THE IMPROVEMENT OF THE OPERATIONAL PROPERTIES OF A WHEEL LATHE BASED ON SIMULATION TESTS

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Abstract: The supporting structure of a machine tool significantly determines the accuracy, quality and repeatability of machining. The material and technology of its execution determine the operational properties of a machine tool. The article attempts to answer the question whether it is possible to improve the accuracy and quality of the surface being machined on a wheel lathe by partial modification of an already existing structure. It presents further attempts at structural changes and their influence on such properties as static and dynamic stiffness as well as the natural frequencies and mode shapes. The author decomposed the supporting structure of the machine tool and, on the basis of FEM numerical analyses, indicated its weakest links. In subsequent model tests they were modified. The article is a development of the previously undertaken issues related to experimental and simulation studies of a wheel lathe. The simulation results obtained indicate that it is possible to improve the operating properties of the machine tool by changing the dynamic properties. On the basis of simulation studies it was found that filling the steel bed of the machine tool with polymer concrete would have the greatest influence on improving the dynamic properties.

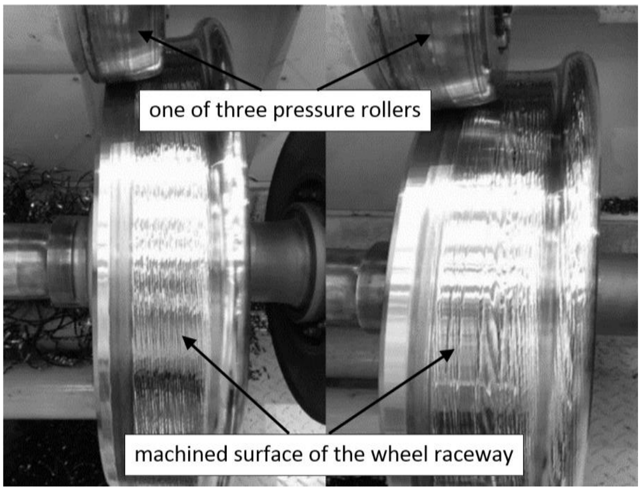
Key words: vibration, Finite Element Method, CNC machine tool, hybrid bodies, polymer concrete.

1. INTRODUCTION

In recent years, there has been an increased interest in modernising railway lines and developing rolling stock. From the point of view of machining, it is connected with an increased demand for machine tools for machining and regeneration of wheelsets. This is also accompanied by trends in increasing the productivity of machine tools and their accuracy. The latter results from the introduction not only in Poland, but also in the whole Europe of high speed railways.

Correct cooperation between the wheelset and rail requires periodic monitoring of the working surface of the wheels and the machining of the running profile on dedicated machine tools. The outline of the road profile is standardised and its shape remains unchanged. Due to the rotationally symmetrical shape of the wheel, most often Abovefloor Wheel Lathes or Underfloor Wheel Lathes, Vertical Turning Lathes, as well as Wheel Milling Machines are used to regenerate the wheel set. The latter were produced in the past in the USSR, and aspects related to machining on these machines can be found e.g. in the literature [12]. During the machining process, vibrations can occur, for example, due to uneven wear of the running surface of a wheelset and the consequent change in the thickness of the cutting layer or unbalance of the worn out wheelset during rotation. The need to ensure precision and stability of the machining process requires the construction of rigid and vibration-resistant machine tools.

It therefore seems appropriate to develop issues related to research and development of machine tools for the railway industry. The present study concerns issues related to the design of machine tools, namely the design of the supporting structure of a wheel lathe. The article is a continuation of the subject matter presented in the article [11], where the problems related to obtaining the desired performance and at the same time the surface quality of wheel set machining are addressed. Preliminary tests of the wheel lathe, showed an unsatisfactory quality of the machined surface (Figure 1). The work test consisted of machining a wheelset by turning the wheel raceway. During the machining of the wheelset on the wheel lathe, the wheelset remains supported in the taper. The forces generated during the turning process, are balanced by three pressure rollers arranged around the circumference every 120 degrees. Therefore, an attempt was made to identify the causes of the poor condition of the machined surface and how to improve it. One of the hypotheses put forward was self-excited vibrations. Generally, the causes of them can be improper machining parameters, poorly selected cutting tool and insufficient machine tool stiffness and structural damping. The applied machining parameters and tool were selected in a manner analogous to similar machine tools of this manufacturer. Therefore, the article concentrates on the topic of designing the supporting structure of a wheel lathe. Special attention has been paid to the problems of the machine tool design based solely on the experience of the designer. The operational features of such a supporting structure are compared with solutions obtained by using FEM simulations to support the design process. The simulation was performed using the ANSYS system.



(a) (b)

Fig. 1. Comparison of surface quality for different machining parameters [11]: a) A = 5mm, S= 99m/min, f = 1.5 mm/rot, d =856mm b) A = 4mm, S= 90m/min, f = 1.5 mm/rot, d = 856mm

The article also provides an answer to a frequently asked question which of the machine tool's supporting structures, from the point of view of operational features, is more advantageous: welded steel, cast iron or maybe a hybrid one. As the experience of the author [9, 10] shows, the latter is often applied too late, i.e. at the moment when the machine is started and the first tests of work show excessive vibrations, making machining with the required parameters impossible.

