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IDENTIFYING THE CAUSES OF DETERIORATION IN THE SURFACE FINISH OF A WORKPIECE MACHINED ON A RAIL WHEEL LATHE

IDENTYFIKACJA PRZYCZYN POGORSZENIA STANU POWIERZCHNI OBRABIANEJ NA TOKARCE KOŁOWEJ*

Operational problems often remain undetected until a machine is commissioned and first machining is attempted. Heavy-duty machines are a specific group of machine tools due to the character of their manufacturing process. As they are often manufactured as single units, which involves high production costs, there are no prototype versions built and no tests are performed on them. Therefore, before the machine is complete, computer simulation methods are often the only validation tools of a machine project at the stage of designing. The variety of applications and the individuality of production are the reasons for the lack of standards defining the rigidity and precision of the cutting process performed by heavy machine tools. In this case analysis, the authors are considering a heavy duty rail wheel lathe, in which some issues were found during its exploitation which make it impossible to achieve the required shape, dimensions and surface finish, while working at set parameters. This article presents a comprehensive approach to the identification of the form and frequency of a machine tool supporting structure's self-vibrations and their potential sources in the case study of a horizontal lathe for railway wheelsets. The authors, drawing on the results of their long-standing research and their experience in the field of heavy machine tool design and testing, indicate self-excited vibrations as a key factor machine's operational behaviour, which is rarely considered in this type of machines.

Keywords: *self-excited vibrations, modal analysis, finite element method, machine tool CNC.*

Problemy eksploatacyjne są często wykrywane dopiero po uruchomieniu maszyny i po pierwszych próbach obróbki. Obrabiarki ciężkie są specyficzną grupą maszyn do obróbki ze względu na charakter ich procesu produkcyjnego. W procesach produkcyjnych tego typu maszyn, ze względu na jednostkowy charakter produkcji i koszty, nie buduje się wersji prototypowych i nie wykonuje się na nich testów. Tym samym, przed wersją ostateczną, metody symulacji komputerowych są często jedynymi narzędziami walidacji projektu na etapie projektowania. Różnorodność zastosowań i indywidualność produkcji są przyczyną braku opracowanych norm określających sztywność i precyzję obróbki wykonywanej przez ciężkie obrabiarki. Autorzy rozpatrują przypadek tokarki ciężkiej do zestawów kół kolejowych, w której podczas eksploatacji stwierdzono pewne problemy, które uniemożliwiają wytwarzanie przy zadanych parametrach w celu osiągnięcia pożądanego kształtu, wymiarów i jakości powierzchni. W artykule przedstawiono kompleksowe podejście do identyfikacji kształtu i częstotliwości drgań własnych konstrukcji nośnej obrabiarki oraz ich potencjalnych źródeł, na przykładzie poziomej tokarki do zestawów kolejowych. Autorzy w swoich badaniach zgodnie z uzyskanymi wynikami i ich doświadczeniem z zakresu projektowania ciężkich obrabiarek i badań podkreślają drgania samowzbudne, które są rzadko brane pod uwagę w tego typu maszynach, ale mają znaczący wpływ na zachowanie modalne maszyny.

Słowa kluczowe: *drgania samowzbudne, analiza modalna, metoda elementów skończonych, obrabiarka CNC.*

1. Introduction

Due to the general tendency to increase machining efficiency and the evolution of machine tools toward the so called 'High Speed Cutting' direction, manufacturers of machine tools face new challenges. These are associated with the need for designing machine tools, where the engineering cannot be based on past experience and the tried and tested design solutions are no longer valid. Heavy-duty machine tools are a specific group of machine tools due to the nature of their production process [22, 23]. Bearing in mind a limited demand, the design and construction process as well as the manufacturing process are either small-scale or unit-intensive. As these machines are often manufactured as single units, which naturally results in high manufacturing costs, there are no prototype versions built and no tests are performed on them prior to the commissioning. Therefore, computer simulation methods are often the only validation tools available at the design stage, until the machine is complete. The variety of applications and the individuality of production are the reasons for the lack of standards, which would define the rigidity of the machine and the precision of the machining performed by heavy machine tools. The lack of prototype testing results in the fact that no corrections are made to wrongly designed solutions. Tight deadlines for new orders forces engineers to rely on tried and tested solutions, which are not always the optimal ones. Operational problems often remain undetected until a machine is commissioned and the first machining is attempted. The author's experience shows that these problems are often connected with forced and self-excited vibrations, which limit the machine's capabilities to meet the required cutting parameters. This situation also occurred in the case of the lathe discussed in this study. The dynamic properties of machine tools have a significant influence on the cut-

