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Intelligent fixtures for active chatter control in milling

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Abstract

The mitigation of chatter vibrations in milling has collected the interest of several researches in the last decades. One of the most industrially oriented alternatives is represented by active fixtures, complex mechatronic devices capable of actuating the workpiece during machining operations, with the purpose of stabilizing the process by generating counteracting vibrations. Most of the previous works show different fixture architecture and model based control techniques. This paper deals with the development and testing of such an active fixture, presenting the main design aspects and the features of the black-box control-logic used. Experimental tests are presented to show the achievable chatter mitigation.

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Keywords: Milling; Chatter; Control; Active Workpiece Holder (AWH)

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1. Introduction

Chatter vibration represents one of the most limiting factors in assessing the achievable performance, in terms of productivity, of modern machining operations. Indeed, these detrimental vibrations could drastically decrease the achievable material removal rate and sensibly increase tool wear [1]. Literature reports several different approaches aimed at predicting or preventing such unstable vibrations by selecting optimal machining parameters, through numerical models [2] or dedicated experimental procedures [3]. These approaches are limited by the required expertise and by the fact that the predicted optimal condition is representative only of a specific tool-material combination. On the contrary, active approaches capable of monitoring the process and exerting adequate counteracting actions, are more easily adaptable to general applications. Spindle-speed-variation technique is a clear exemplification of these approaches [1]. Recently, the use of active materials has strongly contributed to the development of alternative active approaches based on the integration of mechatronic systems in the machine-tool structure. Among these, active fixtures, often referred to as Active Workpiece Holders (AWHs), seem to be more industrially appealing solutions. Their direct retrofittability and adaptive control logics would indeed demand for lower expertise and reduced set-up time [4].

This paper deals with the development of such a mechatronic device aimed at mitigating chatter vibrations by monitoring the milling process in real-time, through integrated sensors (i.e., accelerometers), and by generating adequate counteracting vibrations, by means of integrated piezo-actuators. The black-box control logic used in the present work allow avoiding preliminary system identification and modeling as generally required for the control development and implementation in this kind of devices [4,5].

The paper covers the main features of the mechanical design of the two degrees of freedom active fixture, along with the crucial aspects of the sensors/actuators selection and integration. Subsequently, the main aspects of the development and implementation of the novel control algorithm are presented to show its peculiar features. Finally, the developed active fixture prototype is experimentally tested in 2.5 axis milling operations in order to show the achievable improvements in terms of chatter mitigation.

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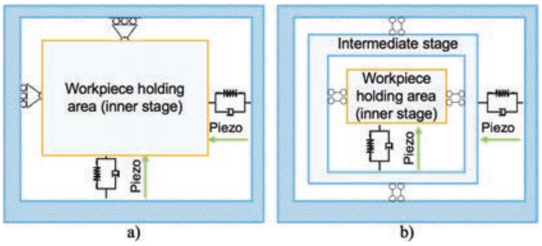

Fig. 1 Schematization of a) parallel-kinematics architecture, b) serial-kinematics architecture.

2. Active Fixture development

The development of an active fixture goes through the integration of different subsystems with peculiar issues to be tackled, as for most mechatronic devices. The next sections cover the most relevant aspects of the prototype development, discussing the main issues that needed to be addressed in order to achieve the desired global behavior.

2.1. Fixture architecture and mechanical design

Referring to literature, two main fixture architecture can be chosen to develop a two degrees of freedom (DOFs) device capable of counteracting tool vibrations in the plane normal to tool axis. The simplest solution could be represented by a compliant mechanism with parallel kinematics, schematized in Fig. 1a, which would allow for symmetric fixture response and to reduce the overall masses and dimensions [4].

The main drawback of this kind of architecture is represented by the relevant cross-talk effect that could arise between the actuated axes and that could be overcome only by employing additional sensors and control strategies. For this reason, a different architecture was selected for the active fixture prototype, opting for a structure based on serial kinematics, schematized in Fig. 1b, that would allow for theoretically decoupled motion of the stages [5]. Nevertheless, the nested structure would imply different geometries and masses for the inner and outer stages, that would hence show different dynamics and should be carefully regarded during the fixture design.