2. MATERIALS AND METHODS

2.1 General characteristics of machine tool bodies

The supporting structure of a machine tool usually consists of several bodies mechanically connected by twisting, monolithic structures are used less often. The task of the supporting structure is to provide the machine tool with the required static and dynamic stiffness, good vibration damping, stability and unchangeability of the position of the components for which they form the base and thermal stability. The material used for machine tool bodies is mainly cast iron and steel. Good vibration damping properties and ease of forming complex shapes are convincing for the use of cast iron. While steel is more advantageous due to the larger Young's modulus and thus the possibility of obtaining greater static stiffness with less body weight. It is therefore reasonable to use steel structures for moving parts, but some machine tool manufacturers also use steel as a structural material for supporting structures. In case of precision and medium size machine tools, the supporting structures are also made as composite and polymer concrete [1,2,4-6]. The latter is beneficial due to very good vibration damping. The use of so-called hybrid bodies, i.e. steel bodies filled with polymer concrete, is increasingly often considered [8]. This allows for a stiffness appropriate for steel construction and damping appropriate for polymer concrete.

2.2 Subject of the research

The subject of research was a double-saddle wheel lathe (Figure 2). This lathe is dedicated to machining new and reconditioning used wheel sets of rail vehicles. Originally proposed saddle bodies were characterized by a column form, which resulted in a relatively low first natural frequency. At the same time, they exhibited relatively high static stiffness. However, it should be emphasized that the whole structure of the machine tool was made as welded steel.

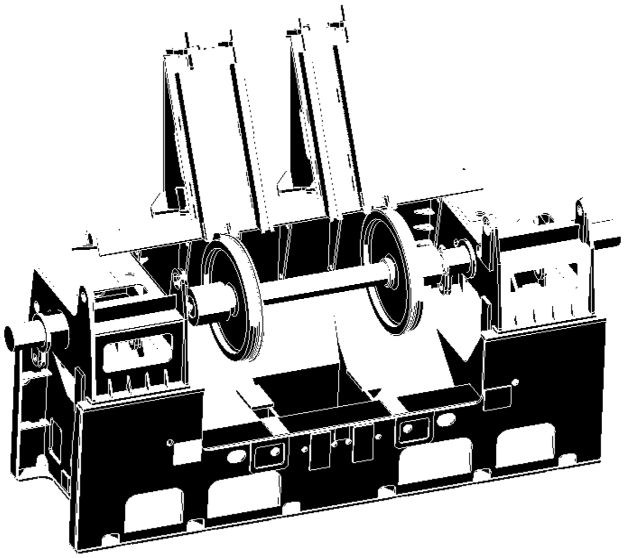


Fig. 2. Wheel lathe design

As a result of the FEA analysis, the possibility of a resonance between the carriage and the wheel set was found (Figure 3). These results confirm the tests carried out on the machine tool (Figure 4). During the tests, the occurrence of self-oscillation and a significant decrease in quality of the machined surface were found (Figure 1) [11].

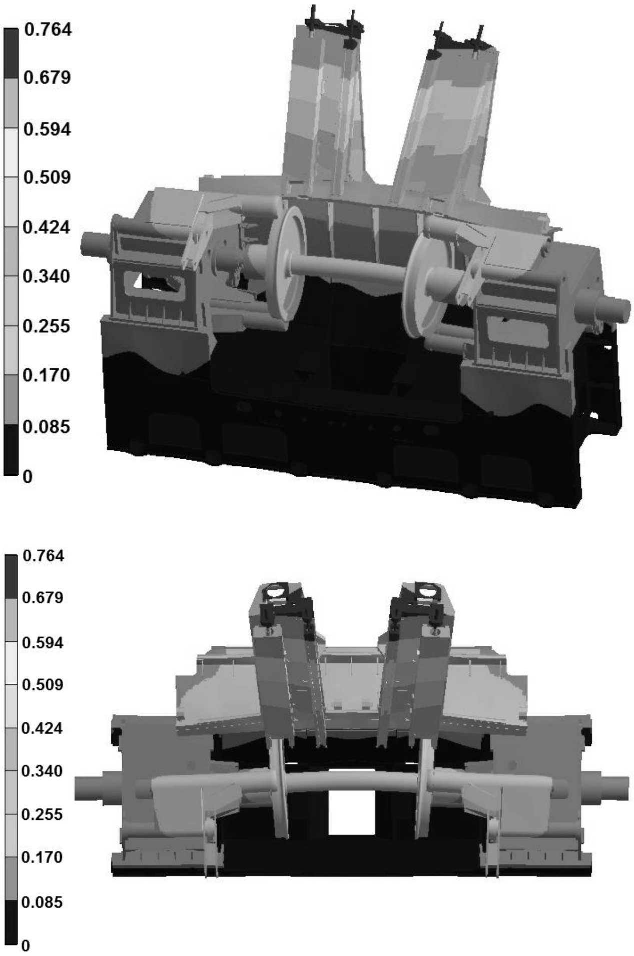


Fig. 3. Modal FEM analysis result - first form of the machine tool natural vibration with wheelset, f1=35Hz