(*) Tekst artykułu w polskiej wersji językowej dostępny w elektronicznym wydaniu kwartalnika na stronie ***www.ein.org.pl*** ting process and are considered among the most significant for the evaluation of a machine tool's structure. They are often described by frequency characteristics which allow for evaluating the stability of the machine-process system. The machine's dynamic properties may influence the dimensional accuracy and the surface roughness of a workpiece [24]. The analysis of the dynamic properties of a cutting machine's support structures should be one of the basic steps of a machine tool construction process. This article presents a comprehensive approach to the identification of the form and frequency of machine tool supporting structure self-vibrations and their potential sources, using the example of a horizontal lathe for railway wheelsets.

The paper presents an innovative approach, unprecedented in the literature, to identify the causes of deterioration in surface finish of a workpiece machined on a rail wheel lathe. The approach is based on the use of numerical methods to determine the vibration frequencies of a workpiece being machined and of the machine components which are directly responsible for the accuracy of the cutting process.

2. Materials and methods

2.1. Characteristics of the research object

The smooth work of a rail with a wheelset requires a fixed profile and consistent quality of the wheelset contact surface. Specialised wheel and track lathes are used for this purpose. The application of a rail wheel lathe for the regeneration of the wheelset contact surface needs dismounting the wheelset. These machines require high precision and efficiency of the process due to the constantly increasing demands for precision of the running components. In the process of facing a wheelset, vibrations may occur as a result of e.g. uneven wear of the contact surfaces [6, 10, 16, 19, 22, 28, 29]. The rotation of an unbalanced wheelset may be a source of vibrations and instability of the cutting process. Furthermore, it may generate great cutting forces during the machining which are transferred to machine components. Consequently, a wheel lathe should have a rigid and vibration-resistant structure [5, 7, 15, 18, 21, 25].

The analysed machine belongs to the group of blind wheel lathes, i.e. both the entry and the departure of a wheelset is from the front of the machine. During the machining the wheelset is fixed at both ends by means of claws ejected from the tailstocks. In addition, the wheelset rests on two rollers on either side and is frictionally driven by a third roller pressed against it from the top. This solution is currently becoming more and more popular. The use of a driving roller prevents the formation of a notch, which commonly occurs with solutions using driving centres. Such notch may be particularly dangerous in case of high-speed railways. This method of fixing increases the accuracy of rotation and reduces the radial runout. Also, the forces acting on the locating centres are reduced.

In order to effectively identify the causes of excessive vibrations in the lathe, an analytical and numerical analysis of vibrations in the "lathe – cutting process" system were applied. A hypothesis was formulated that a loss of stability of the "lathe – cutting process" system, i.e. the occurrence of self-excited vibrations was the cause of excessive vibrations. The analytical solution, that is increasing the limit of stability, requires determining the frequency of the self-excited vibrations in the first place, e.g. by solving the identity $Im[K(j\omega)]=0$, where: w – pulsation of self-excited vibrations, $K[(j\omega)]$ transmission of the open system, and then by determining the stability reserve. The stability reserve may be changed, e.g. by decreasing the dynamic susceptibility of the mechanical system or by changing the reinforcement coefficient in the cutting process. This classical method is tedious and difficult in the analytical procedure. Thanks to the numerical method (FEM) it was possible to identify the probable frequency of self-excited vibrations and test changes in the dynamic susceptibility of the mechanical system resulting from structural changes proposed.

