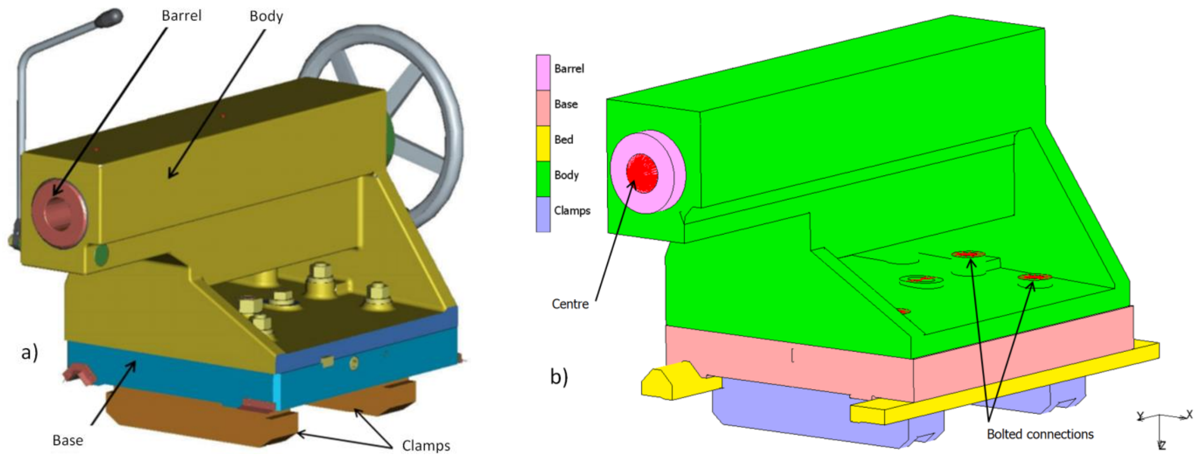

Fig. 1: a) CAD model of a tailstock assembly of a universal centre lathe; b) Computational model.

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Body, base, barrel, clamps of the tailstock and the lathe bed were modelled volumetrically. The lathe bed was simplified and only part of guideways was modelled. The rest of the lathe bed was considered to be absolutely rigid. The centre, bolts connections and moving mechanism of the barrel were simplified for this computation (Fig. 1b). Stiffness was examined in three mutually orthogonal directions by loading the tip of the centre. The calculations were made in the FEM system MSC Marc 2011. Only small deformations were considered in this calculation.

2. Calculation settings

Body, base, barrel, clamps of the tailstock and the lathe bed were considered to be pliable in the computational model and were discretized using linear volume elements with 8 (hex8) and 6 (penta6) nodes. Bolt connections were simplified in the computational model using two types of ties. The contact surface between the bolts and body of the tailstock were linked by an imaginary point in the centre of the hole by using absolutely rigid links (Rigid Link in MSC Marc). Also the contact surface between the nuts and clamps were simplified in the same manner. These two imaginary points were connected by links with a possibility of preload (Overclosure link in MSC Marc). These overclosure links were loaded with the preloading force recommended by the bolt producer. Lašová et al., (2006) described other methods for simplifying preloaded bolted joints. The centre of the tailstock was considered to be absolutely rigid and was simplified using rigid links. The tip of this simplified centre was loaded by force. Also the mechanism that ensures the movement of the barrel was considered to be absolutely rigid.

Hook's law was considered to be valid for all the materials used. Material parameters are stated in Tab. 1. Cast iron (EN-GJL-300) was used for the body of the tailstock, the base of the tailstock and the lathe bed. The barrel and clamps were made of steel (S235JRG2).

Tab. 1: Mechanical properties of used materials.

Material	**Density [kg·m^{-3}]**	**Young's modulus [GPa]**	**Poisson's ratio [-]**	**Compressive strength [MPa]**	**Tensile strength [MPa]**	**Yield strength [MPa]**
EN-GJL-300	7350	130	0.25	860	300	-
C45	7800	210	0.3	540		325

For the evaluation of the contact normal stress the values of maximum pressure load was considered to be loaded without simultaneous movement. The value of 40 MPa is valid for contact surfaces without heat treatment and the value of 70 MPa is valid for hardened contact surfaces.

The friction contact (contact touching in MSC Marc) was defined between parts of the model: barrel and body, body and base, base and bed, bed and clamps. This contact allows the transfer of all forces except for tensile normal forces. The values of tangential forces that are transmitted via this contact are derived from the coefficient of friction that was specified as 0.15 for all contacts. Only for the contact between the base of the tailstock and lathe bed the coefficient of friction was specified as 0.01 because in the real machine there is a bearing material with low friction.