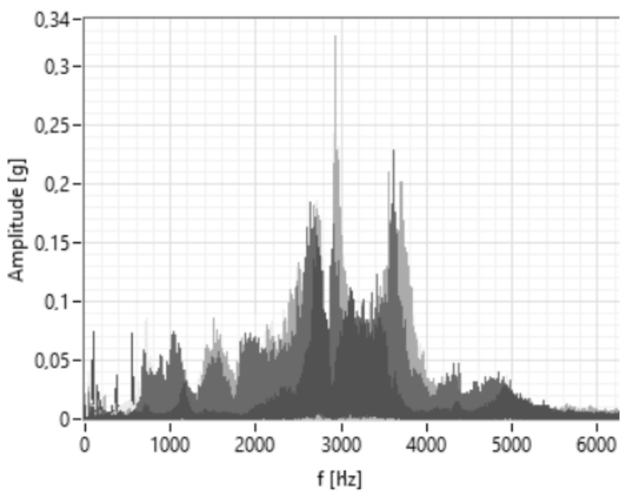


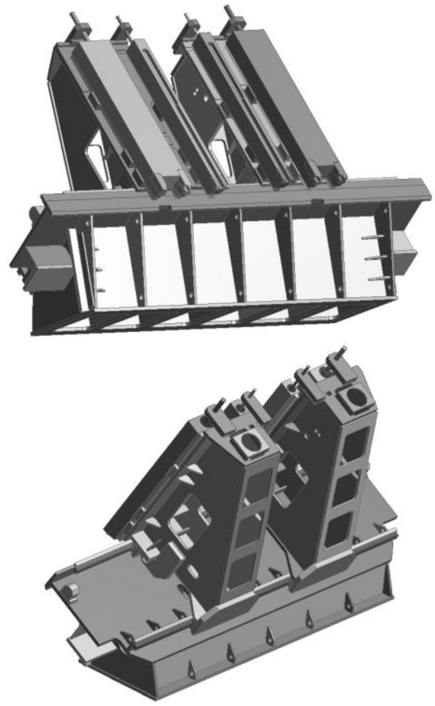
Fig. 4. Frequency spectrum of signals recorded during experimental tests under load on the rolling diameter

Therefore, the possibilities of improving the properties of the machine tool by changing its design were analysed. It was assumed that a change in the structural form of individual machine components should not change the machining space and should not increase the space occupied by the machine tool. The material properties shown in Table 1 were assumed for the simulation study.

Table 1. Material properties for numerical analysis

Property	Structural steel	Gray cast iron	Polymer concrete
Young's Modulus, MPa	$2{\cdot}10^5$	$1.1{\cdot}10^5$	$0.35{\cdot}10^5$
Poisson's Ratio	0.3	0.28	0.3
Density, kg/m3	7850	7200	2250
Logarithmic damping decrement	0.002	0.003	0.03

The proposed structural changes resulted from the analysis of the results of the FEA simulation of the original design and were mainly related to the modification of the carriages and the top of the bed (Figure 5). In case of changes in the construction of the carriages, the column form was abandoned and the bed form was adjusted so that the direction of movement of the carriage and slider corresponds to the original assumptions. The adopted solution is characterized by a more favourable position of the centre of gravity of the carriages and should increase their natural vibration frequency. At the same time, it was assumed that the current geometric form of the slides will be preserved. Components such as guides and the drive system, which moves the carriages and slides, will not be changed either. The proposed design changes compared to the original design are shown in Figure 5(a) and Figure 5(b). On the other hand, the modification of the main part of the bed, carried out in the next stage, consisted first of increasing the thickness of the body walls from 25mm to 30mm (Figure 5(c)). In a further step, changing the steel bed material to cast iron and filling the steel bed with polymer concrete were also considered.



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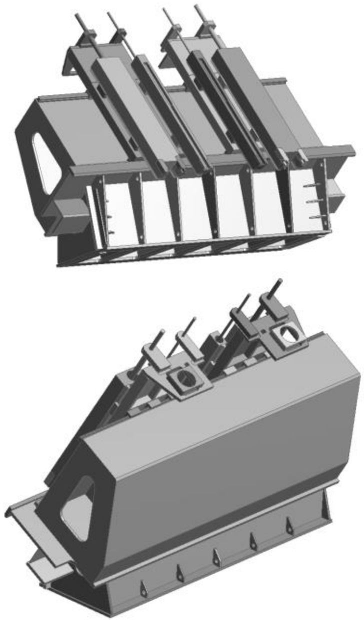

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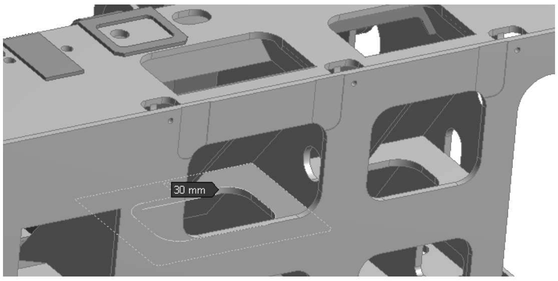


(c)

