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Design of an Ultra-Light Portable Machine Tool

PAWEŁ DUNAJ, MICHAŁ DOLATA, BARTOSZ POWAŁKA, PIOTR PAWEŁKO, AND STEFAN BERCZYŃSKI
Faculty of Mechanical Engineering and Mechatronics, West Pomeranian University of Technology, 70-310 Szczecin, Poland
Corresponding author: Paweł Dunaj (pawel.dunaj@zut.edu.pl)

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ABSTRACT Designing machine tools for in situ machining is a challenging task due to their unique structures and restrictive functional requirements. Such a machine tool should be characterized by low mass, adequate machining accuracy and high machining stability. The paper presents the design of an ultra-light, axisymmetric, portable machine tool for in situ flange face milling. Due to a high compliance of the designed machine tool and a need of maintaining its low mass, structural modification aimed at stability increase were highly limited. Therefore, it was decided to select the spindle ensuring machining stability. The selection of the spindle was supported by finite element analysis. Based on numerical analyzes results a prototype with a proper spindle was build. Then, the accuracy of the finite element model and the predicted stability were experimentally verified, showing a good agreement with the real counterpart. Finally, two general conclusions were formulated: (i) in the case of machine tools characterized by high compliance and limited possibility of modifying their design a good choice may be the selection of a spindle that allows to obtain parameters that ensure stable machining, and (ii) it is possible to build low-dimensional, reliable finite element model without using substructuring or reduction methods, and a well-thought-out discretization and replacement elements of complex load bearing systems instead.

INDEX TERMS Dynamic stiffness, finite element modeling, flange facer, in situ machining, onsite machining, portable machine tools.

LIST OF USED SYMBOLS

a_p	depth of cut [mm]
j	imaginary unit
n	spindle speed [RPM]
R_a	arithmetical height of profile [μm]
R_p	maximum profile peak height [μm]
R_t	total height of profile [μm]
R_z	maximum height of profile [μm]
V_f	feed rate $[mm/min]$
z	teeth number
η	loss factor
ω	frequency [rad/s]
ω_c	chatter frequency [rad/s]
$\boldsymbol{C}$	damping matrix
$\boldsymbol{K}$	stiffness matrix
$\boldsymbol{G}_{CP}$	machining process matrix $[N/mm^2]$
$\boldsymbol{G}(\omega)$	frequency response function matrix [mm/N]

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I. INTRODUCTION

The demand for machining large parts occurs in the aerospace industry [1], traditional and renewable energy industry [2], in the mining industry, etc. The classic method of machining large parts is the use of large-scale machine tools [3]. These machines must provide adequate machining space, and it is necessary to transport the workpiece to a specialized workshop [4]. These activities are associated with high energy consumption and time necessary to dismantle parts and provide transport [5]. As noted in [6], the concept of placing a large-size workpiece inside the machining space of a machine tool has its natural limitations.

The alternative are portable machine tools which can be used for in situ machining of large parts [7]. Bringing the machine to the part allows for time and energy savings. It also significantly reduces the costs of transportation. After a machine tool has been built, it is very difficult or even impossible to introduce structural changes to improve the performance of this machine. Hence the need for a reliable modeling process that results in a feasible design of a portable machine that fulfills its functional requirements [8]. One of the basic criteria adopted when evaluating a machine tool is its dynamic stiffness [9]. It directly affects the accuracy of machining and machine stability limits. Dynamic stiffness of machine tools is usually computed using a finite element (FE) model of the machine [10].

The development of an accurate machine tool model mapping the actual geometry of the support and guide system and modeling the contact layer, involves models with a large number of degrees of freedom (DOF), which increases the computation time [11]. The dimensionality of the model is particularly important if the machine model will be analyzed in terms of many variable positions of its elements [12]. In the case of portable machines, due to their design solutions, the stiffness may vary depending on the instantaneous relative position of the kinematic pairs [13].

Frequently the problem of dimensionality is solved by the application of reduction and substructuring methods [14], [15]. In [16], authors used spring-damper elements to couple reduced models of machine tool components. However, the process of model reduction itself was not presented.