2.2. Lathe model

Machining of a wheelset while maintaining the required cutting parameters showed the occurrence of vibrations in the LCWC system: (lathe, chuck, workpiece and cutting tool). In consequence, it was impossible to obtain the required machining accuracy and the resulting surface finish showed high waviness and roughness (Fig. 1).

Any attempts to identify the causes of this situation did not produce desirable results. Also, experimental studies according to [12], to determine the vibration frequencies occurring during the machining have been carried out (Fig. 2, Fig. 3).

An analysis of the kinematic chain of the propulsion system excluded the possibility of vibrations being induced in the propulsions. In order to determine the causes of vibrations and methods to counteract them, the finite element method was applied. A number of numerical analyses according to current trends in numerical simulations [2, 3, 4, 13, 14] were carried out on the lathe model for the evaluation of its static rigidity, the form and frequency of self-excited vibrations and the response of the system to harmonic extortion.

2.3. FEM model

In order to determine the dynamic properties of the lathe model ANSYS software has been used. For simulations the modal analysis module has been applied. With this module the first modal shapes and the corresponding frequencies have been designated. Discrete models of the wheel lathe have been developed basing on a CAD 3D model. The support has been designed independent as a FEM model. All of

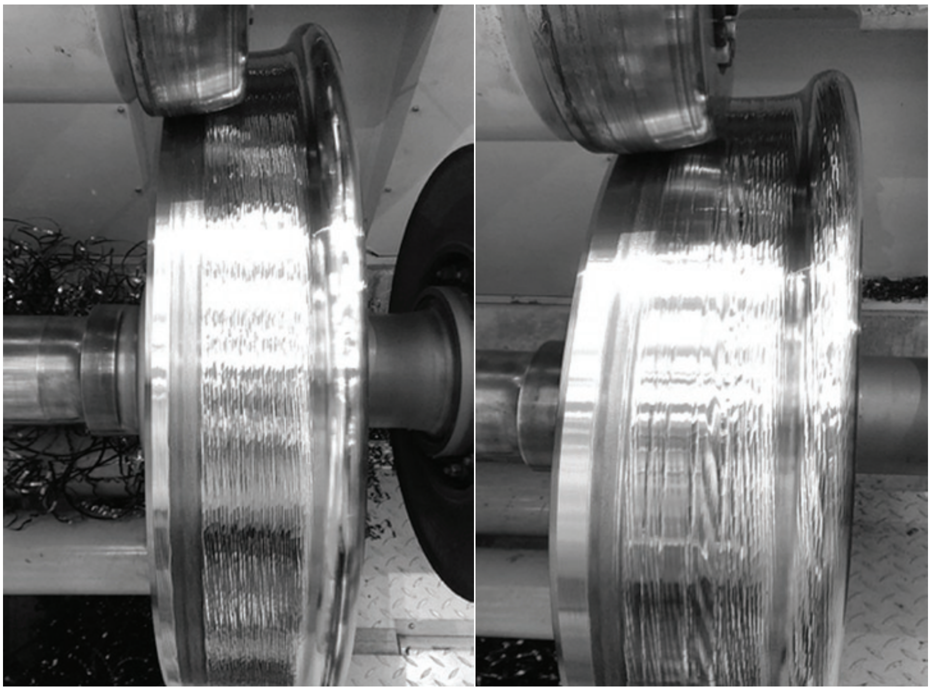

Fig. 1. Comparison of surface finish quality for different machining parameters A) A = 5mm, S= 99m/min, f = 1,5 mm/rot, d = 856mm B) A = 4mm, S= 90m/min, f = 1,5 mm/rot (21%) d = 856mm.