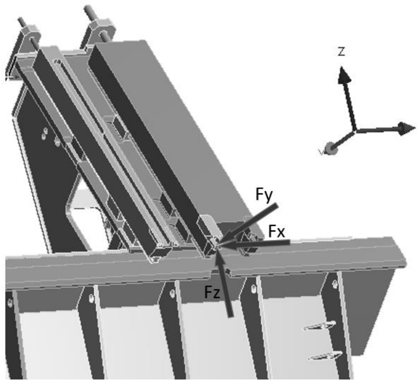
Fig. 5. Comparison of the design of the upper bed and carriage of a wheel lathe before and after modification: a) original design, b) design after modification, c) view of the main bed in which the wall thickness has been changed

3. RESULTS AND DISCUSSION

3.1 Comparative analysis of the results of FEA simulation of wheel lathe models before and after modification.

The impulse to look for a better structural solution for the supporting structure of the lathe was the work test and the self-oscillation of the structure, which resulted in an unacceptable quality of the surface after turning. The first modal analysis of the machine tool showed that they were related to the occurrence of the first mode of natural vibration at a frequency of about 35 Hz (Figure 3, 7(c)). In order to identify the component which, from this point of view, is the weakest link in the structure, the machine tool structure was decomposed. A carriage was extracted from the whole machine tool, and in the next stage a carriage with the upper part of the bed was extracted and a static stiffness and modal analysis was carried out for them and for the whole machine tool. Static stiffness was determined from the displacements obtained when the tool was loaded with a force of 200N. The static stiffness was calculated as the quotient of the forcing force and the displacement measured in the direction of the force. The load was applied independently in three directions. The location and method of carriage loading is shown in Figure 6(a).

The results of analyses (Figure 6) showed a significant increase in static stiffness in the radial direction (Y direction, see Figure 8), i.e. an increase from approx. 40% to 70% depending on the analysed part of the machine tool. For the other directions, no significant changes were noted in relation to the original design. Analysing the results obtained, it was also found that the static stiffness does not differ significantly depending on whether it is analyzed only the carriage itself, the carriage with bed or the whole machine tool. The three structures are shown in Figure 7 for the original lathe and in Figure 8 for the lathe after the design changes.



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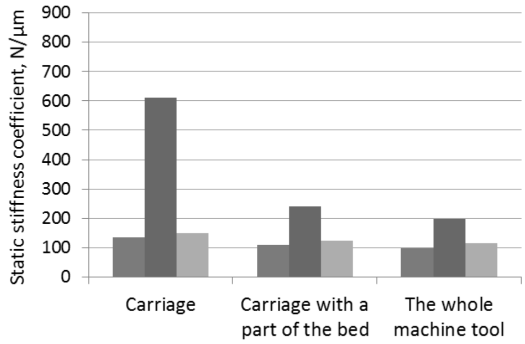

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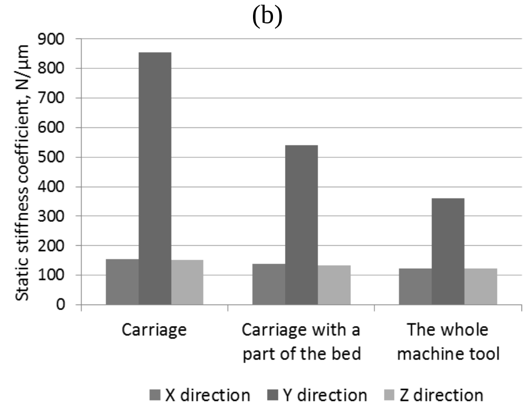

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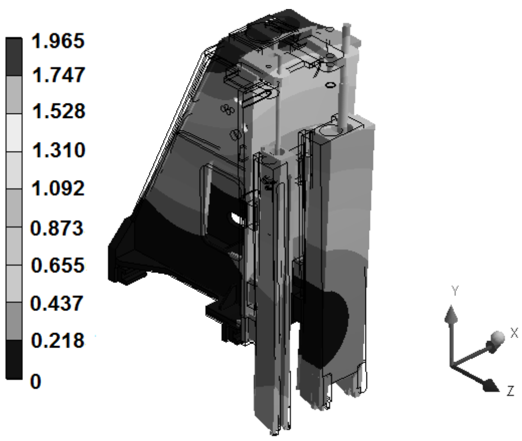
Fig. 6. Overview of machine stiffness coefficients after decomposition: a) load scheme for static stiffness analysis and harmonic response analysis, b) original construction, c) modified construction

Then a modal analysis was carried out to determine the first three frequencies and the mode of natural vibration. With reference to the first natural frequency it was found (Figure 7 and 8) that the first form of natural vibration of the whole machine tool has changed. This is a consequence of a change in the structural form of the carriages and the top of the bed (Figure 5).
However, the change of the corresponding frequency is negligible and is about 1 Hz. Therefore, it can be assumed that the machine, after such a modification, will also work in the resonance range and that the introduced design changes will not improve the quality of the machined surface. However, it is worth noting that in relation to the models of the carriage itself and the carriage with the upper part of the bed, a significant increase in the first natural frequency was obtained. In the case of a carriage, there was an increase of about 135%, and in the case of a carriage with a bed section, the frequency increased by about 42%. Taking into account that there was no fundamental change in the model of the whole machine tool, it was decided to modify also the main (lower) part of the bed in the next step. It was considered to be the weakest link in the supporting structure at this point.