Regardless the fixture architecture selected, the design of the fixture frame plays a crucial role in assessing the achievable active fixture performance. It should be capable of simultaneously ensuring the needed stiffness, imposed by precision requirements, while allowing enough flexibility for the actuators to produce adequate workpiece displacements [6]. These conflicting requirements are usually met by integrating monolithic flexure hinges in the frame in order to adequately decouple axes motion and achieve the desired compliance level. A first dimensioning of these mechanical components can be carried out according to some of the empirical equations presented in literature [7]. Numerical models could then be useful to optimize the positioning of the flexure hinges within the fixture frame, with the purpose of achieving the desired dynamic response that would contribute to the maximization of the device bandwidth, as exemplified in Fig. 2.

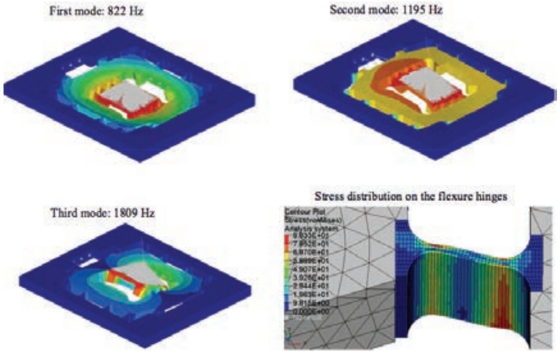

Fig. 2 FEM analysis of the developed prototype fixture (workpiece mass not included) and stress distribution on the flexure hinges.

Moreover, detailed numerical model would allow estimating the stress distributions within the fixture, needed to investigate the fatigue limits in highly dynamic application, such as the proposed one.

The development of the AWH was carried out according to these considerations and following the general guidelines provided by the work of Haase [8]. The developed prototype is shown in Fig. 3.

2.2. Actuators selection and implementation

As shown in Fig. 3, the developed prototype integrates four piezoelectric stacks actuator for each dynamic axis. The selection of suitable actuation devices represented the main issue to be tackled for the development of a device with adequate bandwidth and reliability.

In recent years, several alternative actuation techniques became available and their level of performance is continuously growing. Nevertheless, piezoelectric technology still represents the most appealing alternative, mainly due to the achievable power density and bandwidth. The integration of this kind of actuation devices in the machine tool components with the purpose of interacting with the machining process, is a renowned topic, but the features of the piezo-ceramic used could drastically reflect on the achievable bandwidth and performance.

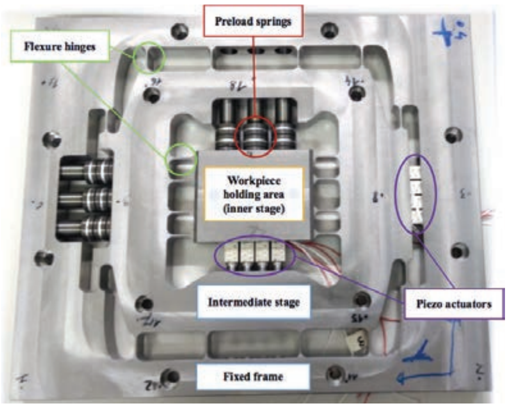

Fig. 3 Active fixture prototype (external dimensions 343x289x20mm).

Self-heating issues represent one of the main limiting factors in long-term dynamic operation of piezoelectric actuators. In that sense hard-doped piezo ceramic represents the best trade-off between the achievable level of force/displacement generation and the influence of self-heating [9].

All these considerations where taken into account in selecting the suitable actuators to be implemented in the AWH prototype. Should indeed be considered that this device should be capable of generating counteracting vibrations at the chatter frequency, that can easily exceed the kHz range, and possibly for a long operative time. Considering the commercially available alternative, Noliac actuators based on NCE46 hard-doped ceramics have been selected. The main features of the selected actuators are reported in Table 1.

Table 1 Features of the selected SCMAP-NCE46-10-10-2-200-H36-C01 Noliac actuators (referred to one single actuator).

Dimensions WxHxL (tips excluded)	10x10x36 mm
Estimated blocking force	3200 N
Capacitance value (typical)	5300 nF
Maximum displacement (typical)	32.3 µm
Maximum operative voltage	200 V

Particular attention has been put in adequately housing the piezo actuators within the AWH structure. These components are capable of withstanding mostly compressive stresses, hence adequate counter measurements should be taken to prevent them against bending and tensile stresses.

