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PARAMETERS OPTIMIZATION OF MACHINE TOOLS BODIES

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ABSTRACT
This paper introduces to optimization of heavy machine tools bodies. It presents proposed classification of machine tools bodies, from optimization point of view. Parametric optimization results of exemplary machine tools body – planer mill slider, also were presented. Optimization was performed with using Ansys system. The aims of optimization were maximization of static stiffness and mass minimization. In this paper the comparison of FEA results of the slider body after optimization and FEA results of slider, proposed by designer, were also presented.
Keywords: analysis and modeling, machine tools, optimization

1. INTRODUCTION
A large majority of products available at this time are obtained as optimization results. The optimization process concerning mainly mass minimization. In the consequence of it, mainly in case of mass and series production the meaningful cost reduction were obtained. We assumed that the using properties of product in the optimization process have to be keep on the constant level. Machine tools designs analyzed from few years pointed on the lack of optimization stage during the production preparation process. Furthermore pointed the scientifically field, which should be enable to prepare new constructions with better using properties. It is connected with necessity of preparation of machine tools optimization methodology supported by practice examples.

2. MACHINE TOOLS BODIES CLASSIFICATIONS FROM OPTIMIZATION POINT OF VIEW
The heavy machine tools are very specific machine group. It comes from few reasons. The first of them concerning on overall dimensions and demanded accuracy. Overall dimensions contain in range of several to tens meters and demanded accuracy contains on the level of micrometers. The following causes are very large costs connected with production of single machine tool and relatively short time of order realization and the same, time-constraints of design process. In many cases new constructions are worked out on the base of checked design solution, which are not always optimal. It is consequence of lack of prototypes and possibility of investigate and then correction of improper design solutions.
At the beginning of heavy machine tools optimization process we have to pay attention, that costs connected with wrongly selected design solution (shapes and dimensions) could have very important economical results. Therefore designer always has to take into consideration the advantages of optimization process and importance of assumed criterions. From optimization process point of view we can divide the heavy machine tools bodies in the following method:
1) Bodies for which significantly are mass reduction with simultaneously keeping static stiffness on the unchangeable level. It contained movable bodies, such as, for example: carriages of vertical lathes or planer mills, high speed rotational tables. In many cases for this type of bodies the high speed velocity corresponded to high speed machining (HSM) is demanded. Taking into account the large

mass of these bodies (even a few tons) the selection of feed drive is most difficult and significantly increases the costs. Additionally in case of these bodies significant is problem of thermal displacements connected with realization of main drive (e.g. the electrospindle) and feed drive (e.g, ball nut, linear drive).

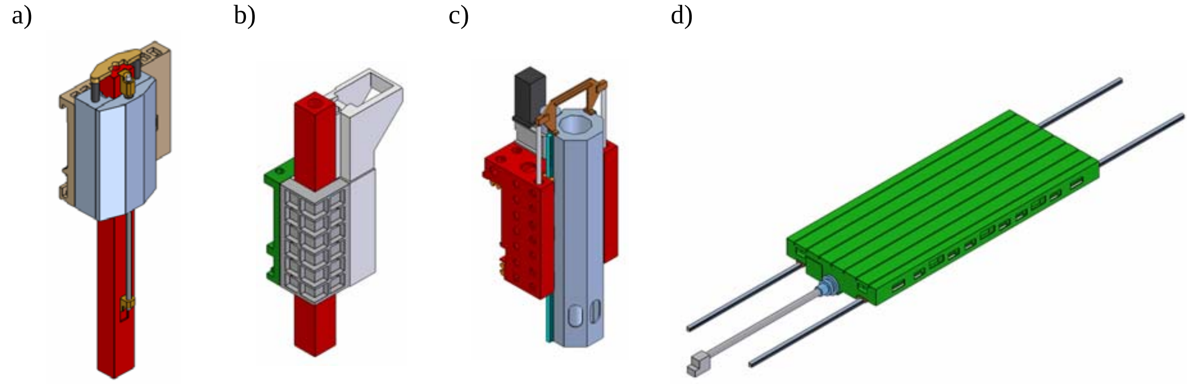

Figure 1. The movable subassemblies of heavy machine tools: a) the saddle of vertical lathe KDC [3], b) the saddle of vertical lathe KCI type[2], c) the saddle of planer mill HSM 180 to high speed machining [1], d) the table of planer mill HSM 180 [1]

2) Bodies of supporting structure (Figure 2), for which geometry and mass affected on dynamic properties of whole structure, e.g. the columns, the crossrails ect. For these bodies the most important are ensured of suitable static stiffness and dynamic stiffness connected with natural frequencies. Mass reduction with unchangeable static stiffness should have an effect on increasing of natural frequencies. In this case, for the sake of large overall dimensions mass reduction should have also economical meaning.