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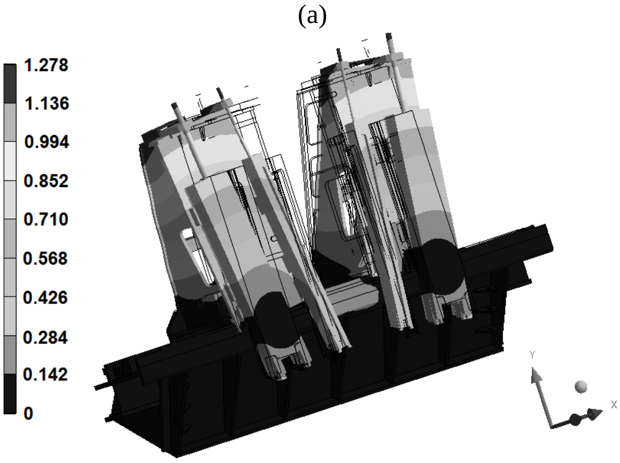

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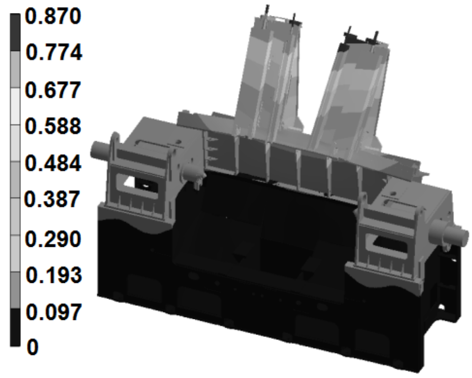

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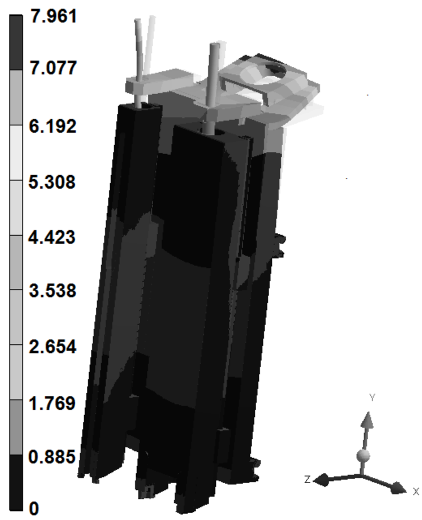
Fig. 7. Modal FEA results for the first form of natural vibration of the original machine tool components: a) carriage f1=71.6 Hz, b) carriage with upper part of the bed f1=50.1 Hz, c) whole machine tool f1=36.5 Hz

Therefore, the thickness of selected walls and ribs of the main part of the bed was increased (Figure 5) and FEA analyses were repeated in order to assess the impact of introduced changes on the operational properties of the lathe. As a result, an increase in static stiffness in the radial direction (Y direction) and a slight increase in static stiffness in the other directions were again obtained (Figure 9). The modes of vibration in relation to the previously modified structure have not changed, but a slight increase in the natural frequency was observed (Figure 10). Analyzing the results obtained, it was found that the first natural frequency is still in the resonance range of machine operation. Therefore, it was decided to analyze the response of the system to the harmonic force in order to assess the dynamic stiffness of the lathe and thus the vibration amplitude at the assumed dynamic load.



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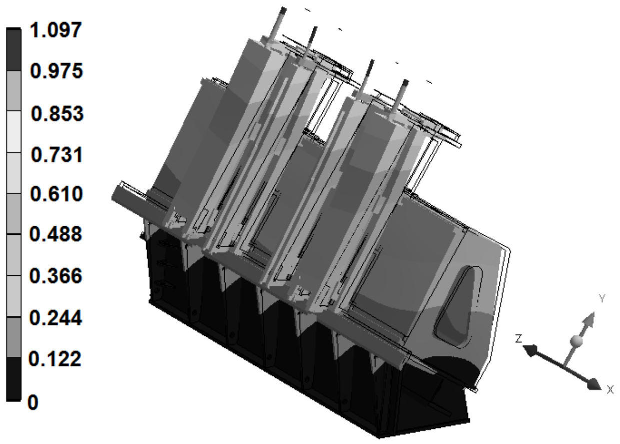

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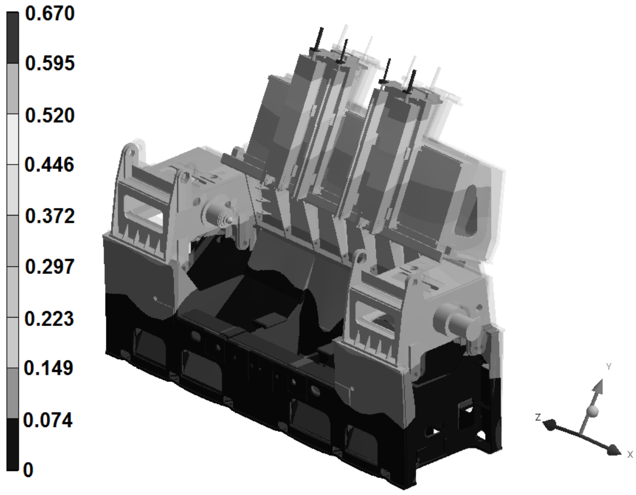