Specific geometric and static boundary conditions were set for the investigation of all outlined goals. All displacement in the lower and side surfaces of the simplified bed were prohibited in the model - Fig. 2b. For this computation, the computational model was loaded with static forces applied on the tip of the centre - Fig. 2a.The value of the loading forces was calculated via analytical calculations from the load spectrum of the lathe and these forces are stated in Tab. 2 in column 1. Forces in column 1 were used to investigate the contact normal stress and for strength evaluation. Forces stated in columns 2, 3 and 4 were rounded and used for simple investigation of the stiffness in three mutually orthogonal directions.

Tab. 2: Loading forces.

Calculation - *n*	**1**	**2**	**3**	**4**
Force in the X direction [N]-$F_{X,n}$	-21 000	-20 000	0	0
Force in the Y direction [N]-$F_{Y,n}$	-41 500	0	-40 000	0
Force in the Z direction [N]-$F_{Z,n}$	-20 000	0	0	-20 000

Also the bolted connections in the model were loaded with static preloading force - Fig 2b. The preloading force of the bolt joints was always exerted on the node in the centre of the screw contact surface, while this central node was rigidly tied to nodes that correspond with their position to the contact surface between the tailstock body and the bolt head. The computational model was loaded in three steps. In the first step the geometric boundaries were activated (prohibition of displacement of the lathe bed). In the next step the bolted joints were preloaded and in the last step the force was applied.

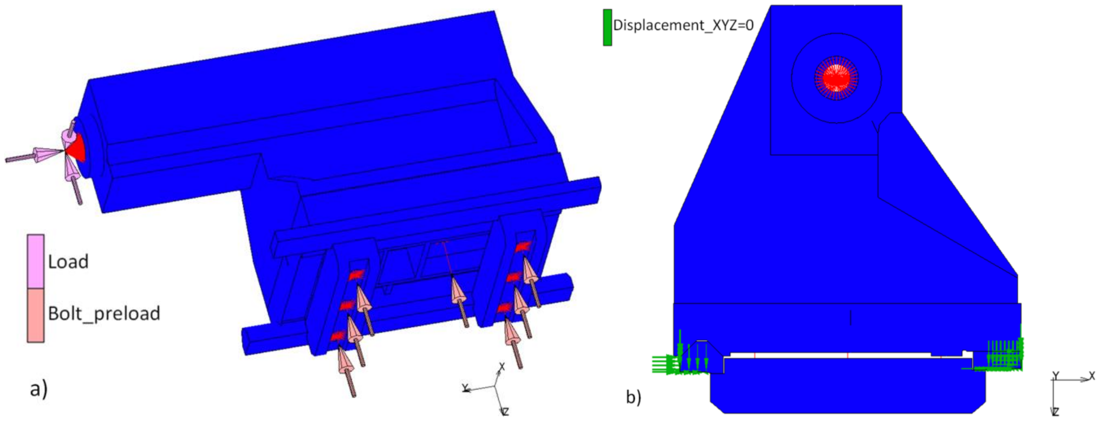

Fig. 2: a) Static boundary conditions; b) Geometric boundary conditions.

3. Evaluation of contact pressure

Values of resultant contact normal stress in all surfaces were lower than values of maximum pressure load except the values of the contact normal stress between the bed and the clamps. Values of stress in this contact surface reached 185 MPa while the permissible load is 70 MPa, but this permissible value was defined for analytical calculation when considering absolutely rigid contact surfaces with uniform distribution of stress. Also in this case the peak value of the stress was present only in one node and if the values of the contact normal stress were averaged the permissible load would not be exceeded.

4. Strength evaluation

Strength evaluation was based on the values of the stress that were linearly extrapolated to the nodes from the values of the integration points. The Von Mises criterion was used to evaluate the strength of the barrel and clamps. For strength evaluation of the body, base and bed, all individual components of the stress tensor were investigated according to the maximum stress criterion. The distribution of the major principal values of the stress on the tailstock is shown in Fig. 3a). The maximum value of the tensile stress was 101 MPa and the compressive stress 100 MPa. It is possible to calculate the safety factor from these values and from strengths of the material (Tab. 1) $s_{tensile} = 860/101 = 8.5$ and $s_{compressive} = 300/100 = 3$. The worst safety factor of the assembly was calculated for the clamps. The distribution of the equivalent tensile stress of the clamps is shown in Fig. 3b) when $s_{clamps} = 325/200 = 1.6$.