Many researchers use the Craig-Bampton method to reduce models of connected components [17]. Applying this method to a variable configuration system can be challenging due to the necessity to keep many boundary nodes. In [18], the authors proposed a residual structure to which the reduced components were attached, and which should consist of elements changing their configuration during the movement of the machine tool, such as e.g. a ball screw drive. This kept the reduced component models independent of the machining process.

In [19], authors proposed the application of a reduced model substructural synthesis for rapid evaluation of position-dependent dynamics of machine tools. The method employed component mode synthesis concept to reduce the DOF of machine tool component models. The assembly of the substructures was performed using a constraint formulation, owing to which it was possible to significantly reduce the degrees of freedom of the model.

Garitaonandia *et al.* [20] applied a two-step procedure based on CMS reduction. In this procedure, the boundary DOFs of two assembled components are extracted from the boundary set of the assembly. Then the constructed assembly was connected to another substructure. The procedure was repeated until completing the whole assembly considered as the component.

The Craig-Bampton method was applied by Eguia *et al.* to build a reduced model of the portable machine [8]. The reduced model was suitable for the time domain analysis that enabled design optimization with respect to machine accuracy. To achieve this objective, the method combined a reduced model of the machine and a mechanistic force model. The method was validated experimentally on a conventional three-axis milling machine.

The evaluation of a portable machine tool dynamics was performed by Law *et al.* [21]. They used frequency-based substructuring (FBS) to predict the frequency response function of the portable machine tool mounted to any part with known dynamics at connection points. The method described by Law *et al.* requires experimental tests on the object, which allow to identify the dynamic properties of components subjected to the substructuring process. In practice, it comes down to conducting impact tests of the final version of the machine, its components, and the workpiece.

The model describing dynamic properties of a machine tool can be used to assess its resistance to the occurrence of self-excited vibrations. Analyses of this type are performed at the machine tool design stage.

In [22] an offline compensation methodology, involving tool trajectory modification, based on the calculation of tool center point displacements from a prediction of machining forces and static stiffness characterization was implemented on a portable machine tool. The offline compensation was validated through a posteriori comparison with the nominal machining conditions.

A hexapod in situ machine tool was analyzed in [23]. The authors verified the attachment stiffness (i.e., of the feet attached to the surroundings with different materials) in the conventional dynamic model of the parallel manipulator, on the dynamic behavior of the Free-Hex machine tool dedicated to in situ operations. Overall, it was found that the attachment stiffness has an influence on the natural frequencies of the machine tool.

The paper presents a novel design of a portable machine tool along with the method of modeling such structures. During the modeling process, substructuring methods were not used to reduce the dimensionality of the model. Instead, it was proposed to use low-dimensional replacement models for modeling ball screws, linear guides, and bearings. In addition, special care was taken in terms of the selection of finite elements so that the dimensionality of the model could be as low as possible while maintaining the high accuracy and good quality of finite elements. Based on finite element model stability lobe diagrams were developed and a spindle ensuring stable machining was selected. The proposed axisymmetric structure of the machine tool allowed to maintain unchanged dynamics during the machining of large-size pipe flanges. Owing to this, the machining parameters can be selected easily while ensuring the machining stability.

The paper is structured as follows: Section 2 introduces the design of the portable machine tool. Section 3 describes how a finite element model of a designed machine tool was built. Next, a stability analysis was performed based on a finite element model of a portable machine tool. A proper spindle was selected based on the analysis of the stability lobe diagrams. Section 4 presents the prototype built and its experimental verification. Section 5 presents the discussion and main conclusions.

II. PORTABLE MACHINE TOOL FOR CYLINDRICAL SHAPE ELEMENTS

The presented portable machine tool is designed for drilling and milling axisymmetric surfaces of pipe flanges. The workspace is in the form of a spatial ring with an internal radius of 600 mm, an external radius of 1,200 mm, and a height of 500 mm. The kinematics of the machine is based on the main rotary axis (C) on which cross sledges (Y and Z) and an angular two-axis head with attached spindle (A_C and B_C) are mounted. The machine has five numerically controlled axes: three axes carrying out the main movements (C, Y, and Z) and two axes correcting the relative position and orientation between the spindle and the workpiece (A_C and B_C).