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Fig. 8. Modal FEA results for the machine tool components' after modification of the carriage and upper part of the bed: a) slide f1=168.5 Hz, b) carriage with upper part of the bed f1=71.3 Hz, c) whole machine tool f1=37.8 Hz

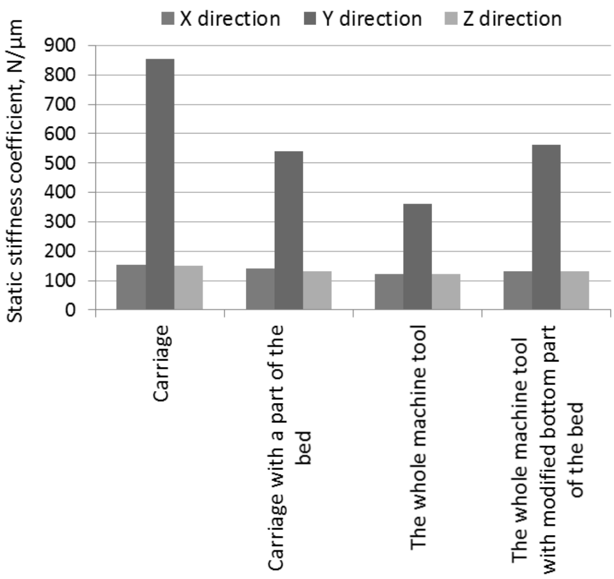
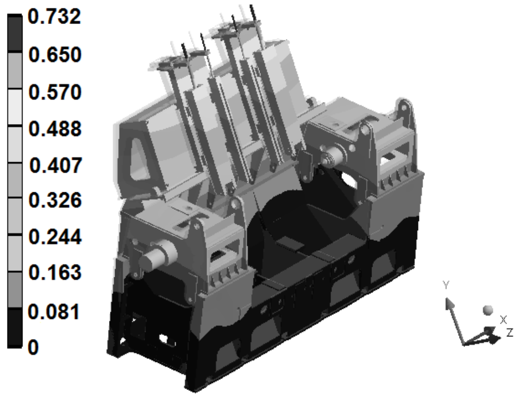


Fig. 9. Comparison of stiffness coefficients of fragments of a modified lathe structure



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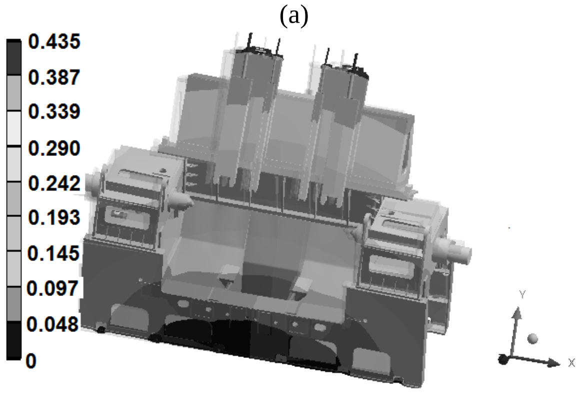

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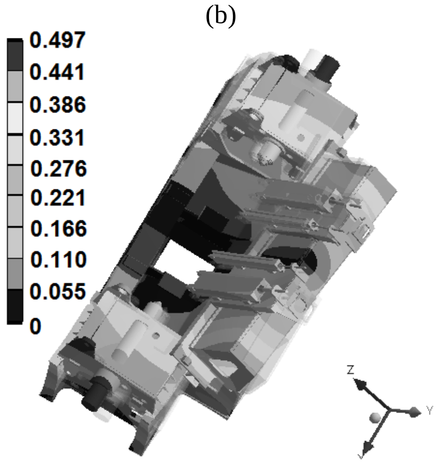

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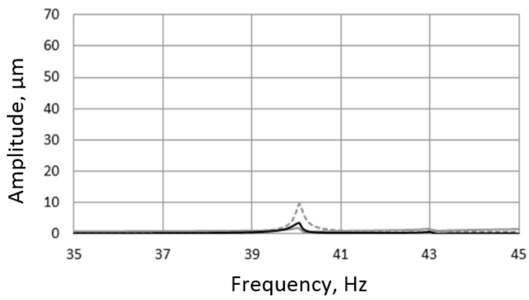
Fig. 10. Modal FEA results for the lathe with modified carriage, upper and main part of the bed: a) f1=42.3 Hz, b) f2=54.3 Hz, c) f3=67.3 Hz

3.2 The response of the lathe model to harmonic force

The paper [3] presents issues related to data preparation and performing modal and harmonic analysis in ANSYS software. Based on this information and on the logarithmic damping decrement (Table 1) and the first natural frequency, damping coefficients specific to different materials of the support structure were introduced in the analyzed wheel lathe model. For numerical analysis, a force of 200 N was assumed to be applied to the tool's blade independently in each of the three directions (Figure 6(a)). The response of the system was also checked on each of the three directions regardless of the direction of the force. The results obtained are presented in Figures 11, 12 and 13. The

