




Article

A Dedicated Design Strategy for Active Boring Bar

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Featured Application: The work aims at supporting the design of active boring bar to effectively damp process vibrations. Machining industries, especially the ones performing deep internal turning operations, could benefit from the potential application of such technology.

Abstract: Unstable vibrations (i.e., chatter) onset is one of the main limits to productivity in deep boring bar processes. Active damping systems allow to increase machining stability in different configurations (i.e., tool setup), without requiring cutting system dynamic characterization. Design of an active boring bar involves the development of monitoring system (sensors), actuation system and control logic. While several control logics were evaluated and discussed, few design solutions were presented in the literature, focusing only on building prototypes to demonstrate control logic effectiveness. In the presented work, a deep analysis of the main issues and requirements related to active boring design was carried out and a systematic approach to tackle all the critical aspects was developed. The results of the proposed method are: (i) optimal actuators positioning able to damp vibration along two directions; (ii) preload system design guaranteeing the correct actuator preloading for the operating conditions; (iii) covers design to protect actuators and ensure the dynamic and static equivalence between active and standard boring bar. Following this approach, an active boring bar was designed, realized and tested. The results prove the required equivalence between active and original boring bar and assess the damping effect.

Keywords: active damping; boring; machining; dynamics

1. Introduction

In the case of deep boring processes, the use of high aspect ratio tools may be unavoidable. As a result, due to the tool low stiffness, unstable vibrations (i.e., chatter) [1] may occur compromising surface quality and tool life. Chatter onset depends on (i) system dynamics (i.e., tool Frequency Response Function); (ii) cutting parameters (i.e., radial depth of cut, spindle speed and feed per revolution); (iii) tool insert geometry; (iv) tool and workpiece material.

As presented in [2], two different strategies may be used to ensure a stable machining: exploiting the lobbing effects, limiting stable and unstable cutting conditions (by means of out-of-process or in-process strategy) or changing the process/system dynamic behavior. In the out-of-process methods, chatter prediction models [3] are used to detect cutting parameters that gives a stable machining. However, these approaches require several inputs, such as the dynamic behavior of the tooling system and accurate cutting force model. On the contrary, in the in-of-process methods, cutting parameters (usually spindle speed [4]) are corrected during machining, as soon as chatter is detected. However, exploiting the lobbing effect is not effective in cases of high compliance tools, where stability field is limited and stable cutting parameters are associated to low productivity levels. Hence, increasing the limits by changing the system or process behavior is preferred. The process stability can be modified by dedicated cooling strategy (e.g., cryogenic [5]), which affects the mechanism of chip formation,

and hence, could stop or release vibrations. However, these techniques depend on the material to be machined. Therefore, it is generally preferable to improve the tooling system dynamics by means of passive or active techniques. Passive techniques employ devices that, once mounted on a structure, modify its dynamic response. In particular, if correctly designed (i.e., tuned), they can reduce structure peak amplitude and, in case of machining process, increase chatter stability limits [6]. Even if these devices, compared to active ones, are cheaper and do not require external power, their effect on the system response is limited to a short frequency range. This implies that dynamic characterization of the component, on which passive devices will be installed, is required to properly design the device. Moreover, the same device is not effective for different configurations [7].

Active approaches require a more complex architecture able to monitor, diagnose and act on the process in order to suppress chatter vibration. Amplitude vibration is continuously acquired by sensors and the signal is processed in real time to detect chatter onset. Control logic, properly designed and implemented, receives the signal and drives the actuators in order to contrast chatter. Despite the complexity, these approaches are more suitable for industrial application, since they are able to mitigate chatter in different configurations (e.g., different tool set-up, different cutting parameters). The design of an active boring bar involves the design of monitoring system (sensors), actuation system and control logic.

Up to now, research works has mainly focused on the development and implementation of different control logics, such as optimal control [8], direct position feedback [9] and direct velocity feedback (DVF) [10,11]. Among all, DVF is the most common and mature control logic, since it was demonstrated to be a robust approach [12] and its effectiveness was proved in several applications for chatter mitigation [13,14]. Beside control logic, active strategies effectiveness is strongly related to the inertia of the applied sensors and actuators. However, no work was dedicated to the design of the structure of such devices. Indeed, few design solutions were presented in literature focusing only on building prototypes to demonstrate control logic effectiveness. As a result, some critical aspects in actuation system design have not yet been addressed and a systematic approach for active boring bar design has not been presented.

The aim of the presented work is to develop a structured approach for active boring bar design based on DVF control system, focused on actuation system and its integration on the cutting tool. Starting from a commercial boring bar, the method supports actuators selection, positioning and housing in order to increase the damping effectiveness for chatter suppression, while respecting technological limits. The proposed approach covers aspects that have not yet been addressed in the literature, which are crucial for a correct active system functioning. In particular, this approach includes: (i) an effective model for preload estimation in order to protect actuators from traction stress even during machining, (ii) a design strategy for preload system and (iii) a design strategy for protective covers (necessary to protect actuators) that guarantees static and dynamic equivalence between original boring bar and active boring bar. Following the proposed method, an active boring bar was designed and manufactured. Using the developed prototype, experimental tests were carried out in order to validate the method and assess the prototype performances.

2. Proposed Method

An active boring bar is composed of at least a sensor, capable of detecting the onset of the vibration to be mitigated, an actuator that generates counteracting forces based on a tailored control logic embedded in a real-time controller, as schematized in Figure 1.

In particular, in case of DVF control logic, the controller receives, as input, vibration velocity and determines a counteracting force proportional to the velocity. As result, DVF acts as a pure damping strategy, reducing vibration amplitude around natural frequencies, which are the most critical for chatter onset.

However, damping effectiveness depends not only on the control logic, but on the entire actuation system that, therefore, needs to be carefully designed. In particular actuators choice and their integration on the system are found to be crucial aspects, due to the necessity of satisfying several requirements.

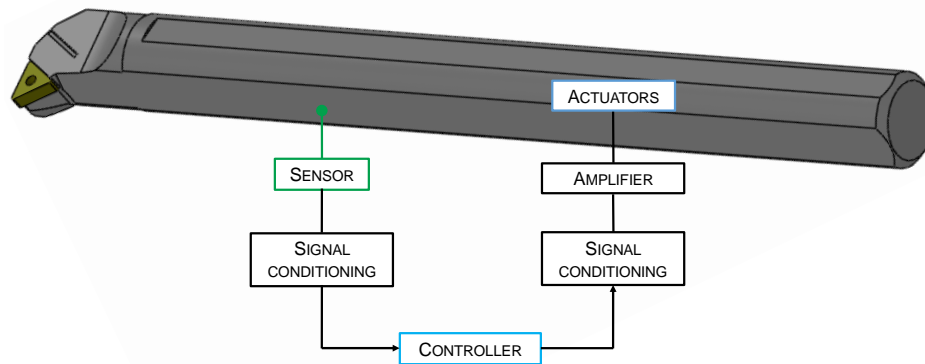


Figure 1. Active boring bar scheme.

