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Design, Implementation and Analysis of Composite Material Dampers for Turning Operations

Lorenzo Daghini, Andreas Archenti, and Cornel Mihai Nicolescu

Abstract**—**This paper introduces a novel design for boring bar with enhanced damping capability. The principle followed in the design phase was to enhance the damping capability minimizing the loss in static stiffness through implementation of composite material interfaces. The newly designed tool has been compared to a conventional tool. The evaluation criteria were the dynamic characteristics, frequency and damping ratio, of the machining system, as well as the surface roughness of the machined workpieces. The use of composite material in the design of damped tool has been demonstrated effective. Furthermore, the autoregressive moving average (ARMA) models presented in this paper take into consideration the interaction between the elastic structure of the machine tool and the cutting process and can therefore be used to characterize the machining system in operational conditions.

Keywords**—**ARMA, cutting stability, damped tool, machining.

I. Introduction

VIBRATION control has been and still remains a subject of primary importance in modern manufacturing industry. Removing high volumes of material in shorter time as well as obtaining the right quality from the first part produced are goals that one would like to achieve. Tooling systems, and especially cantilever tools, and cantilever structural units of machine tools are the least rigid components of machining systems and therefore the most prone to vibration that could lead to cutting instability. The objective of this paper is to implement efficient damping devices based on identification of parametric models describing the dynamic stability of machining systems. The present paper focuses on the design and the dynamic analysis of damped boring bar used in internal turning. This operation is widely known to be critical from the point of view of dynamic stability, and it is often carried out with cutting parameters far from optimal. In order to improve process performance it becomes necessary to introduce means of vibration control in the machining system. Vibration control has the purpose to achieve efficient energy dissipation by controlling the damping of the system. Since damping involves the conversion of mechanical energy associated with a vibration to other forms, there are several mechanisms to remove energy from a vibrating system, which will be presented in a later section.

A. Dynamic Evaluation of Machining Systems

In order to understand the principle for design of efficient damping systems it is necessary to understand the dynamic behavior of machining systems. Machining systems may be represented by a closed loop system comprising the machine tool elastic structure (ES), i.e. the machine tool structure including tool, tool holder, workpiece etc., and the cutting process (CP), i.e. turning, milling etc., see Fig. 1 [1], [2], [3]. The interaction between the machine tool's elastic structure and the cutting process describes the behavior of the machining system. This behavior directly affects the process accuracy.

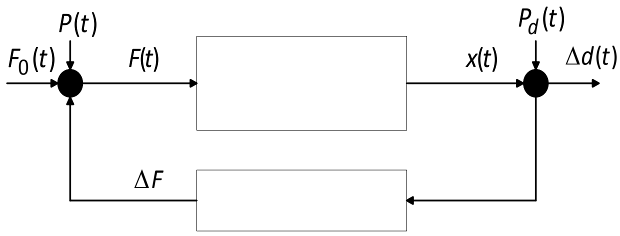

Fig. 1 A typical representation model of a machining system from a process-machine interaction point of view

In Fig. 1, $F(t)$ is the instantaneous cutting force, $F_0(t)$ is the cutting force nominal value, $x(t)$ is the relative displacement between cutting tool and workpiece, $\Delta d(t)$ is the total deviation of the relative displacement $x(t)$. $P(t)$ and $P_d(t)$ are disturbances such as tool wear, thermal dilation of the elastic structure, variation of rigidity of the elastic structure during a machining process, variation of cutting parameters etc. Critical factors for optimization of designs for damped structural components for machine tools are the modal parameters (frequency and damping ratio). These parameters, that also control the stability of the cutting process, can be extracted from the analysis of the interaction between the two subsystems. Traditional evaluation of machining system dynamic behaviour has invariably been approached in the following steps:

1) Identification of the dynamic properties of elastic

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structure of machine tools. Generally this step is done experimentally often using experimental modal analysis (EMA).

2) Identification of the characteristics of cutting process, i.e. the dynamic parameters describing the transfer function of the subsystem represented by cutting process in Fig. 1.
3) Evaluation of dynamic stability of the machining system from step 1 and step 2.