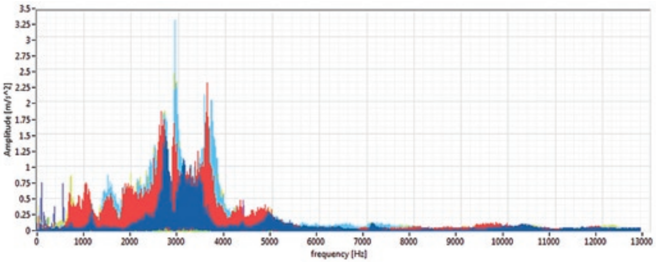

Fig. 2. Frequency spectrum of signals recorded on the milled diameter under load

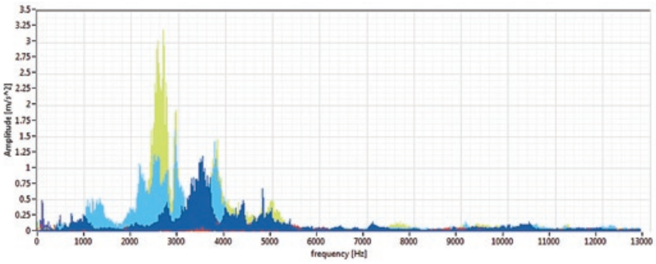

Fig. 3. Frequency spectrum of signals recorded at the edge under load

Table 1. Material properites for FEM analysis

Property	Structural steel
Young's modulus [MPa]	$2*10^5$
Poisson's ratio	0.3
Density [kg/m^3]	7850

the models have been designed as solids. Discretisation of models has been performed basing on finite 3D eight-node elements of HEXA type and four-node of TETRA type. The total number of finite elements of the entire lathe model with a wheel set equals to 949791 with 4092309 nodes. Views of the lathe model after discretization are shown in the figure (Fig. 4). The method of adopting the boundary conditions for the model of the entire lathe resulted from its foundation. Therefore, all degrees of freedom have been taken at the foundation of the bed. Due to the fact that the lathe bodies have been made as steel welded, the same material properties (appropriate for steel) for all elements of the models were adopted (Tab. 1).

3. Results and discussion

First of all, a modal analysis of the wheelset itself was performed by determining the frequencies and forms of self-excited vibrations for the support in the chuck (Fig. 5). The first of the bent forms is characterised by a frequency similar to the one obtained in experimental studies. The support in the chuck and accounting for the construction of the tailstocks should cause a decrease in the frequency and a better match with the results obtained during the experiment.

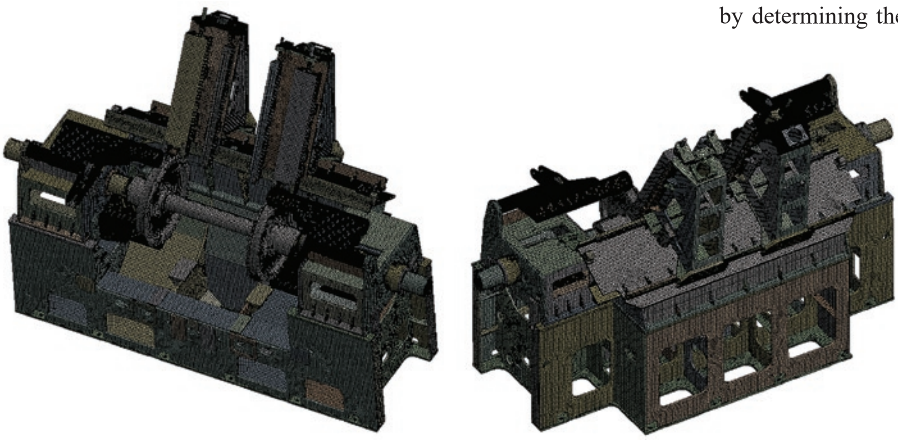

Fig. 4. FEM model of the analysed lathe

Subsequently, a modal analysis of the machine with a mounted wheelset was performed. As a result of the analyses (Fig.6), the same frequency of vibrations of the supports, the tailstocks and the wheelset were found for the first two forms of self-excited vibrations. The frequencies found are lower than those determined for the wheelset itself, which is due to the susceptibility of the wheelset supporting and locating system.

As a result of the analyses, it was also found that there was no effect of the change of the sliders' position on self-excited frequencies. It can be observed that all of the eight resonance frequencies identified coincide with the results of experiments, and certain modal shapes may have a negative influence on machining accuracy.