2.1. Requirements for Active Damping System

In order to effectively contrast chatter, both frequency and direction of the unstable vibration must be considered. Indeed, the first aspect defines the frequency range of vibrations that must be damped (crucial for sensors and actuators choice), while the second one is related to actuation directions (crucial for actuators optimal placing). Chatter frequency can be estimated by means of chatter prediction models or experimentally measured and it is related to system dynamics. Actuators and sensors resonance frequency should be at least five times greater than chatter frequency in order to guarantee sufficiently low inertia and an effective response time of the active system. The direction along which the chatter arises depends both on chatter type and on boring bar dynamic behavior. In particular, in case of primary chatter, unstable vibrations occur in the cutting velocity direction (Figure 2a), while in case of regenerative chatter, instability is related to vibrations along radial depth of the cut direction (Figure 2b).

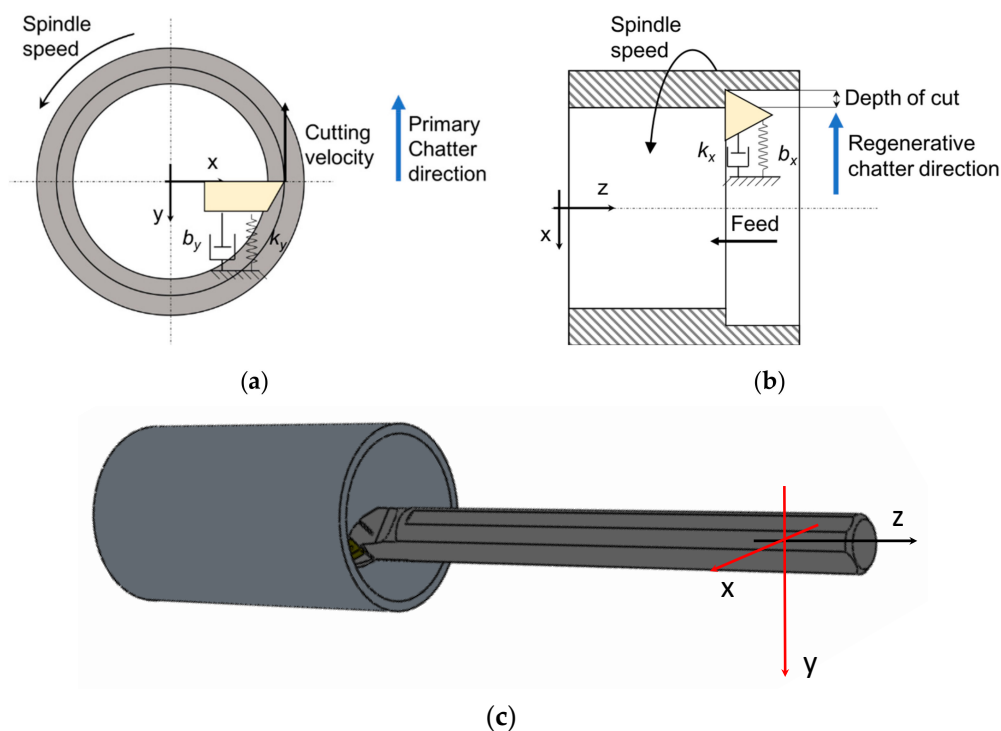


Figure 2. (a) Primary chatter; (b) regenerative chatter; (c) boring bar most flexible directions (red lines).

Therefore, in the first case, optimal actuation direction is along cutting velocity, while in the second case, it is necessary to actuate in the radial depth of cut direction. As observed by Pratt and Nayfeh [15], chatter passes from primary to regenerative varying radial depth of cut, due to the presence of tool nose, which determines cutting force direction. In deep boring process, tools, due to the high aspect ratio, are highly flexible in both these direction (Figure 2c). Then, in order to ensure a damping effect independently of depth of cut, active boring bar must be able to reduce vibrations both along cutting velocity and feed direction.

In addition to these requirements, since DVF control logic acts only on frequency range close to the system natural frequency, the active boring bar design should consider static behavior as a constraint: standard and active boring should share the same static behavior and adding the actuation system should not worsen the static and dynamic response of the structure with the control off.

Lastly, technological limits must be considered. In particular, since tool overhang is selected based on the maximum hole depth that can be manufactured, while radial size is constrained by the minimum hole radius, the final solution cannot imply a significant overhang reduction or radial size increase.

Summarizing, five requirements were identified:

1. The damping system should be capable of generating counteracting forces in the frequency range of the unwanted vibrations to be mitigated.
2. The active boring bar should be capable of damping vibrations along two directions.
3. The dynamic and static behavior of the active boring bar, with control off, should be as close as possible to the ones of the standard system.
4. The housing of the actuators should not reduce the actual exploiting overhang of the standard boring bar.
5. The housing of the actuators should not exceed a given radial dimension.

2.2. Most Effective Actuation Strategy Selection

In this section, the chosen actuation strategy is presented and motivated. Different actuation solutions were presented in literature for active boring bars. Chen et al. [16] proposed a solution using magnetic actuators, mounted on a tailored system. Even if this kind of actuators are found to be effective, due to their capability of giving high loads, in the proposed solution the presence of the actuator, mounted on the external surface of the tool, reduces its overhang. In [17], a tool adaptor with an integrated active system is designed and numerically tested. The main drawback of this solution is the required actuation forces, significantly higher compared to other solutions due to inertia, since actuation is applied on a heavy and stiff region (holder). Therefore, in the most common applications [18,19] (Figure 3), actuators are integrated in the tool body and actuation direction is parallel to the tool axis. In this scheme, motion normal to the tool axis is achieved by exploiting the moment (M_{act}) generated by applying the actuator forces (F_{act}) with a specific arm respect to the tool axis.

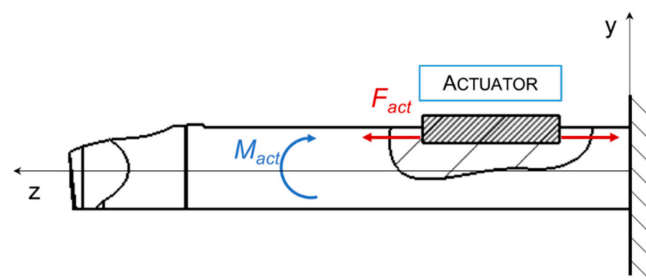


Figure 3. Chosen actuation strategy.