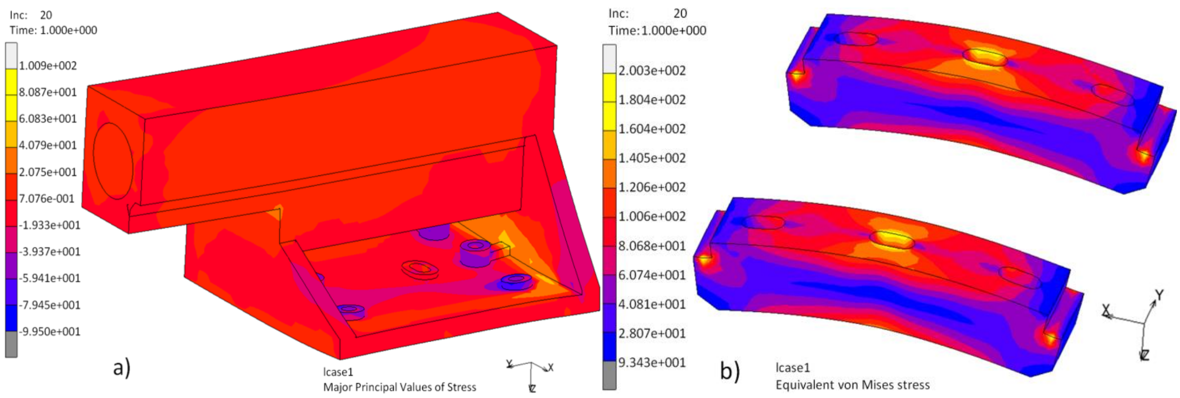

Fig. 3: a) Stress distribution of the tailstock body [MPa]; b) Stress distribution of the clamps [MPa].

5. Stiffness evaluation

The stiffness was investigated in the directions of three coordinate axes based on the loading forces and the resulting displacements of the tip of the centre, according to the equation:

$$k_{i,n} = \frac{F_i}{u_{i,n}} \tag{1}$$

while F_i is loading force from Tab. 2, where the index i is coordinate X, Y or Z and the index n is the number of calculation (2-4) from the same table.

Ascertained displacement of the tip of the centre $u_{i,n}$ and calculated values of stiffness according to equation (1) are given in Tab. 3. In Fig. 4a you can see an example of the discovered displacement of all nodes in the direction X for calculation 2 (load only in X direction). In Fig. 4b you can see an example of the discovered displacement of all nodes in the direction Z for calculation 4 (load only in Z direction).

Tab. 3: Displacement of the tip of the centre and stiffness of the tailstock.

Calculation	**2**		**3**		**4**	
Quantity	**Displacement**	**Stiffness**	**Displacement**	**Stiffness**	**Displacement**	**Stiffness**
Unit	mm	kN/mm	mm	kN/mm	mm	kN/mm
X direction	-0.195	**103.6**	0.133	-	0.099	-
Y direction	0.021	-	-0.128	**312.5**	-0.016	-
Z direction	0.067	-	0.063	-	-0.085	**235.3**

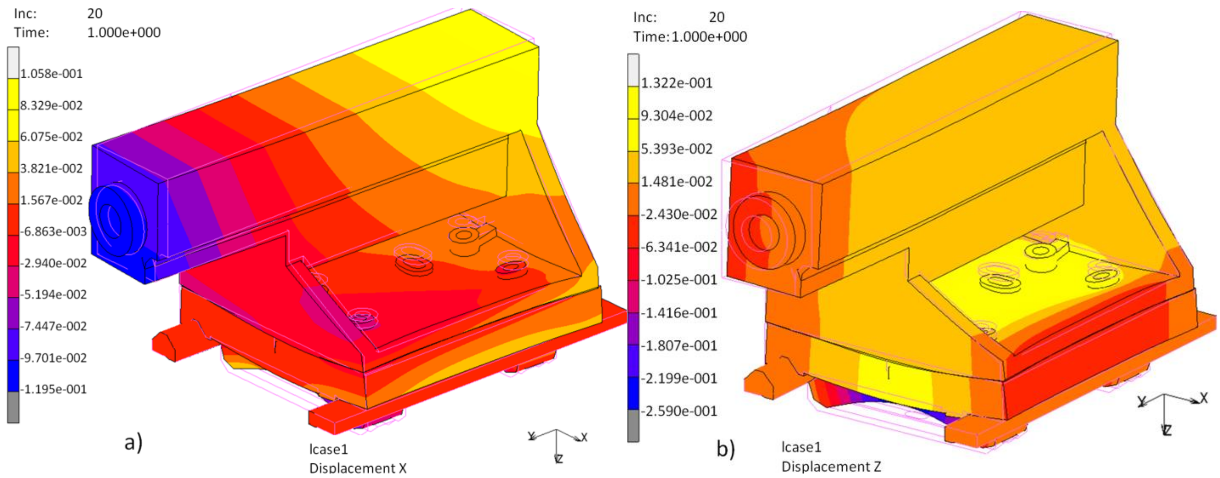

Fig. 4: a) Displacement X, calculation 2 [mm]; b) Displacement Z, calculation 4 [mm].

6. Conclusion

The aim of this computation was to investigate the strength of the entire design, evaluate the static stiffness of the tailstock assembly and check the contact normal stress in the guideways of a tailstock. All values were gained via a FEM computational model that is described in this article. The results of all three tasks are briefly described. The discovered values of stiffness show that the tailstock has the lowest stiffness in the X axis. Computation also showed that the clamps are the most stressed part of the assembly and also have the highest value of contact normal stress.

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