In order to accomplish an adequate housing of the actuators, hemi-spherical end-tips have been glued to the actuators extremities and dedicated engraved hemi-spherical supports have been machined in the AWH frame and in the actuators set screws, as shown in Fig. 4.

Even though such a unilateral constraint would intrinsically prevent any tensile stress on the actuators, adequate preload has been applied to prevent the arising of play within the joints, potentially leading to shocks on the actuators tips in demanding dynamics applications. The preload is applied by means of three stacks of Belleville springs acting against the stages, oppositely to the actuators, as shown in Fig. 3. The required preload force has been estimated thanks to numerical simulations of the FEM model of the fixture under operational conditions.

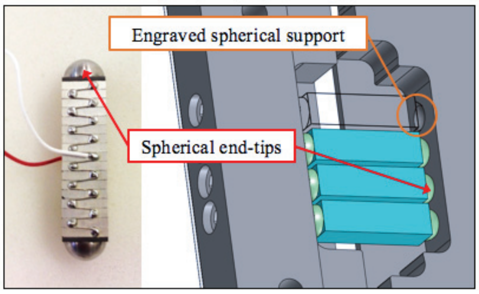

Fig. 4 Detail of the piezo-actuators tips and housing within the AWH frame.

3. Control Logic development/implementation

As an alternative to renowned control logic techniques used for active damping in milling applications [10], the proposed approach doesn't require a prior investigation and modelling of the system to be implemented. In a previous research [11], authors investigated the feasibility of using low-frequency actuation to disrupt the regenerative effect that self-maintain the chatter vibrations, without requiring dedicated models of the systems. In this application, instead, an intelligent control based on a set of integrated modules is used to identify the chatter frequency to be mitigated and the optimal control signals needed for the purpose (i.e., the actuation parameters that would allow a condition of disruptive interference of the chatter vibrations). In the frame of intelligent control techniques, Artificial Neural Networks (i.e., ANNs) are employed as function approximators. The methodology here proposed, developed in prior activities by Paragon S.A., is a combination of ANNs and Genetic Algorithms (GAs), and falls into the category of indirect design approaches, more specifically under predictive control schemes. The application does not yet have a product name, any titles or descriptive names provided here are provisional.

The software application developed exists in several forms, the latest being a version developed using MATLAB®. This was done for several reasons, such as the wide-spread use of this particular development environment, simplification of the software application in so far as utilizing functions already available for DAQ and DSP purposes, and to allow for greater application flexibility or extension via use with Simulink®. A diagram of the software modules–sequences is provided in Fig. 5.

3.1. Frequency analysis and excitation

An external disturbance is generated; in this application the machining process. The vibration of the structure (i.e., primary field) is measured at each sensor position (i.e., accelerometers in this application).

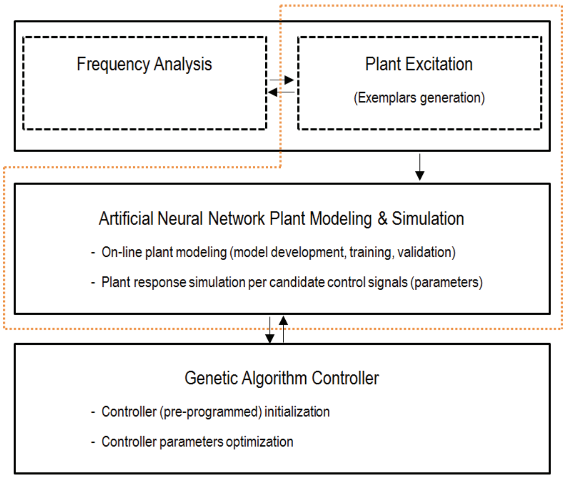

Fig. 5 AVC software modules and functions.

The data acquired from each sensor is analyzed using Fast Fourier Transform (FFT), and the dominant frequency (i.e., chatter frequency and possibly its sidebands) and its corresponding characteristics per sensor/channel are identified. The sequence ends with the selection of the highest dominant frequencies identified. These are the vibration frequencies to be controlled/reduced.

As the goal is to create a system that reduces vibration through active means, in order to build and validate a representative model it is required to include data on how it reacts to such control actions (i.e., the residual field).