Although the solution is the most feasible (applying the actuation force directly on the most flexible directions is complicated), it shows limitations due to the need of balancing two opposite

needs. On the one hand, the actuation effectiveness increases with the distance between actuator and tool axis, on the other hand this distance is limited by technological requirements.

Since actuation system effectiveness is limited by the maximal radial dimensions, actuators with high energy density (i.e., force/volume ratio) and capable of providing an adequate actuation force in the required frequency range (e.g., 0–1 kHz) are required. Based on these requirements, piezoelectric actuators represent a preference choice. Two types of actuators may be employed: patches (or layer) or stack (or multilayer). Thanks to their flexible structure, the first ones can also be mounted on a curved surface and they do not required preload, their size is reduced and they do not require dedicated housing slots on the bar. However, their application in an active boring bar is very limited, since they cannot provide adequate actuation forces.

On the contrary, multilayer actuators are able to generate higher loads and are suitable for dynamic applications, however, due to necessity to protect them from shear and tensile stresses, their integration on the boring bar is more complicated. Even if solutions with piezoelectric actuators were presented in literature [10], a systematic approach for active boring bar design that covers all the most crucial aspects has not yet been developed. In particular, the following open issues were highlighted:

1. Solution for actuators integration and placing to optimize the system effectiveness considering the inertia of the system and electronics, two actuations directions and available space.
2. Solution for actuators housing protecting them shearing stress, positive tensile stress and from chip and debris.
3. A strategy that guarantees the dynamic and static equivalence between active boring bar and standard boring bar.

Proposed approach supports active boring bar design considering all these aspects. In particular, the first goal was achieved and presented in Section 2.3. A proposed solution to protect actuators from shear and traction stress is presented in Section 2.4. The needs of protection system for actuators, while guarantee static and dynamic equivalence between standard and active boring bar are combined by means of the employment of protective cover properly designed, as presented in Section 2.5.

2.3. Proposed Approach for Actuators Positioning

Starting from the actuation solution in Figure 3, a strategy for actuators placing, in order to maximize effectiveness, while satisfying requirements, was developed.

As mentioned, one of the main advantages of active damping techniques is their versatility. Using these approaches, it is possible to improve chatter stability in different tool configurations (i.e., overhangs). Since in the proposed approach actuators are integrated on the boring bar, in order to maximize range of available overhangs, the configuration with one actuation section, placed as close as possible to the tooltip, is selected.

In accordance with the requirements presented in Section 2.1, the integration strategy of the actuators on the boring bar body must guarantee: (i) the actuation in both radial and cutting velocity direction; (ii) maximum effectiveness (i.e., maximize the harm of the forces generated by the actuators); (iii) limited radial size; (iv) dynamic and static equivalence. In the proposed strategy, requirements (i), (ii) and (iii) are achieved by means of a section optimization (presented in this section), while the last requirement is achieved by properly dimensioning protective covers in order to compensate the loss of stiffness due to material removed (Section 2.5).

In order to find the optimal actuator position, different configurations were evaluated. In Figure 4a, the most effective configuration in case of one actuation direction is presented; however this design provides low actuation forces and a single actuation direction (y). To increase the effectiveness, multiple actuators can be used, either in parallel (Figure 4b), or rather on two sides with opposite actuation forces (Figure 4c), optimal placement to reduce the required space. To achieve a multidirectional actuation, the solution showed in Figure 4d involves the employment of two actuators for each damping direction. In particular, actuators named X1 and X2 act to damp the vibration along the x direction, while Y1

and Y2 damp the vibration along the y direction. Since this solution exploits two actuators at time, effectiveness can be further increase by activating all the four actuators, as presented in the selected configuration (Figure 4e) that includes the four actuators on 45° position respect to the two directions. Using this configuration, the actuation in x direction is achieved by A and B working with opposite signal of C and D, while actuation in y direction is achieved by A and D working with opposite signal of B and C. As result, even if actuators distance from the tool axis is reduced, using double actuation forces increases the resultant moment.

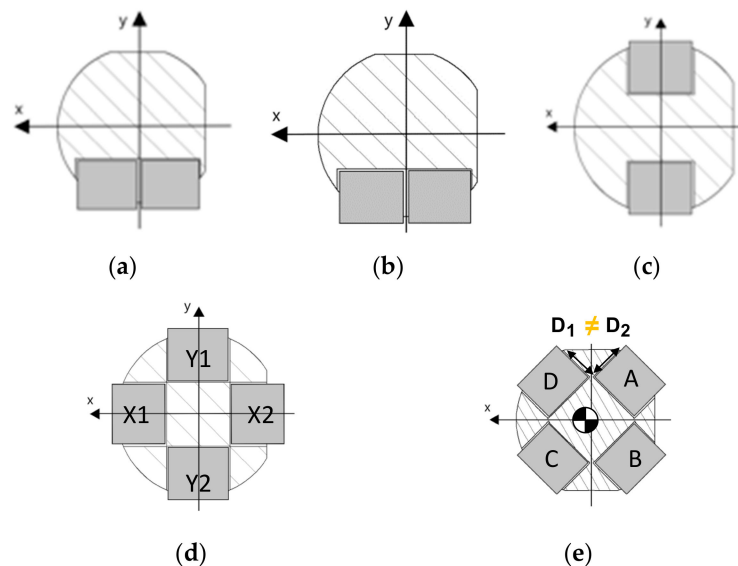


Figure 4. Evaluated solutions of (a) one actuator (b) two parallels actuators, (c) two side actuators, (d) four actuators along x and y direction and (e) selected configuration.

Optimal radial position (i.e., radial distance of actuators from tool axis) allows to maximize the arm of the force, while respecting radial imposed size. Moreover, the centroid of modified boring bar section should be aligned with the one of the original sections, in order to avoid different preferential direction of inflection among the different section. This implies that, in case of a boring bar section that is not symmetric within the x and y direction, actuators radial position has to be different (Figure 4e).

Actuators are then placed at the minimum distance from tool axis that allows to reduce stress concentration due to the shape of the edge, and thickness of each slot was calculated in a way that the inertial centroid of modified section is the same of the one of original section.

In order to guarantee the same static and dynamic behavior of the original boring bar, the loss of stiffness due to material removed must be compensate. Since the presence of actuators does not provide enough stiffness to compensate material removal, additional components are required. In order to reduce the number of components, protective covers, required to avoid inclusion of chip and debris on the housing slot, can be properly dimensioned to achieve this goal. The methodology for optimal protection cover dimension is presented in Section 2.5.