The results of this analysis (Fig. 7) show inadequate rigidity of the structure. Despite the relatively high static rigidity of the supports, as measured at the cutter mounting site, they are a weak link in the structure. This is a result of their columnar structure, with the centre of

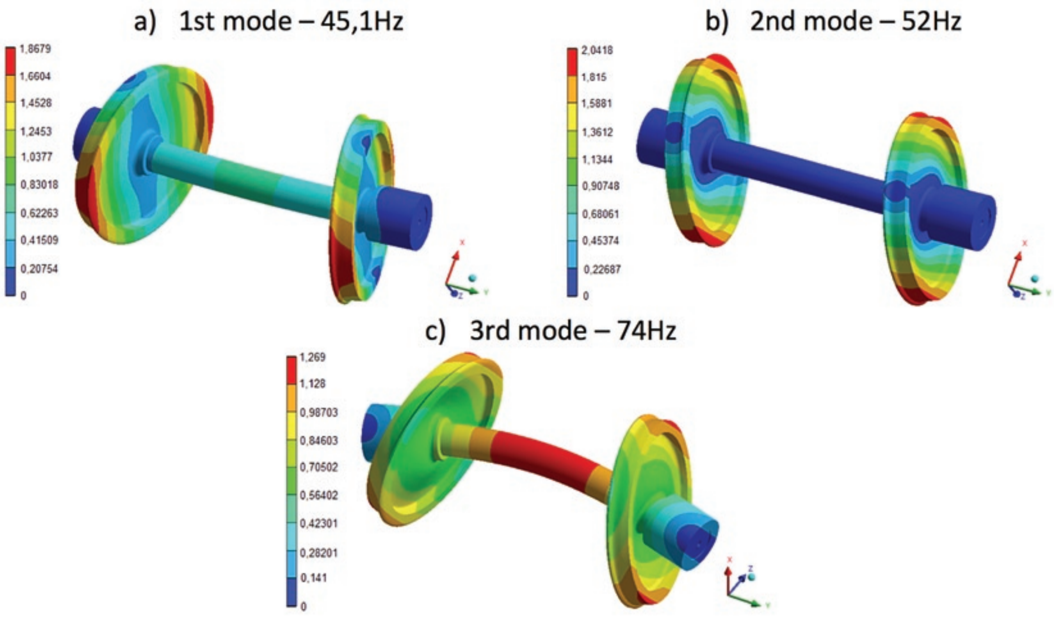

Fig. 5. The first three mode shapes corresponding to the frequencies of the wheelset: a) f1=45,1 Hz; b) f2=52,0 Hz; c) f3=74,1 Hz

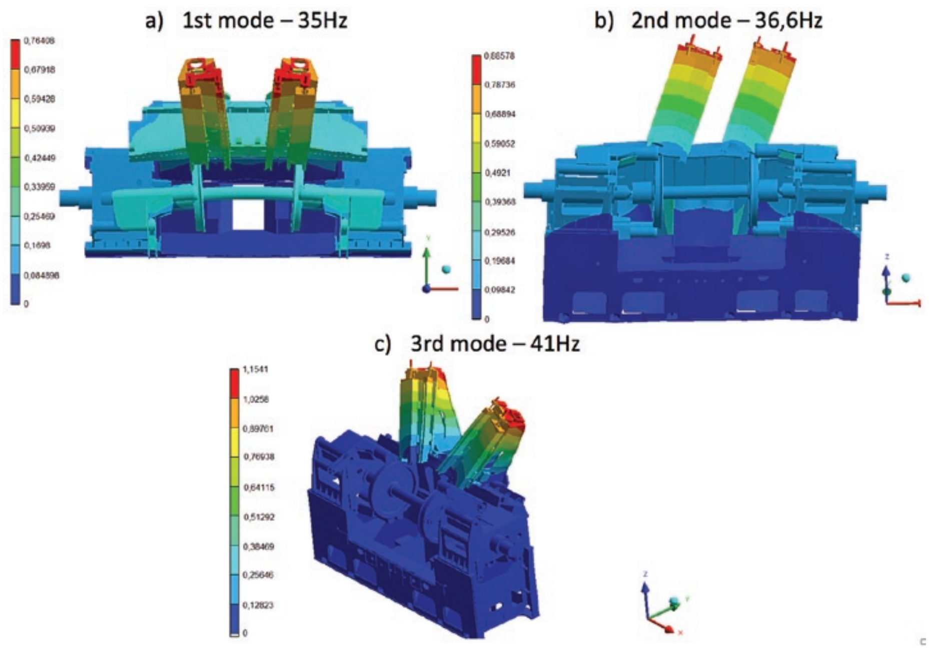

Fig. 6. The first three forms of vibration of the wheel lathe corresponding to the frequencies: a) f1=35 Hz; b) f2=37 Hz; c) f3=41 Hz