Because the evaluation of dynamic stability and of modal parameters, carried out following the above approach, does not take into consideration the actual operation conditions, the results are affected by errors. Furthermore, the test described above makes use of external forces that have different nature than cutting forces. The separation in the two subsystems does not take into consideration the mutual interaction between the two subsystems in real cutting operations. In this paper an approach for dynamic evaluation of boring bar based on a novel model-based identification method, that takes into consideration the above mentioned interaction, will be introduced [4]. The term identification is applied to a procedure to formulate an analytic model of the machining system based on the analysis of the "real" signals collected from the cutting processes. This approach captures the interaction between the machine tool structure and the cutting process. This paper introduces a type of identification and parametric modeling that relies strongly on statistical methods because of the random nature of the cutting process characteristics. The parametric models used here are based on stochastic processes and a special class within this family is defined by autoregressive moving average or ARMA models [5], [6], [7]. ARMA models offer an acceptable trade-off between flexibility and parsimony with respect to the number of model's parameters. There are several benefits arising from employing parametric models for analysis of dynamics of machining systems. ARMA models are characterized by high frequency resolution and objective assessment of component significance. This is in contrast to non-parametric models such as Fourier transform techniques where the frequency resolution is approximately the inverse of the available data time length. Parametric models can be used to analyse transient data sequences generated by sudden arising of cutting instability (chatter) [8].

This paper will first explore the state of the art in control of machining vibration, then the concept followed in the design phase of a boring bar with enhanced damping capability will be described, and, finally the performance evaluation of the prototype through experimental modal analysis (EMA) and identification of process-machine interaction characteristics (frequency and damping ratio) will be presented.

II. State of the Art in Control of Machining Vibration

Basically, there are two main categories of vibration control systems: active and passive. Active control requires sensors to detect the vibration, an electronic controller to process the signal from the sensors and actuators to interfere with the mechanical response of the system. From the actuators a secondary oscillatory response is generated in order to cancel the original response of the system causing cutting instability. Such active control systems need cables for data transfer and energy supply that can interfere with the machining process. Active control systems have been proved to be efficient in laboratory environment but its industrial application has not been welcomed by the end-users due to the complexity of the hardware. The passive control technique does not need complicated hardware and the end-user does not need to introduce new handling routines. Implementations of passive damping in tooling equipment are already available on the market.

A. Active Control

The objective of active vibration control is to reduce the vibration of a mechanical system by automatic modification of the system's structural response. The principle of active control of vibration in machining is to analyze in real time the signal emitted during machining, recognize instability (chatter) and compensate for it. For this purpose different techniques can be used. One way is to predict the arising of chatter and consequently change the cutting parameters before the full instability occurs. A first strategy for automatic chatter recognition and online modification of the cutting speed to a stable area was proposed by Weck [9]. This idea has been further developed in later researches [10], [11], where the signal emitted by the cutting process is sensed and used to recognize chatter, the cutting speed is modified thereafter. Another approach for active control is to compensate in real time for the dynamic forces that arise during the cutting process. Harms et al. [12] suggest a tool design with piezoelectric actuators and force sensors with interchangeable tool head. Browning et al. [13] propose an active control system for boring bars using accelerometers on the tool for providing the controller with both reference and error signal. The signals are processed and sent eventually to the actuators located in the tool clamp, which compensates by providing dynamic forces to the boring bar. The apparent advantage of the active vibration control approach is the perfect adaptability to the changes in the cutting conditions; all the above mentioned techniques are based on online adaptation to the ongoing process to ensure stability. The drawback of this approach is the required computation resources and hardware: the system has to process the acquired signal for chatter recognition in real time, and the amount of data can be large. In addition to this, the presence of cables between the control system and the tools could compromise the machining operation.

B. Passive Control

The principle of vibration passive control is to convert the mechanical energy into some other forms, for instance heat. A common way to achieve passive damping is by using viscoelastic (VE) composite materials to dissipate the energy that causes vibration. The use VE composite materials for damping purposes is quite common, this technique has been used in other fields of application, such as automotive and aeronautics [14]. VE composite materials are used for damping enhancement generically in three different ways: as free-layer dampers (FLD), as constrained-layer dampers (CLD) and in tuned viscoelastic dampers (TVD) [14]. The

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latter has been successfully adapted for designing tooling systems [3]. The basic principle of TVD technique is to add a mass residing on a spring and a viscous damper at the point of maximum displacement. This additional single degree of freedom (SDOF) system must have the natural frequency close to that of the boring bar in order to transfer the vibrational energy to the TVD. If the damper is properly designed it will dissipate the mechanical energy [15]. A solution is proposed by Rivin et al. [16] where the weight is integrated in the tool, hanging on rubber rings. The absorber is tuned by changing the stiffness of the additional system. Another example of application of TVD principle is proposed by Lee et al. [17], the TVD is, in this work, tuned by changing the inertia mass. TVD technique has been presented even for milling operations by Rashid et al [18]. TVD technique is already successfully used in several successful commercial products. Rashid [3] presents as well a solution with integrated damping interface applied to workholding systems for milling operations. When implementing such a solution it is of vital importance for the design to properly locate the pre-stressed VE composite layers in the structure to optimally exploit the property of VE material to give largest deformation in shear [19].