2.4. Actuators Housing Strategy and Preload Estimation

To protect the actuators against lateral forces, spherical end-tips are adopted (Figure 5a). The main advantage of this configuration are: (i) since the joint between actuator and slot can be schematized by a hinge, shear stress are not transferred to the actuators (Figure 5b) (ii) thanks to the spherical shape of the components, the contact between actuators and boring bar is guaranteed even in case of tool deflection (Figure 5c) and (iii) the correct alignment of actuators, after tool deflection, is guaranteed.

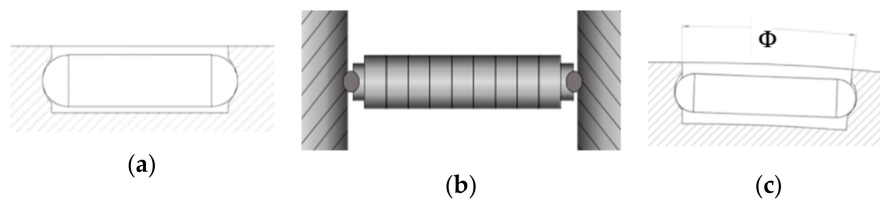


Figure 5. (a) Housing strategy; (b) schematized configuration; (c) contact between actuators and boring bar in case of tool deflection.

In order to protect piezo electric actuators from tensile stress, actuators must be preloaded [20]. Manufacturers suggest a value for a basic preload that is the minimum preload in case of no additional forces act on the actuator.

In our case, due to tool deflection, the preload required is the sum of basic preload and an additional preload able to compensate the distancing of the hemispherical slots, in which actuators are installed. The most critical condition is identified by chatter, where cutting forces, as well as distancing of the two sections, are maximum. Cutting forces in chatter condition may be measured or estimated, while relative displacement between interface sections can be computed using a Finite Element (FE) approach: exploiting the condition of dynamic equivalence between original and modified boring bar (requirement 3. in Section 2.1), the model of the original boring bar can be used to estimate the maximum distancing of the section in correspondence of the contact surface with actuators. An impulsive force with the same amplitude and direction of the maximum cutting force in chatter condition should be applied at the tool tip (i.e., where cutting forces are exchanged) in order to estimate frequency spectra of displacement in correspondence of the interfaces between tool and actuator. Then, minimum additional preload should be calculated as the force able to compensate maximum displacement. In the case when the boring bar section is not symmetric within the x and y axis, global preload must be adjusted in order to obtain a null resultant moment on the section.

Once preload is estimated, the most effective and easy to apply preload system was detected. The most adopted solution in literature is a mechanical mechanism composed by flexure, moving platform and screw [20] (Figure 6a). This solution ensures the adjustment of the preload, but the presence of different moving components affects the effectiveness of the actuator, since part of the motion of the actuator is reduced by the flexibility of the additional components. Considering the importance of the effectiveness in the defined application, a different mechanism that exploits mechanical interference is proposed. In the proposed design, the actuator is forced inside the housing with a defined interference that preloads the actuator inside the slot. This solution allows to reduce the number of components required, hence increasing the actuator effectiveness. In the proposed design, preload is applied using a threaded insert with a spherical housing, as presented in Figure 6b. The advantages of this solution are that it is easy to assemble, and the required preload can be easily obtained by a proper design of the insert. However, interference must be accurately identified, since preload cannot be adjusted or changed.

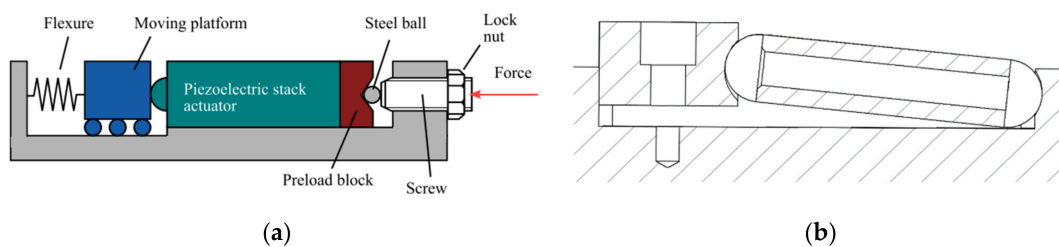


Figure 6. Preload mechanism: (a) mechanical [20]; (b) by interference (selected solution).

The correct interference must compensate the deformations, due to preload, of actuators and slots also considering local deformations in the contact area between actuators and slots. In the

proposed approach (schematized in Figure 7), each actuator is modeled as a single lumped stiffness (K_P), while each slot is modeled as a three lumped stiffness in parallel, in order to consider both boring bar (K_B) and contact stiffness (K_H).

K_H can be computed based on Hertzian theory. In particular, given compression force, material property of the two bodies in contact and their geometry, it is possible to estimate contact circle radius and depth of penetration, and then its equivalent stiffness. K_P can be experimentally measured, while boring bar stiffness (K_B) is evaluated by means of an FE approach estimating the slots elongation due to computed preload applied in correspondence with the estimated contact area.

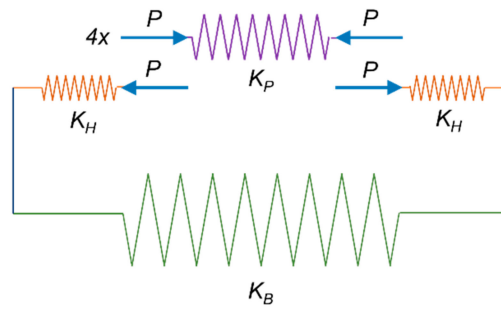


Figure 7. Proposed model for interference estimation.

To guarantee the contact between actuator and boring bar, slot length must be equal to actuator length. Due to preload P , actuator length decreases in x_p , while the distance between the wall of the slot increase as a result of both the elastic yielding due to contact between them (x_H) and of the boring bar deformation (x_B). Then, assuming a slot of nominal length L_0 and an actuator of length L_p , the following equation must be respected:

$$L_0 + x_B + 2x_H = L_p - x_p + i. \tag{1}$$

In order to guarantee the contact between slot and actuator, interference should be able to compensate the relative displacement between them.

$$i = x_B + 2x_H + x_p. \tag{2}$$

In accordance with the model proposed in Figure 7, interference (i) may be expressed as:

$$i = \frac{2P}{K_H} + \frac{4P}{K_B} + \frac{P}{K_P}. \tag{3}$$

2.5. Protective Covers Proposed Design Approach

As mentioned before, the role of the protective covers is twofold: from one hand they protect the actuators from chip debris and cutting fluids, from the other hand they may be used to compensate the loss of stiffness on the section due to the presence of slots. The protective cover should be designed to fulfill these objectives, while not exceeding radial encumbrance. The most suitable solution was found to be a single cover for each actuator screwed on the boring bar body. Moreover, to reduce dimensions and decrease the number of components, covers can be integrated with the insert used to provide preload (Figure 8).