original design and the whole machine tool after the first stage of modification, i.e. after modification of the carriages and the upper part of the bed, were taken into account.
The analysis of the results shows that the introduction of the design modification resulted in a desired decrease in the vibration amplitude for the X and Y directions and an unfavorable, almost 100 % increase in the amplitude for the Z direction. Therefore, the second stage of modification and subsequent simulations were carried out, changing the damping properties of the main (lower) bed body. In the first place, the material model was changed to the one suitable for grey cast iron, while doubling the thickness of walls and ribs of this body. The increase in thickness was dictated by technological reasons, concerning the minimum thickness of walls and cast ribs and almost twice as low a modulus of elasticity. In the next step, an attempt was made to simulate a hybrid body, introducing polymer concrete filling of the lower part of the bed made of steel. The obtained body strength corresponds to the steel body, in terms of stiffness it is better than the steel body, and in addition, it should definitely better dampen vibrations (Figures 11, 12 and 13).
As a consequence of adopting the bottom part of the bed in the form of a hybrid body, an increase in the natural frequency of vibrations by about 40% in relation to the original design was obtained. This is due to the increased stiffness of this part of the machine tool. A significant decrease in vibration amplitude was also obtained, and thus an increase in dynamic stiffness from approx. 3 times in the Z direction to approx. 9 times in the X direction, in relation to the original machine tool supporting structure (Figure 14).



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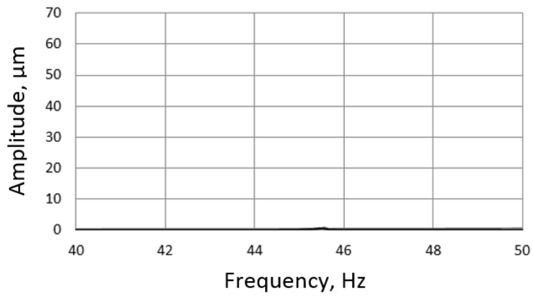

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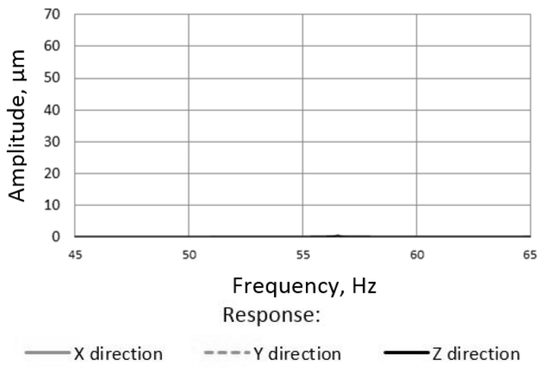

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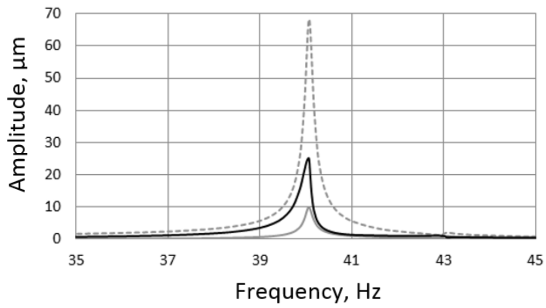
Fig. 11. Machine tool response to the harmonic force of 200N in the X direction: a) original machine design; b) after modification of the carriage and the upper part of the bed and with the lower part of the bed made of cast iron; c) after modification of the carriage and the upper part of the bed and with the lower part of the bed filled with polymer concrete



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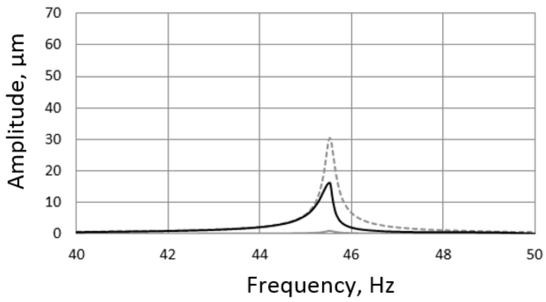

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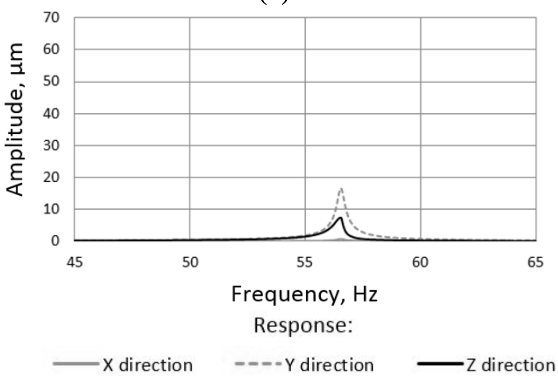