To achieve the lowest possible weight of the machine tool, its main body consists of bolted profiles and aluminum plates, and the movable bodies are made as a welded aluminum structure. The weight of the machine tool is therefore approx. 200 kg. The concept design of a portable machine tool with the workpiece is shown in Figure 1.

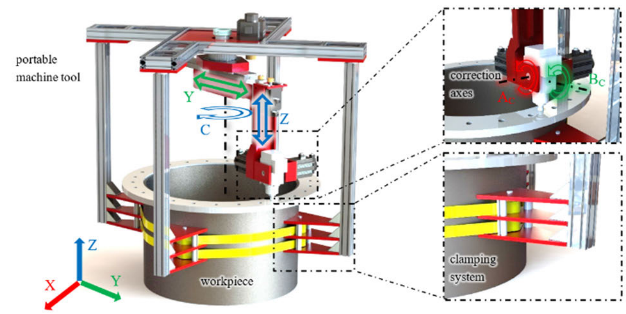

FIGURE 1. Concept design of a portable machine tool.

The distinguishing feature of the presented machine tool is its kinematic structure. The axial symmetry of presented machine tool ensures that its dynamic properties remain constant as the angular position around the main axis of the machine tool (C) changes. The idea is that this should enable unchanging machining parameters to be used to ensure stable machining.

Another distinguishing feature of the presented design is the non-invasive system of mounting the machine tool to the workpiece. This system consists of a set of belts and a system of rollers installed on the retaining feet. This ensures that the machine tool can be quickly assembled on different surfaces of the workpiece, thus shortening the time of pre-machining operations.

Positioning accuracy is ensured by the application of an automatic position correction system. This system uses a measuring probe mounted in the machine spindle to determine its position in relation to the workpiece. The measurement of the relative position of the machine tool and the workpiece (performed automatically by the machine) allows to determine the deviation of the Z axis of the machine tool from the main axis of the workpiece. This identification enables adjustments to be made to the machining program. Positioning the tool perpendicular to the workpiece surface when tilting the machine support structure requires the use of two numerically controlled rotary axes – A_C and B_C (the angle changes when the angle of the machine is changed). The idea of the operation of the machine tool positioning system in relation to the workpiece is presented in Figure 2.

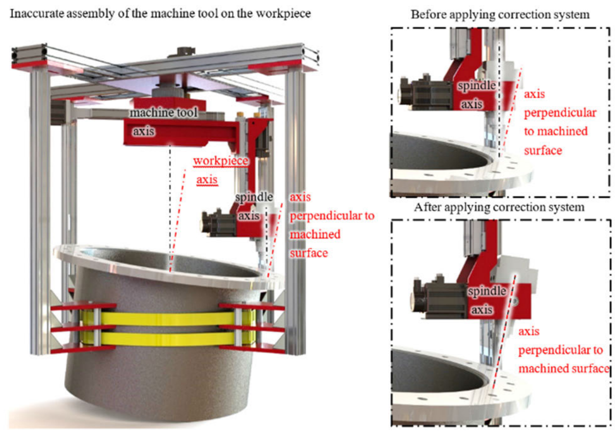

FIGURE 2. Diagram of the correction system operation.

III. FORMULATION OF A SUBSTRUCTURAL FEM MODEL

To assess the dynamic properties of the designed portable machine tool and to determine the conditions of stable machining, a finite element model was built.

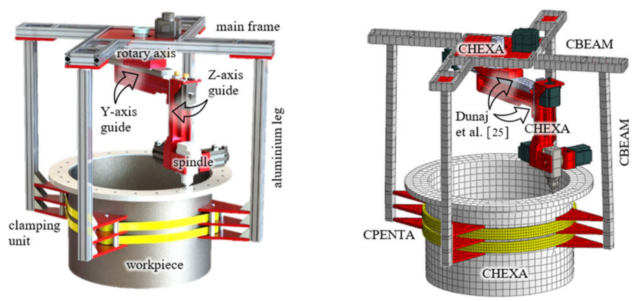

FIGURE 3. Discretized model of a portable machine tool.