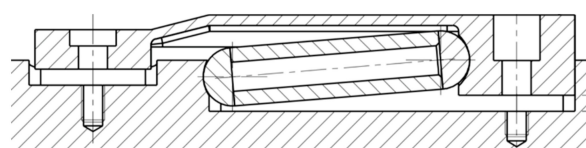


Figure 8. Proposed solution for covers.

An FE approach is proposed to find the dimensions that allow to compensate the loss of stiffness due to the removed material. By means of active boring bar model, which includes both the actuators and the covers, it is possible to predict its response including inertial effect of all components. Varying covers dimensions and detects the dimension that gives, in case of a switched-off control, a static and dynamic behavior as close as possible to the original boring bar.

In Table 1, a summary of issues, requirements and solutions that the proposed method is able to achieve is presented.

Table 1. Issues, requirements and proposed solution for the design of an active boring bar.

	Issues	Requirement	Proposed Solution
State of the art	How to damp vibration		By means of a moment (i.e., a force applied at a given distance from tool axis)
	How to provide force		By means of multilayer piezo electric actuators
	How to integrate actuators	4, 5	By means of slots on boring bar body
	How to protect actuators form shear stress		Hemispherical tips
	How to protect actuators from positive tensile stress		Preload
	How to protect actuators form chip and debris		Protection Covers
	How to compensate the loss of stiffness due to slots	3	Properly designed covers to compensate the loss of stiffness due to the presence of slots
Proposed methodology		1	By properly choosing actuators (hard-doped ceramic)
	How to optimize actuation effectiveness	2, 3, 4, 5	By properly placing actuators - Axial position optimization - Actuation section optimization (four actuators within 45° within x and y direction)
	How to estimate preload	3	As the sum of basic preload and an additional preload that compensate interface section distancing due to tool deflection. Preload is then adjusted in order to balance the moment
	How to realize required preload		Interference
	How to estimate interference		By means a simplified model
	How to realize interference	4, 5	By means of an insert

3. Case Study

Presented approach was applied to a case study. The active boring bar was designed starting from a commercial boring bar made by Sandvik (S32U-PTFNL16W) equipped with a TNMG160508PM-4325 cutting insert, presented in Figure 9.



Figure 9. Original boring bar.

The boring bar is used to machine holes of 50 mm minimum diameter and 160 mm maximum depth on a CN lathe (Mori Seiki SL-2500Y). Maximum radial size was then set to 40 mm and minimum overhang was set to 163 mm. Preliminary cutting tests on K100 (DIN 1.2080) steel material, which chemical composition and hardness are reported in Table 2, were carried out to measure maximum cutting force and chatter frequency (Section 3.1).

Table 2. K100 (DIN 1.2080) properties.

Workpiece Material	Chemical Composition				Hardness (HB)
	C%	Si%	Mn%	Cr%	
NCE46	2.00	0.25	0.35	11.5	240

3.1. Chatter Force and Frequency Measurement

Using the framework shown in Figure 10, where the boring bar is mounted with 163 mm of overhang, cutting forces and accelerations were measured by means of a Kistler 9257A dynamometer and a tri-axial accelerometer (PCB model 356A32), acquired by an LMS SCADAS SCM202V acquisition system. Spindle speed and feed per revolution were set at 700 rpm and 0.2 mm/rev, respectively, while radial depth of cut was gradually increased until chatter condition was detected. For given values of spindle speed and feed per revolution, the machining was found to be unstable with a radial depth of cut of 1.5 mm. Measured cutting force along the three directions in the time and in frequency are shown in Figure 11a,b, respectively. Maximum force amplitude is around 3800 N along cutting velocity direction. Chatter frequency is around 463 Hz, while correspondent cutting force amplitude on cutting force direction is 430 N (Figure 11b).

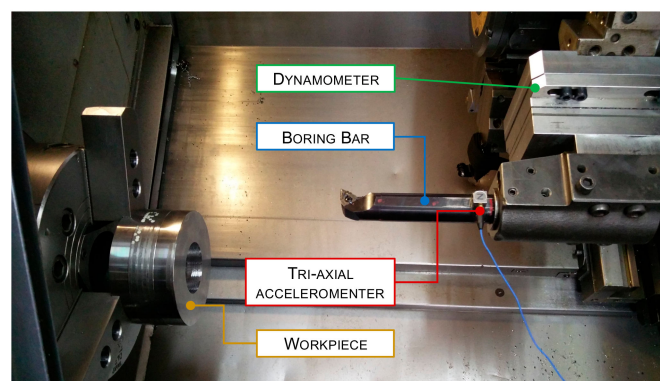


Figure 10. Experimental setup for chatter tests.

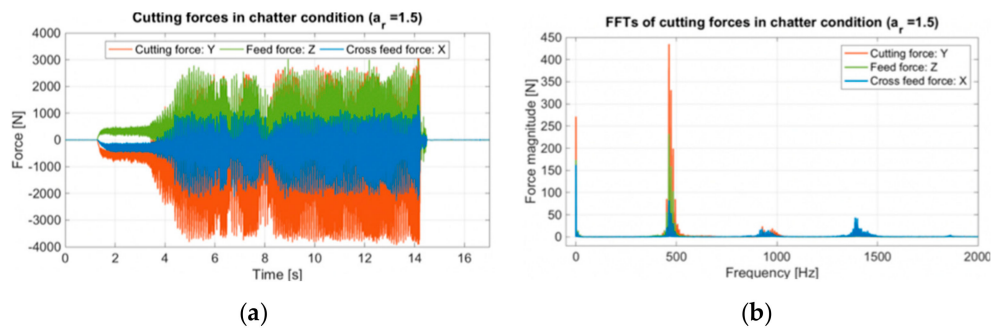


Figure 11. Cutting forces shown in (a) time history and (b) frequency spectrum.

3.2. Active Boring Bar Design

Once chatter conditions were detected, and maximum cutting forces were measured, the active boring bar was designed in accordance with the proposed methodology.

NOLIAC SCMAP-NCE46-10-10-2-200-H36-C01 actuator was found to be the best compromise between force that can be provided, dimensions and response time. In particular, since it is not provided with preload mechanism and case, its size is reduced to 10 × 10 × 36 mm, compliant with the available space. Moreover, the selected actuator is made of hard-doped ceramic that reduces self-heating issues arising at high frequencies. Its characteristics are reported in Table 3.