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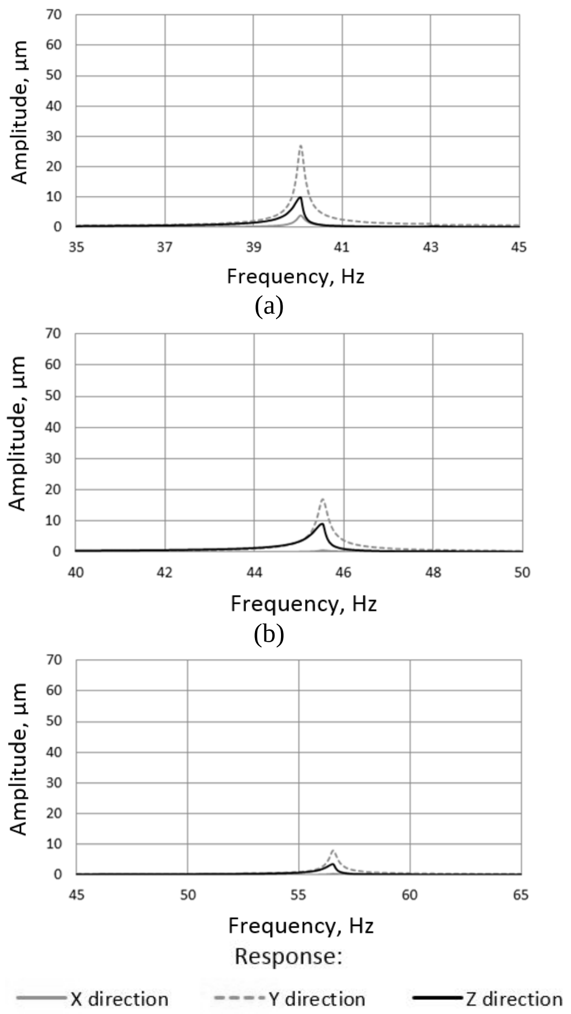
Fig. 12. Machine tool response to the harmonic force of 200N in the Y direction: a) original machine design; b) after modification of the carriage and the upper part of the bed and with the lower part of the bed made of cast iron; c) after modification of the carriage and the upper part of the bed and with the lower part of the bed filled with polymer concrete



(c)

Fig. 13. Machine tool response to the harmonic force of 200N in the Z direction: a) original machine design; b) after modification of the carriage and the upper part of the bed and with the lower part of the bed made of cast iron; c) after modification of the carriage and the upper part of the bed and with the lower part of the bed filled with polymer concrete

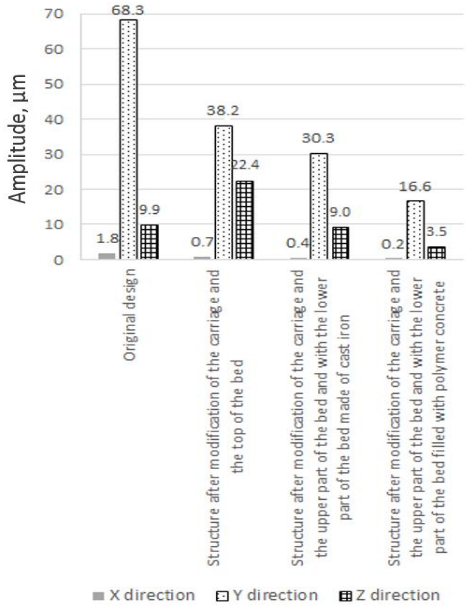


Fig. 14. Results of the FEM analysis of the harmonic response read on the force direction

4. CONCLUSIONS

The conducted FEM studies show that modifying an existing machine tool design to improve its properties is very difficult and indicate the advisability of making numerical analyses even at the machine tool design stage. Based on the simulations, the possibility of obtaining higher static rigidity of the wheel lathe after introducing structural changes to the carriage and the upper part of the bed was shown (Figure 6). An increase in static stiffness was observed mainly in the radial direction. This increase is about 75%. On the other two directions an increase in stiffness of about 10% was found. Additional increase of static stiffness is given by modification of the lower part of the bed. It increases stiffness in the radial direction by about 175% (Figure 9), but does not significantly change stiffness in the other directions.

It was further observed that the design changes of the carriage and the top of the bed only slightly affect the change of the first natural frequency (results in Figure 7 and Figure 8). The observed increase in this frequency is about 4%. Modification of the lower part of the bed affects the change of this natural frequency to a greater extent. It causes an increase of 16% in comparison to the original design (Figure 10).

This range of FEM analysis, however, does not allow to state clearly whether the introduced changes will improve the quality of the machined surface. Therefore, it was decided to perform a harmonic FEM analysis. The presented results show that such analyses are necessary at the design stage. As a result, it was found (Figure 14) that the changes to the carriage and the upper part of the bed should reduce the vibration amplitude by 60% in X direction (feed direction) and about 44% in Y direction (radial direction). Even better result can be obtained by additional filling of the lower part of the bed with polymer concrete. In this case, the vibration amplitude decreases by about 89% in X direction and about 76% in Y direction. A large decrease in vibration amplitude is also possible by making the lower part of the bed as cast iron. For this case, the vibration amplitude decreases about 78% in the X direction and about 56% in the Y direction. Based on this, it can be concluded that the surface quality can be improved after the design changes of the wheel lathe.

In conclusion, it was found that the hybrid bed in the form of a steel body filled with polymer concrete gives the best damping properties for the whole machine tool. Thus, it is the best solution for improving the machined surface quality.

The machine tool design proposed as a result of the modifications is certainly not an optimal one, which is mainly due to the fact that the proposed changes were largely limited by the existing form of the machine tool. Better effects could be achieved by conducting numerical analyses still at the initial design stage and taking into account the form and parametric optimisation of FEA [7].

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Received: March 29, 2021 / Accepted: June 15, 2021 / Paper available online: June 20, 2021