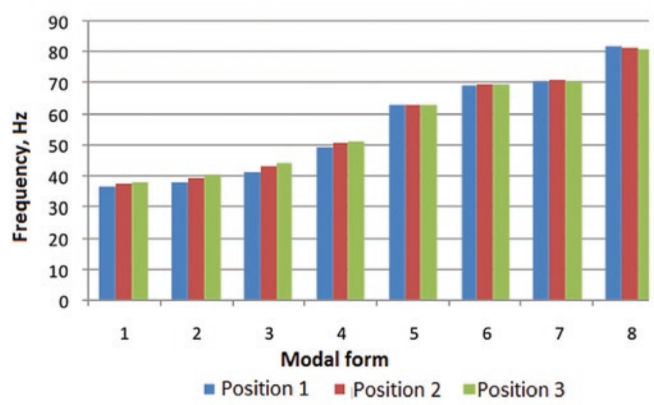

Fig. 7. Determination of the frequency of the lathe's own vibrations for different slider positions

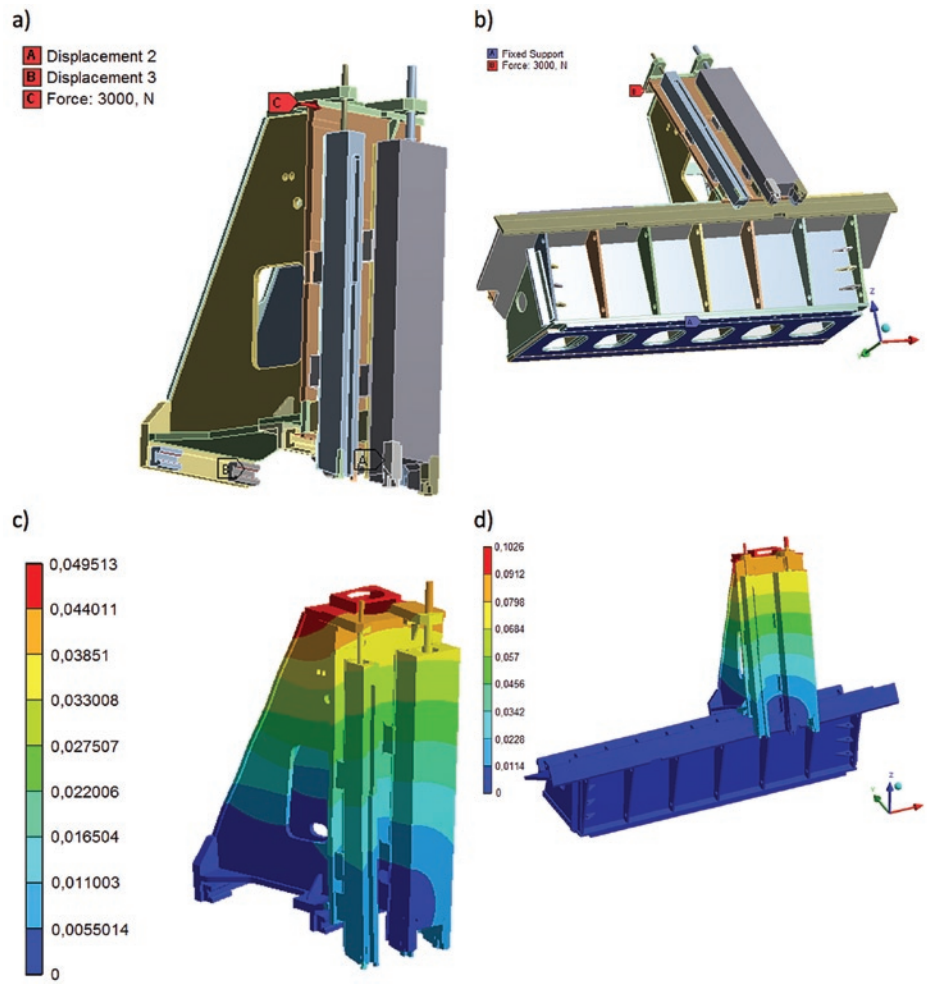

Fig. 8. The results of analyses of displacements to the force 3kN applied on top of the carriage: a) b) load and restraint conditions, c) d) resultant displacements

gravity located high, thus decreasing the frequency of their own vibrations.

The susceptibility of the bed construction is also significant. Assuming a load on the upper part, the bed itself increases the susceptibility by ca. 50% in relation to the support (Fig. 8). The vibrations identified during operation may be forced vibrations. However, in order to take forced vibrations into consideration, their source would have to appear in the system first. It seems that the cause of such vibrations could only be the roughness of the wheelset being machined. It all points to the presence of self-excited vibrations being a result of the cutting process.

In response to the results obtained and the analysis of results of studies carried out on the actual lathe, four ways of improving the operation properties of the machine were proposed. First, the cutting parameters may be modified. However, this would result in reduced efficiency of the process and would not necessarily improve the existing situation because of the very closely situated next main