The finite element model of the machine tool was built using the Midas NFX pre-processor [24]. The discretization of body elements marked in red in Figure 3 was carried out using eight-node, cubic, isoparametric finite elements (CHEXA) and six-node, five-sided, isoparametric finite elements (CPENTA). The applied finite elements were characterized by linear shape functions and three translation degrees of freedom in each node. Welded joints included in the body elements were modeled according to the method presented in [25].

Aluminum profiles included in the main body of the machine were modeled using beam elements (CBEAM), which were formulated based on the classic Euler-Bernoulli beam theory. These elements were characterized by six degrees of freedom in each node and linear shape functions.

The guiding systems of the Y and Z axes were modeled in accordance with the method contained in [26]. The feed drive systems were modeled in accordance with [27] and [28].

The discretization of electric motors, after appropriate simplification of their geometry, was carried out using CHEXA and CPENTA elements. The last element to be discretized was the tool spindle. As the constructed model of finite elements was to help in the selection of a spindle that would guarantee stable machining, they were modeled as a homogeneous solid representing substitute stiffness and inertia properties.

To describe the damping properties of the modeled machine tool a structural damping model was used, according to which the damping matrix C can be expressed as [29]:

$$\boldsymbol{C} = j\eta\boldsymbol{K} \tag{1}$$

where: $\boldsymbol{K}$ is the model stiffness matrix, j is the imaginary unit, η is the loss factor.

The loss factor η values describing the damping of connecting elements, i.e. guides, bearings and, bolted connections were estimated on the basis of isolated connections using the methodology presented in [25].

Then, for the model developed in this way, boundary conditions were defined which approximate the foundation of the actual structure To sum up, the model consisted of 37,737 degrees of freedom and 14,849 finite elements. The discrete model is shown in Figure 3.

For the defined model, using the Nastran Solver processor (SOL108), the receptance functions of the relative displacement between the tool (T) and the workpiece (W), on the feed direction and on the direction perpendicular to the feed were determined. The Z-axis receptance was not taken into considerations because of greater stiffness in this direction and smaller influence of vibrations along the Z-axis on dynamic cheap area. According to the adopted coordinate system, this resulted in the determination of the receptance function in the X and Y axis direction. Receptance functions were determined with 0.5 Hz step in the frequency range of 10-150 Hz. Such a range was selected after a series of preliminary tests allowing the assessment of the significance of amplitude functions of relative receptance.

In addition, the receptance function were determined to demonstrate axial symmetry for three representative angular positions, the selected positions are shown in Figure 4.

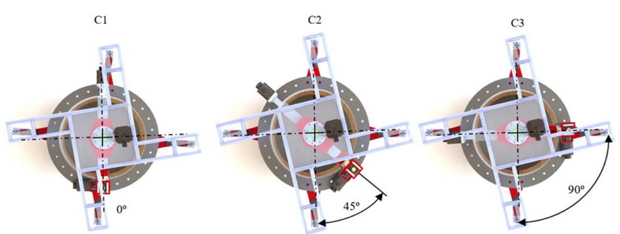

FIGURE 4. Selected configurations for which relative receptance functions were determined.

The results in the form of receptance functions of relative displacements of the tool and the workpiece for the defined configurations are shown in Figure 5.

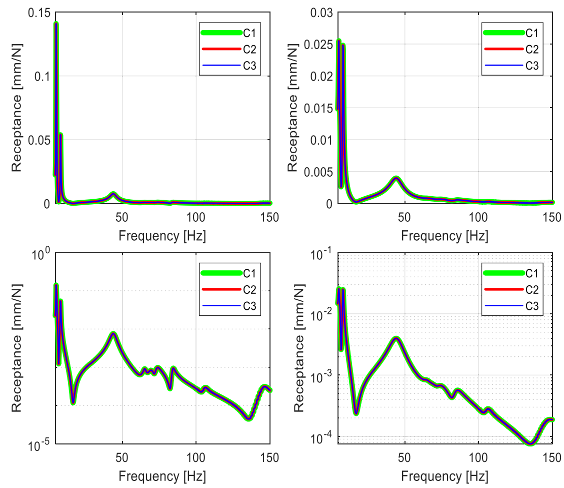

FIGURE 5. Amplitude of relative receptance functions.