Table 3. Selected actuator characteristics.

Piezoelectric Material	Dimensions			Free Stroke [μm]	Estimated Blocking Force [N]	Unloaded Resonant Frequency [kHz]
	Length [mm]	Width [mm]	Height [mm]			
NCE46	10	10	36	32.3	3200	30

Actuation section was placed near the tool tip in order to allow the use of the tool with different overhangs. Considering the selected configuration (Section 2.2) maximum slots depth along radial direction, that respect the radial size limit, was found to be 6.5 mm.

The difference in terms of thickness between slots on tool insert side and the opposite one, which guarantees section centroid alignment, was found to be 1 mm (Figure 12).

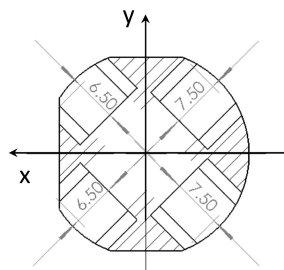


Figure 12. Slots radial dimensions.

Preload required was calculated following the approach presented in Section 2.4, using MSC Nastran® FE solver. The boring bar FE model is showed in Figure 13.

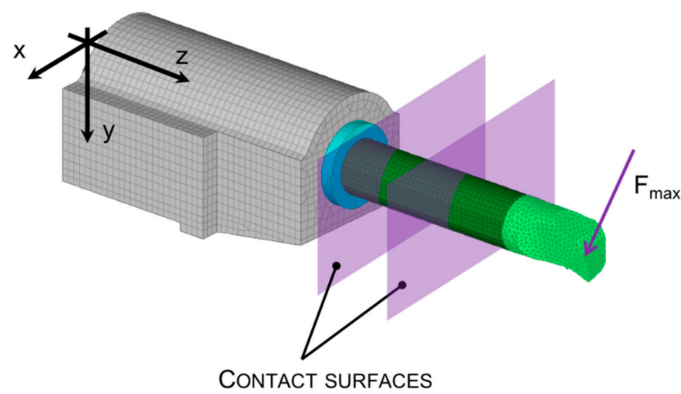


Figure 13. Boring bar model for slots surfaces distancing estimation.

In accordance with Table 4, maximum relative displacement between slot section was predicted to be 8 μm at the measured chatter frequency (463 Hz). The preload that allows to compensate this distancing was estimated in 4 kN. Global required preload is then 5 kN.

Table 4. Maximum distancing of slots sections.

Force Direction	Displacement Direction	Maximum Displacement [mm]	Correspondent Frequency [Hz]
X	Z	0.024	463
Y	Z	0.008	463

Covers are supposed to be jointed to the tool body by means of hexagon socket head cap screws able to provide an adequate force for placing. Hole position was chosen spaced enough from the contact area, where contact stress is less than 0.1% of maximum contact stress. As a result, slot length is obtained as the sum of insert length and actuators length (Figure 14).

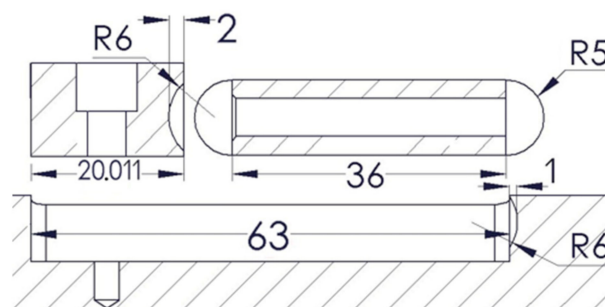


Figure 14. Slots dimensions.

Using the model presented in Section 2.4, interference was estimated for selected preload, based on the equivalent stiffness. Boring bar FE model is shown in Figure 15.

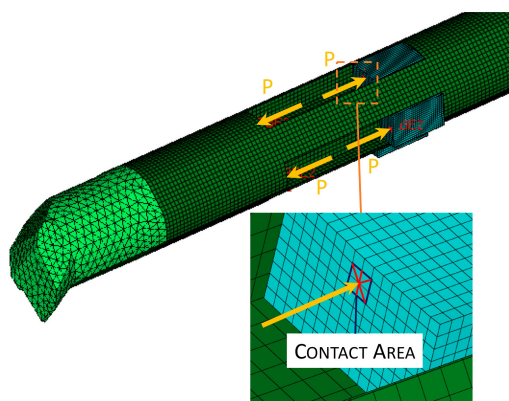


Figure 15. Boring Bar Finite Element model for stiffness estimation.

In Table 5, measured stiffness of actuator and estimated contact stiffness and boring bar stiffness are reported.

Table 5. Actuators, contact, boring bar and equivalent stiffness.

K_P [N/mm]	K_H [N/mm]	K_B [N/mm]
7.20e4	2.39e05	6.39e05

Resulting interference was found to be 0.133 mm.

By means of the FE model shown in Figure 16, according with approach presented in Section 2.5, cover thickness, that guarantees the compensation of material removed for actuators housing, was found to be 1.5 mm (for the two covers on insert side) and 1.8 mm (for the other two covers).

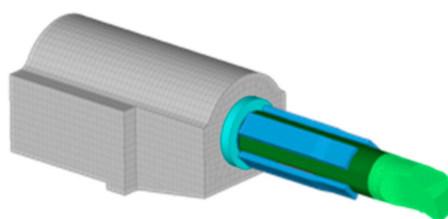


Figure 16. Active boring bar FE model for protective covers design.

In Table 6 properties of the section, in correspondence of actuators, of the different models are reported in terms of cross sectional area (A) and second moment of area with respect to central axis of inertia x and y (J_x and J_y , respectively).

Table 6. Comparison between original boring bar, boring bar with slots and active boring bar with covers.

Configuration	A [mm ²]	J_x [mm ⁴]	J_y [mm ⁴]
Original boring bar	760.38	45,042.61	47,493.32
Boring bar with slots	373.08	16,074.55	17,581.10
Boring bar with covers	514.32	42,032.73	43,644.03

The employment of designed covers compensates the loss of area due to material removed in correspondence of the slots and allows to obtain almost the same inertial properties of original boring bar ensuring a similar static response.

The comparison between the dynamic responses of the different models is shown in terms of tooltip Frequency Response Function (FRF) in Figure 17c.