own vibration frequency of the machine. Introducing a vibration eliminator in the form of an additional mass of properly selected damping could be another solution. However, as shown by subsequent numerical analyses of the machine, this solution would not significantly change the vibration frequency, only slightly altering their amplitude (Fig. 9, Fig. 10). As a result, we remain within the resonant frequency range.

The third solution involves structural changes made to the upper part of the bed and a change of the supports' structure. This may significantly improve the machine's operating properties. However, this requires a design project, a numerical analysis of the proposed solution and a shutdown of the machine to make it available for introducing changes in its structure. The final proposal also involves an intervention in the existing structure by filling in selected body parts with polymeric concrete. As a result, we would not obtain a significant change in frequency, so the machine would continue to work in its resonance range, but we should reduce the vibration amplitude even by several times.

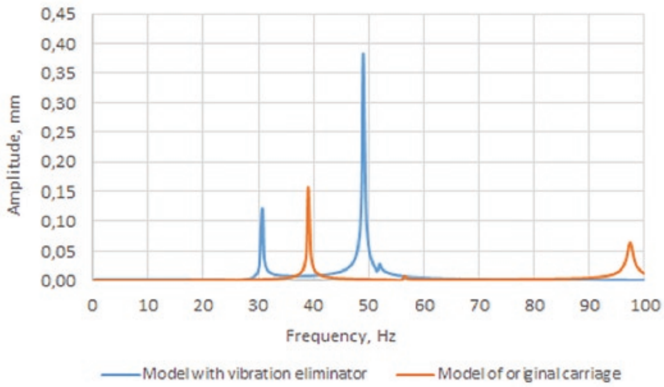

Fig. 9. Frequency response of the system to the harmonic exortion for a point at the top of the slider

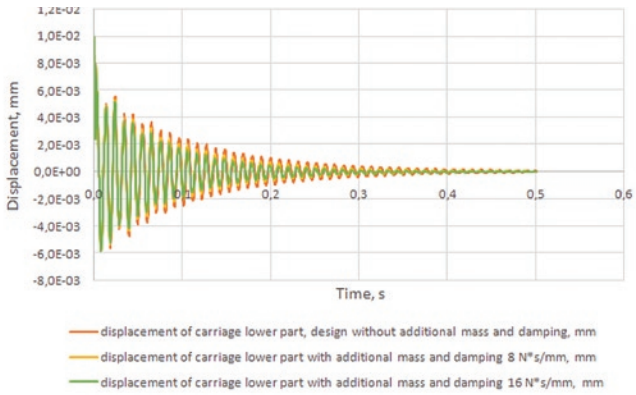

Fig. 10. Displacement of a point in the lower part of the slider to force the harmonic exortion.