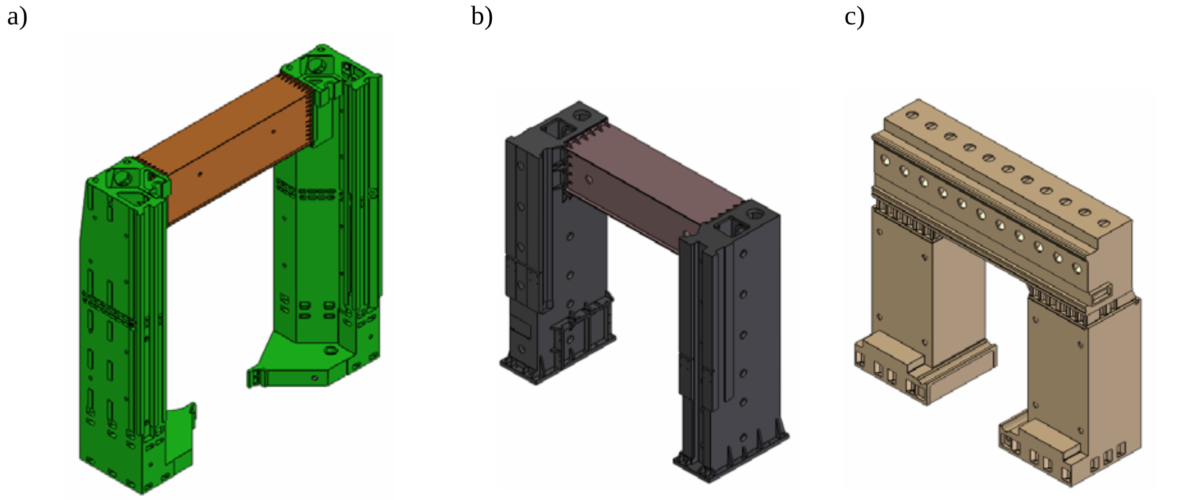

Figure 2. The supporting structure of heavy machine tools: a) the columns with beam of vertical lathe KDC [3], b) the columns with beam of vertical lathe KCI [2], c) the columns with crossrail of planer mill HSM 180 [1]

3) Bodies of supporting structure decided about static stiffness, which have less influence on dynamic stiffness, such as e.g. the beds. In these cases decreasing of mass will has mainly economical meaning, but should not decrease the static stiffness of whole machine tool.
4) Bodies made as uniform bodies, for which advisable is decrease of mass with simultaneously small decrease, in range of few percent, of static stiffness. This group of bodies contains the slides of vertical lathes and planer mills.
All sources of inaccuracy come from bodies geometry should be taken into consideration during the optimization process. There are: static stiffness, thermal displacements (which come from subassemblies realized feed drive and principal motion) and dynamic properties like e.g. natural frequencies.

3. OPTIMIZATION OF PLANER MILL SLIDE

Optimization of planer mill slide was made with using Ansys system. The body of slide proposed by designer is a welded steel design. The geometry of slide in the Figure 3 was presented.

a) b) c)

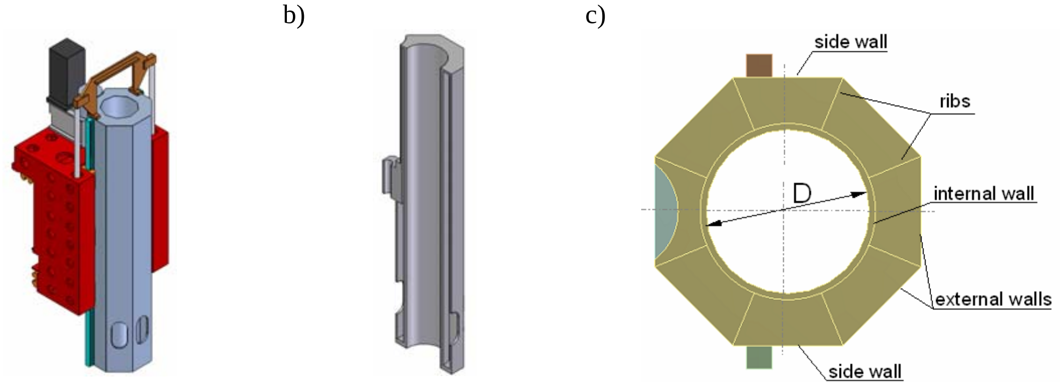

Figure 3. The geometrical shape of movable subassemblies of planer mill HSM 180 [1]: a) the saddle with the slide, b) the slide in longitudinal section, c) the elements of slide geometry to parameters optimization process

In optimization process the shape of design proposed by designer was assumed as reference point. The optimization was made for maximal throat distance of slide as the worst case for static stiffness of whole machine tool. It was assumed, that overall dimensions of slide come from arrangement of ball slides and maximal dimensions of the shape proposed by designer. The optimization process contained: thickness of ribs, thickness of external wall, thickness of walls clamping the ball slides and diameter of axial hole. The statement of input parameters range of change in Table 1 were presented. The analysis of static stiffness of slide body proposed by designer was made until starting optimization process. Displacements of slide on the X, Y direction, as a result of loading force 10kN independent on the individual direction, were analyzed.

a) b) c)