Through specific procedure, with the primary vibration field present, each actuator is excited with control signals (i.e., secondary field) in a randomly defined sequence, not simultaneously; these correspond to all the identified dominant frequencies, but the signals are of random amplitude and phase. The accelerometers in each per actuator excitation sequence measure the resulting residual field at their location.

Specific data sets (i.e., exemplars) are created from the data generated from this sequence, which are in turn utilized for training and validation purposes during the development of the ANN-based behavioral model in the next step. This process runs until the necessary number of exemplars is created.

3.2. ANN model and simulation

In this sequence, the behavioral model is created, trained and validated (on-line or off-line). As it is proprietary information of Paragon S.A., it won’t be described in detail due to confidentiality.

During the development of the prototype, one type of ANN developed was a Feedforward Multilayer Perceptron (MLP) Neural Network, trained with a Back Propagation (BP) approach with the following parameters and steps:

- Input layer parameters: Control signals amplitude and phase (in relation to the dominant frequencies and actuators).
- Output layer parameters: Residual field amplitude measured at each sensor location. The number of nodes in the output layer of the ANN is equal to the number of sensors used.
- The ANN generated is trained (update of network architecture and weights) and validated using the exemplar patterns that were generated in the previous sequence, as shown in Fig. 6.

In the end, the completed ANN model is able, given the control signal inputs to each actuator, to simulate the resulting residual field.

3.3. GA controller

In this final sequence, the control algorithm is initiated. As a global optimization technique, GA is utilized to determine the optimal control signals to the actuators, i.e., in terms of amplitude and phase.

The optimization loop aims to determine the optimal parameters to achieve best possible reduction of amplitudes across the dominant frequencies, and candidate control parameters solution are tested using the ANN model.

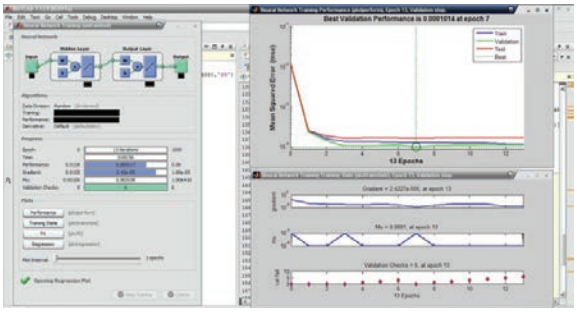

Fig. 6 ANN model training phase.

The optimization loop will execute until the GA converges, as exemplified in Fig. 7, after which the software application ends with the operation of the controller that proceeds to generate the optimized control signals to drive the actuators and the FAS monitors the AVC performance.

Unfortunately, no further details can be published at this time about details such as the gene coding or the fitness function, as they are proprietary information of the developer that was brought as technical background to this research activity.

4. Experimental tests

In order to investigate the performance of the developed AWH in mitigating chatter vibrations, dedicated experimental tests were carried out on a DMG DMC 635V eco 3-axis milling machine, in slotting operations with different toolpaths in order to show also the potentialities of the implemented control logic.

4.1. Experimental setup and tests description

A 16mm indexable mill with two cutting flutes (i.e., Widia VSM11D016Z02A16XD11L170) was used for the tests. The workpiece was a 60x132x25mm AISI P20s steel secured to the AWH by means of two M8 screws. An acrylic cover was placed below the workpiece to protect the electronics within the AWH against hot chips and cutting fluids.

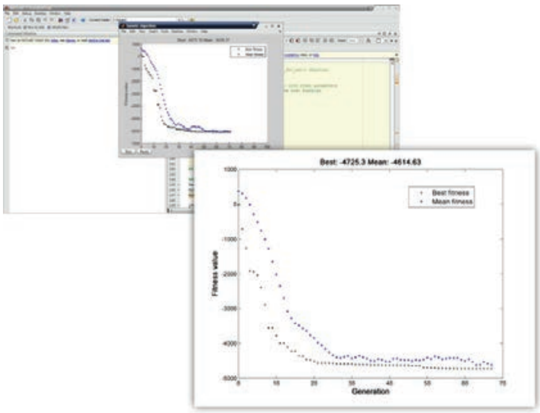

Fig. 7 Controller generation execution and GA (optimization) convergence.