As can be seen, and as presented in Figure 5, the receptance functions for the different angular positions of the machine tool do not show significant differences, so the thesis concerning the invariability of the dynamic properties of the machine tool with the rotation of the main arm of the machine tool around the C axis can be initially confirmed.

The fact that the dynamic properties remain unchanged at different angular positions is an important advantage from a technological point of view, as it allows the selection of fixed machining parameters which guarantee a stable process in the entire area of the workpiece. In addition, this observation allows to determine lobe diagrams indicating areas of stable and unstable machining for a single selected position. Therefore, it was decided to determine the lobe diagrams for the C2 position. The stability limit can be calculated by solving characteristic equation [30]:

$$det\left(I - a_p\left(1 - e^{-j\omega_c 60/(nz)}\right)\mathbf{G}_{CP}\mathbf{G}(\omega)\right) = 0 \tag{2}$$

where: a_p is the depth of cut, ω_c is the chatter frequency, n is the spindle speed, z is the tooth number, $\mathbf{G}_{CP}$ is the machining process matrix, and $\mathbf{G}(\omega)$ is the frequency response function matrix.

The stability calculations were carried out for two machine tool models differing only in terms of the spindle type. As indicated in Section 3, the spindles were modeled as homogeneous solids representing substitute inertia and stiffness properties. A low-speed hydraulic spindle and an electro-spindle were used in the analysis. This was dictated by the low weight, which is a distinctive feature of the designed machine tool. The mass of the hydraulic spindle was 7.5 kg and that of the electro-spindle was 7.2 kg. Both spindles were characterized by almost identical inertia parameters and therefore the receptance functions used for stability analysis (2) do not show significant differences. The main difference in the calculation for both options considered is the use of a 6- and 2-toothed cutter for the hydraulic spindle and the electro-spindle, respectively. The number of teeth results from the need to ensure a cutting speed appropriate to the cutting inserts used.

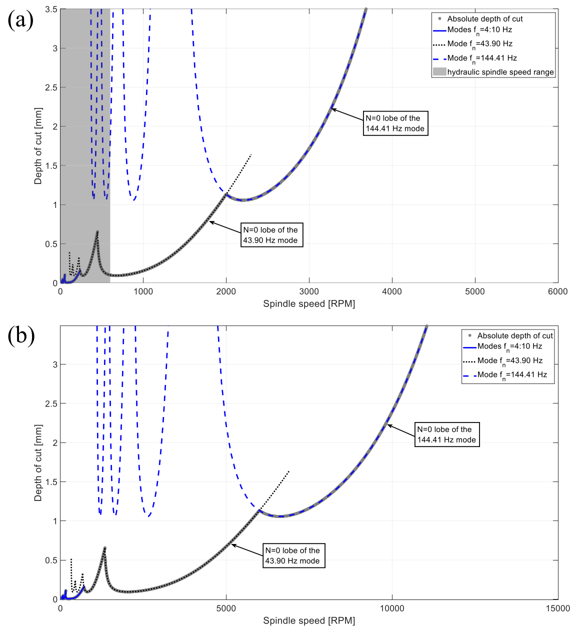

FIGURE 6. **Lobe diagrams determined for (a) the hydraulic spindle and (b) the electro-spindle.**

Figure 6a shows the rotational speed range for the hydraulic spindle (max. 600 RPM). In this speed range, stable machining is only possible with very small cut depths not exceeding 0.5 mm. Stability in this speed range is determined by low-frequency (<10Hz) mode shapes and the mode shape with a frequency of 43.9 Hz. Much greater machining capability is provided using the electro-spindle (Fig. 6b). High speeds ensure machining stability because they are beyond the impact of 43.90 and 144.41 Hz modes i.e., the operating speeds above $N = 0$ lobes of these modes can be reached (Fig. 6).

IV. EXPERIMENTAL VERIFICATION OF MODELING RESULTS

A. PROTOTYPE OF A PORTABLE MACHINE TOOL

In the next stage, a prototype of a machine tool was built (Fig. 7), for which experimental tests were conducted. The experimental tests were divided into two stages: (i) the impulse test to confirm that the dynamic properties of the machine tool remain constant with respect to the C axis and to verify the accuracy of the finite element model; (ii) cutting tests to confirm the projected stability of machining.