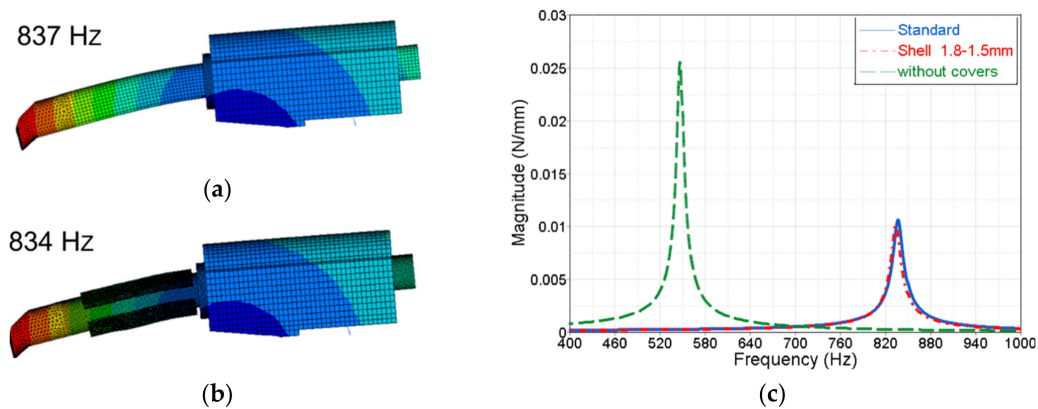


Figure 17. (a) Original boring bar dominant mode, (b) active boring bar dominant mode and (c) comparison between original (i.e., standard) and active boring bar in terms of FRF.

Due to material removal, the dynamic response of the boring bar significantly changes, both in terms of natural frequency and peak amplitude. The employment of designed covers allows to obtain a dynamic response almost identical to the original boring bar. In particular, the difference in terms of natural frequency is less than 1%.

3.3. Active Boring Bar Manufacturing

The designed active boring bar was realized and assembled. The standard bar from Sandvik Coromant (S32U-PTFNL-16W) was machined to obtain the four slots and protective covers were directly milled starting from bar made of 1.2738 steel (40CrMnNiMo8). In order to guarantee the designed interference, manufacturing process for the protective covers was carefully organized. In particular, length of each actuator was measured and the distance between the center of the sphere and the cover back plane of the covers was corrected according to the measured value.

4. Experimental Validation

4.1. Comparison between Original and Modified Boring Bars When the Control Is Off

At first, experimental tests were carried out in order to compare the standard boring bar with the realized active boring bar when the control is off. The aim of this validation is to verify that the designed active boring bar presents an equivalent static and dynamic behavior of the standard one (requirement 3 in Section 2.1). Experimental modal analysis was carried out on both boring bars in free-free (Figure 18a) and constrained (fixed-free) conditions (Figure 18b).

Experiments were performed by means of impact testing using a PCB 08C03 impact hammer and two mono-axial accelerometers (PCB 352C22); a roving hammer strategy was adopted, and required FRFs were computed by means of the dedicated acquisition system (SCADAS SCM202V). Using LMS software, FRFs were analyzed and dominant modes characteristics (mode shapes, natural frequencies) were extracted by means of Polymax algorithm [21].

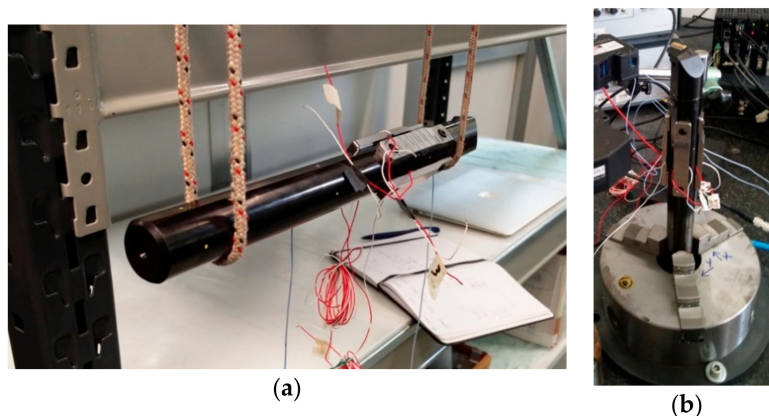


Figure 18. Setup for active boring bar experimental modal analysis. (a) Free-free condition. (b) Fixed-free condition.

Results in terms of dominant mode natural frequencies are reported in Table 7. The difference in terms of natural frequency between original and active boring bar (less than 5%) indicates a good match between the two systems. As a consequence, the effectiveness of design approach to achieve dynamic equivalence was demonstrated.

Table 7. Comparison between original boring bar and active boring bar.

Condition	Standard—Natural Frequency (Hz)		Active—Natural Frequency (Hz)	
	X	Y	X (Error)	Y (Error)
Free-free	1190	1218	1134 (−4.7%)	1169 (−4.0%)
Fixed-free (overhang 245 mm)	329	276	317(−3.7%)	263 (−4.7%)

4.2. Damping System Effectiveness: Dynamic Behaviour of Active Boring Bar When Control Is On

In order to test the designed active system effectiveness, experimental tests were carried out to compare standard and active boring bar dynamic behavior with control on.

4.2.1. DVF Implementation

DVF control logic was implemented in Matlab® Simulink v14 and integrated using VeriStand 2014 in a National Instruments PXIe 1071 controller equipped with a BNC-2110. Velocity signal is acquired by the controller, and the actuation signal is generated by multiplying the velocity of a defined control gain. In order to protect the actuator by overvoltage (± 100 V that corresponds to ± 3 V on the command signal), a saturation block (set at ± 2.5 V) was added before the PXI output.

4.2.2. Sensors Choice and Positioning

Control logic input signal (i.e., velocity) was obtained by integrating the acceleration signal measured using piezoelectric accelerometers (PCB JM357B11, Brüel & Kjær 4393), by means of an external conditioner (Brüel & Kjær 2635). Employment of such accelerometers, characterized by a resonance frequency of 50 kHz, ensures a sufficient synchronism between excitation and response. To make the actuation system able to act in both directions (x and y) two sensors were placed on the boring bar (Figure 19a). Sensor position was chosen in order to realize a “collocated” positioning, so that sensor and actuator act on the same point, as suggested for the application of DVF control logic.

4.2.3. Experimental Setup

In order to evaluate the effect of the actuation system, the prototype was tested with different dynamic forces applied by means of a shaker (Brüel & Kjær 4809), controlled by the LMS SCADAS SCM202V acquisition system. The experimental setup is shown in Figure 19b. The active boring bar was fixed by means of a three-jaws chuck and displacement were measured by means of a laser sensor (Keyence LK-H085) and the sensors installed on the boring bar.