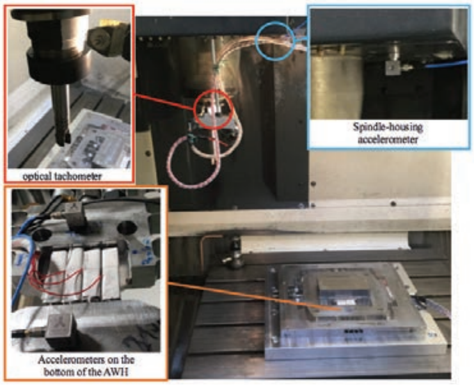

Fig. 8 Experimental tests setup.

Three PCB356A32 triaxial accelerometers have been used as sensors for the derivation of the optimal control signals. Two of these were secured to the bottom of the fixture in order to measure the vibrations of each dynamic stage, while the third one was placed at the base of the spindle housing in order to get a better estimation of tool-tip vibrations. An optical tachometer was used to provide constant reference of a specific point of the tool during rotation, which in turn was used for calculating the phase difference for the control signal, based on the principle of two identical waves with 180^o phase difference canceling each other out (i.e., disruptive interference). The optical tachometer provided a stable point from which to make phase calculations based on the tool’s geometry (number of teeth). The test setup is shown in Fig. 8.

Slotting tests were performed on the specimens. First, some linear cuts, along both X and Y directions, were investigated in order to get a preliminary assessment of the achievable results in simple operations. Subsequently a toolpath composed of linear, diagonal and circular toolpaths was tested with the purpose of investigating the effects of the synthesis controller in adapting the optimal control signals, computed in linear cutting tests, to a general toolpath. The specimen machined with this archetype toolpath is shown in Fig. 9.

Some preliminary tests were conducted to identify both workpiece/fixture and tooltip Frequency Response Functions (FRFs), reported in Fig. 10, and stability limits for the test setup. For sake of brevity the results are here only summarized. The preliminary experimental tests highlighted a natural frequency of the tool equal to 1151 Hz and an axial depth of cut limit, in slotting operation equal to 0.4 mm, for the rotational speed used in the tests (i.e., 3600 rpm).

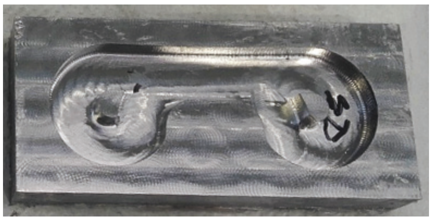

Fig. 9 Specimen machined with the archetype toolpath.

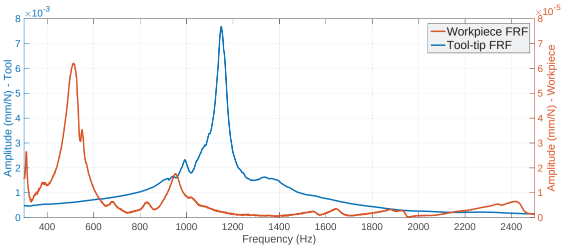

Fig. 10 Measured FRFs of tool (blue line) and workpiece (red line).

As shown in Fig. 10, the effect of workpiece mass pushes the dominant mode of the workpiece/fixture down at approximately 530Hz. The amplitude at the dominant mode of the tool FRF is two order of magnitude higher respect to the workpiece/fixture FRF, suggesting that the chatter vibrations would be originated by the tool flexibility, while the fixture dynamics would play a negligible effect on the stability limits.

In order to ensure that the tests are conducted in chatter conditions, axial depth of cuts of 0.5mm and 0.6mm were used for the experimental tests.

4.2. Results

To show the actual performance in reducing chatter vibrations, the results are discussed by comparing the accelerations level achieved in the tests with 0.6mm axial depth of cut and AWH control turned on, with the vibration levels achieved in the reference test (i.e., 0.5mm axial depth of cut and no control applied). The results are based on the acceleration signal measured at the base of the spindle housing, given that this sensor provides the better estimation of tool-tip vibrations. The triaxial accelerometers placed below the inner and outer stages of the active fixture are, on the other hand, an important part of the controller, as they are necessary for the ANN model generation process, particularly for measuring the response from the random signals generation.