B. EXPERIMENTAL TESTING OF THE PROTOTYPE IN THE FORM OF AN IMPULSE TEST

The first stage of experimental research was carried out in the form of an impulse test. The excitation was performed using a PCB 086D20 modal hammer, exciting the machine tool at the end of the tool successively in three mutually perpendicular directions (corresponding to the coordinate system adopted at the modeling stage). The response to the imposed

FIGURE 7. **Constructed prototype of portable machine tool.**

TABLE 1. **Parameters of signal acquisition.**

Parameter	Value
Sampling rate	4,096 Hz
Frequency resolution	0.5 Hz
Signal acquisition time	2 s
Number of averages	10
Scaling of the frequency response function	global

excitation was measured on the tool and workpiece using PCB 356A01 accelerometers. Data acquisition was conducted using Scadas Mobile Vibco and Testlab 2019.1 software. The estimation of frequency response function was performed with the use of H1 estimator. Detailed information on the parameters of signal acquisition is provided in Table 1.

The results of the experimental tests in the form of relative receptance function for three previously selected configurations are shown in Figure 8.

Analyzing the receptance functions in Figure 8, one can see their very high similarity. This makes it possible to state that the dynamic properties of the designed machine tool do not change significantly along with the change of its position in relation to the C axis. This confirms the thesis formulated in the introduction.

As results from the above, the verification of the developed finite element model was carried out exclusively for the C2 configuration. Figure 9 presents a comparison of the

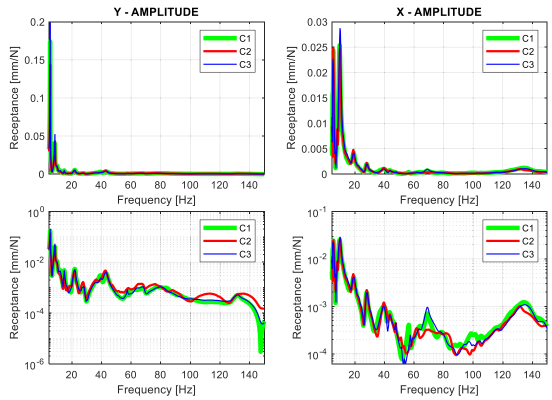

FIGURE 8. Experimentally determined receptance functions for three selected configurations.

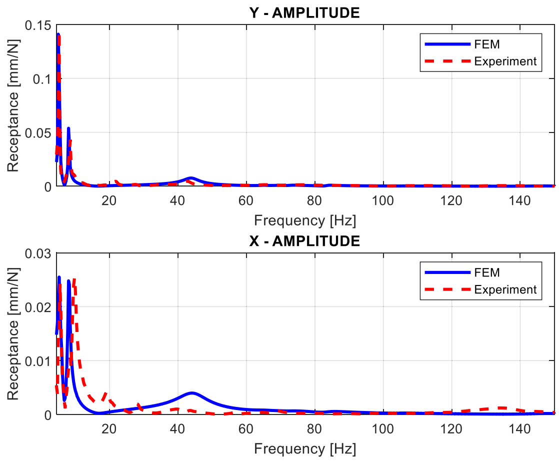

FIGURE 9. Comparison of the receptance functions determined experimentally and by computing.

receptance functions determined experimentally and based on the model.

The comparison of the receptance functions determined experimentally and by computation, as shown in Figure 9, indicates the correctness of the adopted modeling method. The accuracy of the developed model is high, both in terms of mapping the resonance frequencies and amplitudes of the receptance functions. The X-direction receptance function in the frequency range corresponding to the mode shape at 43.9 Hz is characterized by poor agreement. It should be noted, however, that the significance of amplitudes in the X-direction in terms of influence on the stability of machining is much lower than in the Y-direction (due to lower amplitudes). Therefore, the model can be considered to represent its actual counterpart in a satisfactory manner.

C. CUTTING TESTS – VERIFICATION OF MACHINING STABILITY

In the second stage of the experimental studies, work tests were carried out to verify the predicted areas of stable and unstable machining. Based on the analysis of the experimentally obtained stability lobe diagrams, a test plan was developed, indicating six machining cases with different

TABLE 2. Experimental plan – cutting tests.