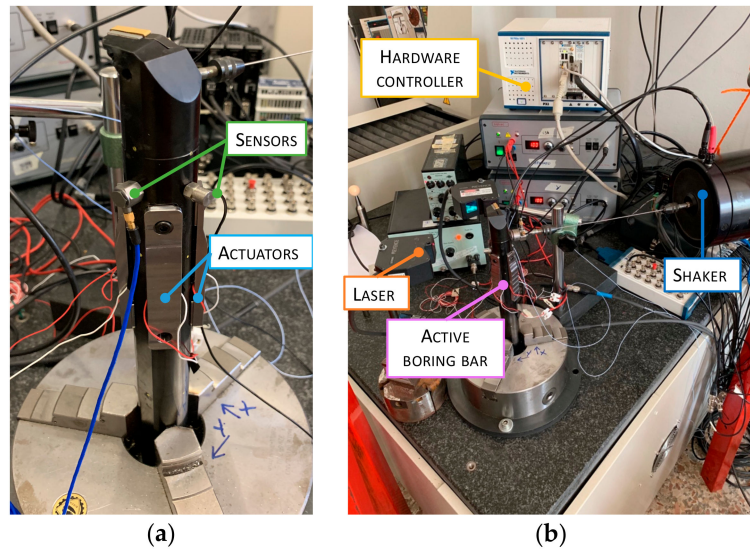


Figure 19. (a) Active Boring bar. (b) Experimental setup.

4.2.4. Experimental Results

At first, the system was excited by single frequency force (pure sine) at 240 Hz (close to the resonance frequency of the system). Results in case of control on and off (Figure 20), show how the active damping system is able to significantly mitigate vibration (reduction of about 85%).

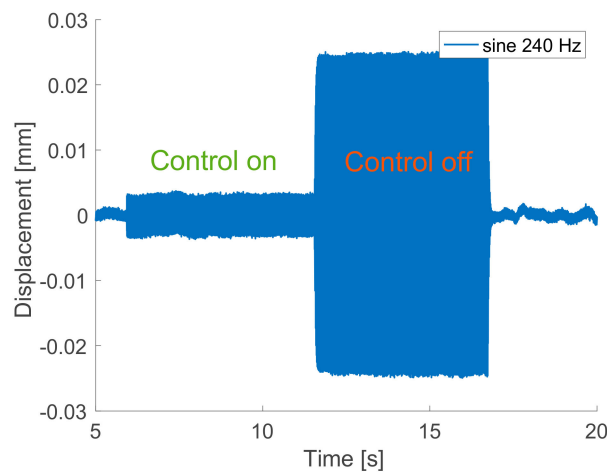


Figure 20. Measured displacement in case of control on and off in case of single frequency force applied.

Then, the system was tested by exciting the active boring bar with a multifrequency force (chirp sine between 0–500 Hz). Results in case of control off and on (Figure 21) highlight the effectiveness of the active system in adding damping to the system: vibrations in the frequency range close to the natural frequencies are significantly reduced. Outside, these frequency ranges in the control do not affect the vibrations level, as expected, and no distortions were found.

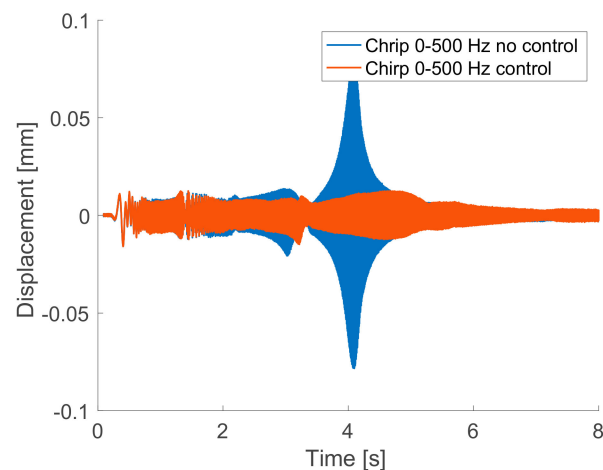


Figure 21. Experimental displacement in case of control on and off in case of multifrequency force applied.

The same result was found in case of impulsive excitation, reproduced by impact testing with instrumented hammer (PCB 08C03) on the tool-tip and acquiring boring bar displacements by using the laser sensors (Figure 22).

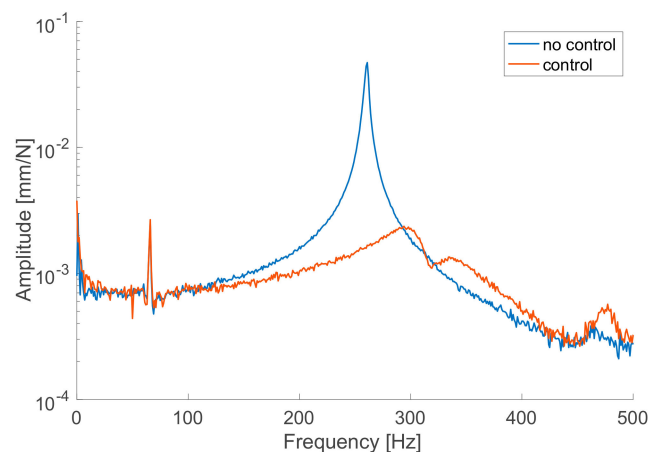


Figure 22. Tooltip FRF in case of control off and control on.

Frequency Response Functions highlight that active boring bar can reduce the amplitude of vibration at the natural frequency of more than 90%. In conclusion, preliminary experimental test on active boring bar confirmed the effectiveness of the active damping system, showing a drastic reduction vibration amplitude in correspondence of the dominant mode shape.

5. Conclusions

In conclusion, an active boring bar design approach tailored for DVF control was presented. The aim of the proposed approach was to provide design requirements and a structured strategy, able to support active boring bar design, considering the most significant criticalities that were yet unaddressed in literature. In particular, the strategy for actuators integration on the boring bar body was evaluated, focusing on:

1. The most effective actuators positioning.
2. Preload estimation in order to protect piezoelectric actuators from traction stress during machining.
3. Preload system design.

4. Protective cover design to guarantees static and dynamic equivalence between the active boring bar and the original boring bar in case of control off.

An active boring bar, designed following this approach, was realized and tested both in the case of control off and of control on. Experimental tests confirmed the benefits of the proposed method, which allowed to:

1. Realize a dynamic equivalence in case of control off, between the original and active boring bar (difference between natural frequency resulted less than 5%), confirming that the proposed approach for protective covers dimensioning is able to compensate the loss of stiffness due to removed material.
2. Provide an actuators integration design strategy able to respect size requirements and realize a significant increase of damping (around 90% of dominant mode amplitude reduction), in case of control on, confirming the effectiveness of actuators integration and placing approach.

Chatter tests using designed active boring bar will be carried out to estimate the impact of the system in terms of chatter stability increases on an actual machining process.

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