4. Conclusions

This attempt to improve a wheelset lathe operation accuracy shows that it is a very difficult task at the exploitation stage. First, experimental research is required, e.g. an analysis of vibrations occurring during the machining. In the next step, it is necessary to develop a model and carry out some numerical analyses to obtain reference results for further analyses. With the data collected (including the machine's own frequencies and rigidity indexes) it is possible to begin structural modifications. At this stage, the possibilities are very limited and the results obtained will not always be satisfactory. Therefore, introducing a new design solution or improving the current machining parameters should be done at the design stage. This may help to reduce or avoid the machining issues described in this article.

Dynamic extortions in wheel lathes generally have low frequency as rotational speeds of spindles are in the order of 1–2 Hz. Therefore, the occurrence of vibrations in the range of several dozen Hz cannot be interpreted as forced vibration. The causes for vibrations in this range should be sought in the loss of stability, that is: the occurrence of self-excited vibrations [11, 20, 22]. Self-excited vibrations are created in closed systems (Fig. 11a, b), where apart from the mechanical system, there is also a cutting process. Their appearance depends both on the dynamic susceptibility of the mechanical system W(jω) and on the model of the cutting process KPS(jω) [9, 17, 24].

According to the Nyquist criterion, a loss of stability (which is equivalent to the occurrence of self-excited vibrations) happens when the spectral characteristics of an open system $K_o(j\omega) = W(j\omega) K_{PS}(j\omega)$ does not include point (-1, jω) (Fig.11c), that is, when the inequality $K_o(j\omega) = W(j\omega)\ K_{PS}(j\omega) > -1$ ($K_o(j\omega)$ characteristic is negative in the frequency range of self-excitation). Therefore, if the dynamic susceptibility of a mechanical system is high, e.g. due to low static rigidity, the stability condition may not be fulfilled and self-excited vibrations occur.

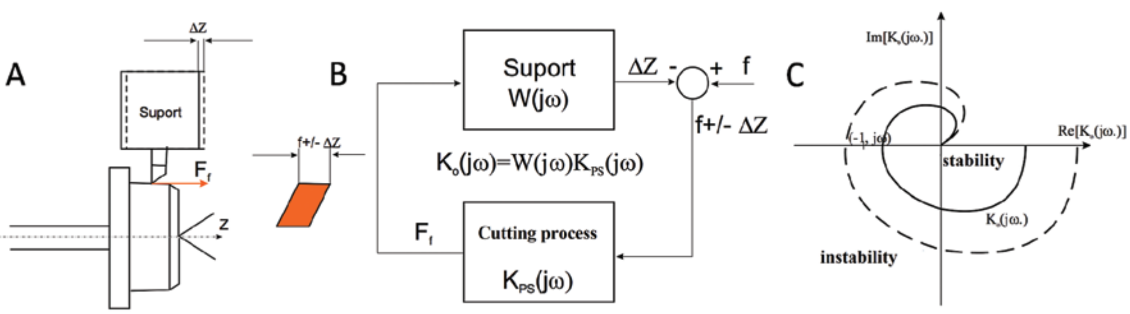

Fig. 11. Self-excited vibrations diagram. A) B) Self-excited vibrations system C) Frequency response to stability loss

A characteristic feature of self-excited oscillations is that their frequency is close to one of the mechanical system's own vibration frequencies. If such vibrations occur, the natural way to eliminate them is a structural change which leads to a change in the self-excited vibration frequency. For an existing machine, it is practically impossible. Then there are other solutions left [1,8] which can be named as technological. They involve changes in the machining parameters, i.e. changes of $K_{PS}(j\omega)$.

Since the frequency of self-excited vibrations is close to one of the mechanical vibration frequencies of the mechanical system, the modal analysis allows for its identification. The classical method of determining the frequency of self-excitatory vibrations is to solve the condition $Im[K_o(j\omega_0)] = 0$, where ω_0 represents self-excited vibration pulsation, but this method requires the knowledge of the dynamics of the cutting process $K_{PS}(j\omega)$.

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