Test No.	Depth of cut a_p [mm]	Feed rate V_f [mm/min]	Spindle speed n [rpm]
1	0.5	896	5,600
2	2.0	896	5,600
3	0.5	896	11,200
4	0.5	896	15,000
5	1.0	896	15,000
6	2.0	896	15,000

parameters. The location of these cases in the lobe diagrams is shown in Figure 10, while Table 2 shows the exact values of the machining parameters used in the tests.

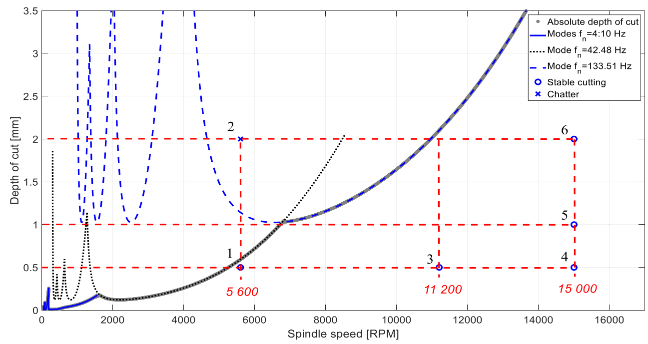

FIGURE 10. Measurement points in relation to the lobe diagrams based on the model.

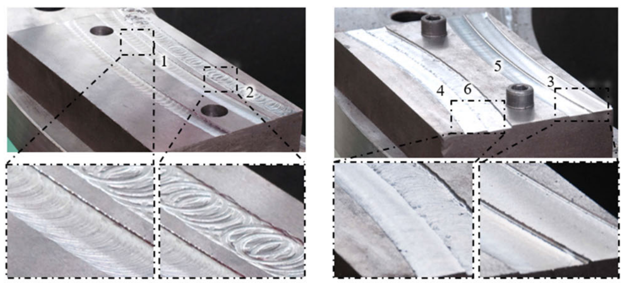

FIGURE 11. Samples after machining (with cutting tests marked 1-6).

The cutting tests were carried out on a specially prepared sample made of general-purpose structural steel (1.0037 acc. to EN 10025:1993), which was fixed to the front face of the flange (Fig. 11). This allowed to eliminate interference of the machining process resulting from, among others, the quality of the milled surface prepared for testing (sample surfaces were ground). The criterion for assessing stability is the roughness parameters of the machined surface obtained during the milling process. The roughness was measured using the Hommel Tester T1000 profilographometer. As a result of the cutting tests carried out, machined surfaces were obtained, which are shown in Figure 11. The results of surface quality measurements are shown in Table 3.

Analyzing the results obtained from the machining trials, they reflect the predicted areas of stable and unstable machining. More precisely, for the 2nd measurement point

TABLE 3. Cutting tests results - roughness.

Test No.	R_t [µm]	R_z [µm]	R_a [µm]	R_p [µm]
1	7.14	5.11	0.76	4.05
2	31.17	15.40	2.39	17.77
3	7.34	4.02	0.55	1.72
4	7.28	3.95	0.49	1.69
5	9.59	5.14	0.56	2.80
6	13.41	6.56	0.74	3.37

($a_p = 2\ mm$, $V_f = 896\ mm/min$, $n = 5,600\ rpm$) unstable machining was observed, which was reflected in both the chatter marks (Fig. 11.) and the surface roughness coefficient $R_a = 2.39\ \mu m$ (Tab. 3). For the remaining measuring points, stable machining was achieved as predicted. Stable machining was achieved at measurement points 1 and 6, which were relatively close to the limit of loss of stability, although the roughness values are relatively high, 0.76 μm and 0.74 μm respectively, which is reflected in the machined surface. The lowest indicator R_a was obtained for test No. 4 ($a_p = 0.5\ mm$, $V_f = 896\ mm/min$, $n = 15,000\ rpm$).

V. CONCLUSION

The paper presents the design of an ultra-light, axisymmetric, portable machine tool for in situ flange face milling.