4.2.1. Linear slotting tests

The linear slotting cuts were simpler to control, faster to assess and required less material per cut compared to diagonal or circular motions. For the linear X cut at Ap=0.6 mm depth, reductions of 83% (X axis) and 75% (Y axis) were achieved compared to the chatter level at Ap=0.5mm. Similarly, for the linear Y cut, reductions of 60% (X axis) and 37% (Y axis) were achieved. In both cases, the controller and the AWH is effective in enabling the machining at Ap=0.6 mm with at least 37% less chatter vibration compared to Ap=0.5 mm, allowing for higher material removal rates and presumably less tool wear. These discussed results are shown in Fig. 11 and Fig. 12.

4.2.2. General toolpath tests

As anticipated, a toolpath composed by linear and circular motions, was used for testing the strategy applied to an archetype toolpath, i.e. the synthesis of the best counter-vibration signals identified from the early linear X and Y cutting tests.

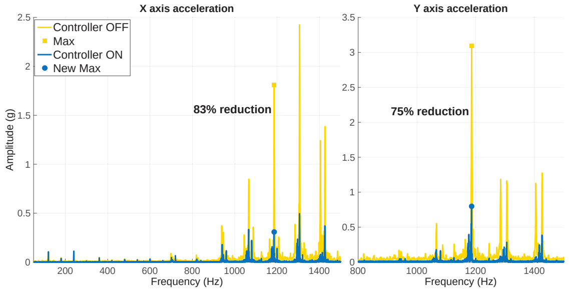

Fig. 11 Example of chatter (X-Y axes) during linear X cut at Ap=0.5 mm (yellow) and the resulting reduction at depth of Ap=0.6 mm (blue).

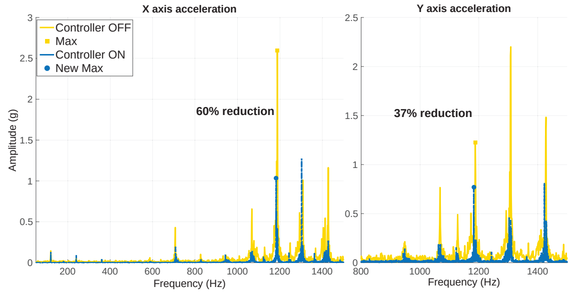

Fig. 12 Example of chatter (X-Y axes) during linear Y cut at Ap=0.5 mm (yellow) and the resulting reduction at depth of Ap=0.6 mm (blue).

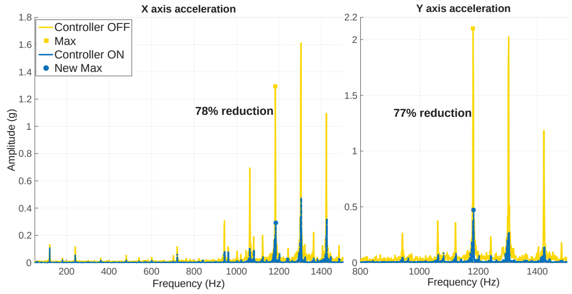

Fig. 13 Example of chatter (X-Y axes) during test-case cut at Ap=0.5 mm (yellow) and the resulting reduction from XY-synthesis controller at depth of Ap=0.6 mm (blue).

The concept was to track the motion of the cutting tool and synthesize the suitable counter-vibration signal from percentages of its X and Y components, using a basic PID control scheme and trigonometric summation of the axes vectors. The controller demonstrated to be effective in tracking the motion and synthesizing the corresponding output. The results for Ap=0.6mm were approximately 77% reduction for both axes compared to the chatter level of cutting at Ap=0.5mm, as shown in Fig. 13.

5. Conclusions

This paper deals with the development of an active workpiece holder aimed at mitigating chatter vibrations in milling. The mechanical design of the fixture was carried out in accordance to previous literature works, but particular focus was put in improving the achievable performance by a careful mechanical design and improved actuation devices. The use of hard-doped piezoelectric actuators allowed indeed to sufficiently extend the bandwidth of the developed device, with respect to the previous works presented in literature.

The novel controller strategy implemented in the AWH demonstrated to be suitable for chatter reduction in the validation tests. The achieved reduction allows to use higher depths of cut with lower vibration level, potentially increasing productivity. Valuable practical information has been gathered through this first practical implementation. The control algorithm performed as expected on the selected hardware platform. The actual operation was simple and straightforward, allowing for a quick process of initial controller tuning using a small part of a single workpiece and from then on no other user intervention is required. The controller can be further improved by testing on larger test cases and integrating into permanent machining installation.

Acknowledgements

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