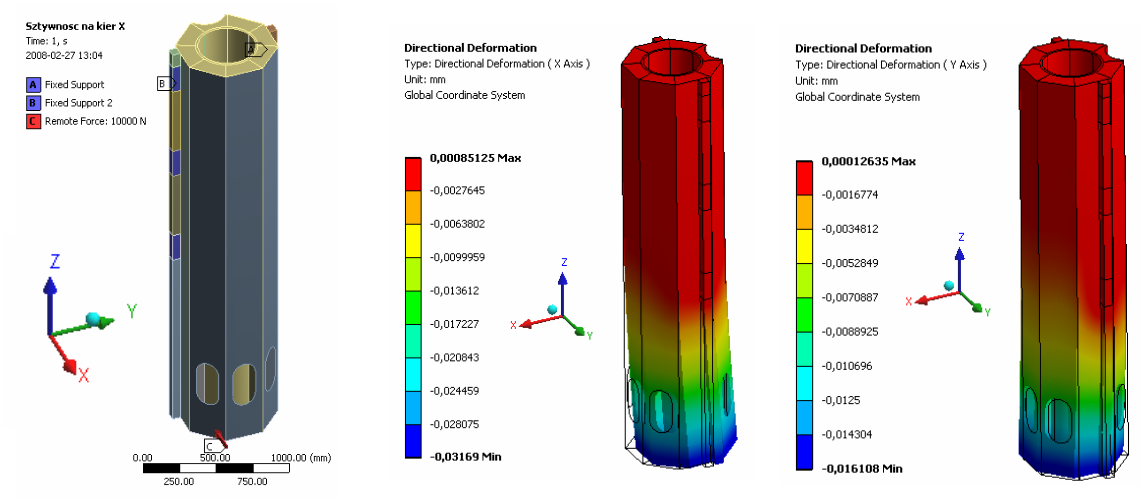

Figure 4. The assumption and results of simulation research of slide with shape proposed by designer: a) the boundary conditions, b) the displacements distribution from static stiffness analysis on X direction, c) the displacements distribution from static stiffness analysis on Y direction

As a results of optimization two design solution were selected. In the first model (Model 1), proposed thickness of ribs, walls and diameter D should improve properly 4,6 and 15,8 % with simultaneously small reduction of mass (about 2%). In the second solution (Model 2) obtained 10% mass reduction and 3% increasing of static stiffness on X direction with simultaneously about 3% decreasing of static stiffness on the Y direction. Proposed as a results of optimization the dimensions of the body were showed in the Table 3. In the Table 4 the comparison of obtained changes of static stiffness and masses also were showed.

Table 1. The range of input parameters changing to parameters optimization process of the slide

Name of input parameter	Range of change	Character of changes	Value proposed by designer
Thickness of side wall, mm	50; 60	discrete	50
Ribe thickness, mm	10; 15; 20	discrete	20
Thickness of side ribe, mm	10; 15; 20	discrete	20
Thickness of external walls, mm	20; 25; 30; 35	discrete	20
Thickness of internal wall, mm	15; 20; 25	discrete	30
Diameter D, mm	250-430	continuos	427

Table 2. The results of slide parameters optimization

Name of parameter	Model 1	Model 2
Thickness of side wall, mm	50	50
Thickness of ribe, mm	10	10
Thickness of side ribe, mm	15	10
Thickness of external walls, mm	35	30
Thickness of internal wall, mm	15	15
Diameter D, mm	428,09	370,64
Mass, kg	3037,4	2789,8
Maximal displacement on the X direction, µm	27,2	30,6
Maximal displacement on the Y direction, µm	15,3	16,5

Table 3. The values of output parameters and obtained as a results of optimization the changes in relation to geometrical shape and dimensions proposed by designer (the sign "+" – means increasing, the sign "-" – means decreasing of value)

Model	mass , kg	reduction of mass, %	index of static stiffness, N/µm		changes of static stiffness index, %	
			direction X	direction Y	direction X	direction Y
Model proposed by designer	3098,1	----	317,5	625,0	----	----
Model 1	3037,4	**-2,0**	367,6	653,6	**+15,8**	**+4,6**
Model 2	2789,8	**-10,0**	326,8	606,1	**+2,9**	**-3,0**

4. SUMMARY

Presented results showed, that design shapes of used bodies are not fully optimal from static stiffness point of view. It should pay attention, that static stiffness has meaningful influence on the machine tool accuracy. On the basis of performed research two ways of the machine tools bodies optimization are possible to showed: increasing of static stiffness with simultaneously small mass reduction and reduction of mass with simultaneously small reduction of static stiffness. For fully optimal machine tools bodies the minimization of thermal displacements and maximization of their dynamic stiffness should be taken into account.

5. REFERENCES

[1] Design documentation of planer mill HSM 180, Rafamet S.A., 2004.
[2] Design documentation of vertical lathe KCI series, Rafamet S.A., 2001.
[3] Design documentation of vertical lathe KDC 700/SOON, Rafamet S.A., 2006.