The main design criterion was machining stability. This criterion was adopted because of the high susceptibility of the machine tool, which, as was proven by model and experimental tests, made machining virtually impossible. Based on the analysis of stability lobe diagrams, a high-speed spindle was selected, which made it possible to achieve machining parameters enabling stable machining. The spindle speeds must be set above $N = 0$ lobes of the dominant modes to provide stable cutting conditions.

The characteristic feature of the presented portable machine tool is its axisymmetric structure which guarantees the unchangeability of the dynamic properties of the machine together with its rotation in relation to the main axis of the machine tool (C). This translates into the possibility of using technological parameters that ensure stable machining, regardless of the angle position of the machine tool.

The study also shows that it is possible to build a low-dimensional, reliable finite element model without using substructuring or reduction methods. Instead, a well-thought-out discretization and substitute models of complex load bearing systems were used. Such a model ensures that the dynamic properties of the machine tool can be quickly determined for different position configurations of its components.

The reliability of the developed finite element model of the portable machine tool was manifested through a high agreement with its real counterpart. To be precise, the model accurately predicted the receptance functions in terms of the resonant frequencies and amplitude values. As a result, the correct prediction of stable and unstable machining parameters was obtained. This was confirmed by a series of cutting tests. The conducted cutting tests accurately reflected the deterioration of roughness parameters of the machined surface as cutting parameters approach the stability limit.

To summarize, a general conclusion can be drawn: in the case of machine tools characterized by high susceptibility and limited possibility of introducing structural modifications to the design, a good choice may be the selection of a spindle that allows to obtain parameters that ensure stable machining.

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PAWEŁ DUNAJ was born in Poland, in 1991. He received the master's degree in mechanical engineering from the West Pomeranian University of Technology, in 2015, and the Ph.D. degree in 2019.

He published 23 articles, since 2017. In last five years, he participated or managed nine research projects carried out in cooperation with industrial partners. His scientific interests focus on issues related to the dynamics of machine tools, with the use of the finite element method for its evaluation.

MICHAŁ DOLATA was born in Szczecin, Poland, in 1983. He graduated from the Faculty of Mechanical Engineering, Szczecin University of Technology, in 2007. He received the Ph.D. degree from the Department of Mechanical Engineering and Mechatronics, West Pomeranian University of Technology, in 2015. Since 2008, he has been employed as an Assistant Professor and a Lecturer. At the same time, he was involved in many scientific researches in cooperation with local industry. He is currently the coauthor of more than 15 articles and conference speech. He holds two patents. His interests included FEM methods and machine design processes. His current research interest includes lightweight mobile CNC machines.

BARTOSZ POWAŁKA graduated from the Faculty of Mechanical Engineering, Szczecin University of Technology, in 1994. He received the Ph.D. degree in mechanical engineering in 1998, and the Habilitation degree in 2007.

He received the Columbus Scholarship from the Foundation for Polish Science, which gave him opportunity to work with the University of Michigan, from 2004 to 2006. He promoted four doctors. He is currently supervising three Ph.D. students. His research interests include machine dynamics and modeling of manufacturing processes. He received the DAAD Scholarship from Leibniz University Hannover in 1999 and the Polish State Professorship Title in 2019.

PIOTR PAWEŁKO received the master's degree in mechanical engineering from the Szczecin University of Technology, in 2001, the Ph.D. degree in 2007, and the Habilitation degree in 2014.

In last five years, he managed or participated in seven research projects carried out in cooperation with industrial partners. He is currently supervising two Ph.D. students. His scientific interests focus on issues related to the designing of machine tools, with the use of the finite element method.

STEFAN BERCZYŃSKI was born in Szczecin, Poland, in 1950. He graduated from the Faculty of Mechanical Engineering, Szczecin University of Technology, in 1974. He received the Ph.D. degree from the Faculty of Mechanical Engineering, Szczecin University of Technology, in 1978, and the Habilitation degree from the Faculty of Mechanical Engineering, Warsaw University of Technology, in 1982.

He was the Dean of the Faculty of Mechanical Engineering and the Faculty of Mechanical Engineering and Mechatronics, Szczecin University of Technology and the newly established West Pomeranian University of Technology, Szczecin. He is currently supervising two Ph.D. students. In 1990, he received the Academic Title of Professor of Technical Sciences.