III. Design of the Boring Bar

A. Concept

The principle followed in this research was to enhance the damping ability of critical structural components of the machine tool, minimizing the loss of stiffness. In this paper the focus is mainly on the boring bar tool holder and, indirectly, on the tool clamping system. To maintain a high level of static stiffness, it was chosen to adapt hydrostatic clamping systems to the boring bar. The effect of this kind of clamping system on the tool deflection is well recognized [20] and can be straightforward computed. If the tool is considered to be a cantilever beam, with infinite clamping stiffness, then its transversal vibration will be derived from the equation:

$$\frac{\partial^2}{\partial x^2}\left(EI\frac{\partial^2 v(x,t)}{\partial x^2}\right) = -\rho A\frac{\partial^2 v(x,t)}{\partial t^2} \tag{1}$$

Where E is the Young modulus, I the moment of inertia, $v(x,t)$ is the transversal deflection function, ρ the linear density and A the section of the beam. The transversal deflection of the beam can be calculated by:

$$\Phi(x) = A_1 \cdot \left(\cos(ax) - \cosh(ax) - \frac{\cos(aL)+\cosh(aL)}{\sin(aL)+\sinh(aL)}\cdot(\sin(ax)-\sinh(ax))\right) \tag{2}$$

where $\Phi(x)$ is the shape function, A_1 is a real constant derived from the boundary conditions at the ends of the beam, a is a constant related to the beam's mass and stiffness, and L is the beam's overhang. The true overhang of a tool clamped in a hydrostatic clamp is equal to the effective overhang, while in the conventional screw clamp there is a difference that can be significant for tools with higher ratio L/d (see Fig. 2).

Considering that the overhang of the tools studied in this research is 120 mm, the overhang of the tool clamped in a conventional clamp would result 10% longer. The improvement in maximum deflection at the tool tip can be calculated using (2), considering that the boundary conditions are the same and that x will be equal to the overhang, thus:

$$\frac{\Phi_h}{\Phi_c} = \frac{\cos(aL_h)\sinh(aL_h)-\cosh(aL_h)\sin(aL_h)}{\cos(aL_c)\sinh(aL_c)-\cosh(aL_c)\sin(aL_c)} \cdot \frac{\sin(aL_c)+\sinh(aL_c)}{\sin(aL_h)+\sinh(aL_h)} \tag{3}$$

$$\frac{\Phi_h}{\Phi_c} = 0.8315 \quad \text{with} \quad L_c = 1.1\cdot L_h \quad a = 1 \quad \text{and} \quad L_h = 1 \tag{4}$$

where L_c, Φ_c, L_h and Φ_h are the overhang and the maximum deflection of the conventional clamp and of the hydrostatic clamp respectively. Theoretically, an enhancement of about 17% can be achieved. Pertaining to damping, the chosen approach is carried out by implementation of VE composite material damping layers in the tool structure.

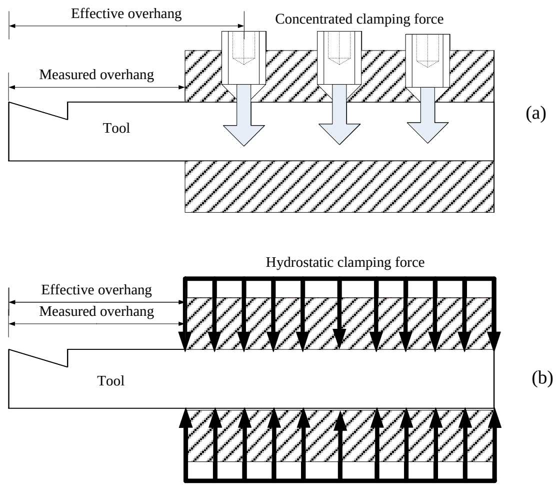

Fig. 2 Comparison between conventional screw clamp (a) and hydrostatic clamp (b)

A general issue to consider when designing dampers with VE composite material is the mechanical impedance between coupled structural elements which control the energy flow path through the structure. It is important that the most of the energy flows through the damper. If the energy has an alternative path of propagation with lower mechanical impedance (Fig. 3(a)), the energy will by-pass the damper [21]. For this reason the damping interface should be the only structural component in the energy flow path, see Fig. 3(b). The use of hydrostatic clamps improves the design by an increase of the contact surfaces compared to those obtained with the conventional screw clamps, and by an evenly distribution of the clamping force over the tool circumference. In addition to this, this design contributes to a uniformly energy distribution in the damping, in the contrary to the screw clamps where the energy would concentrate in small contact areas.

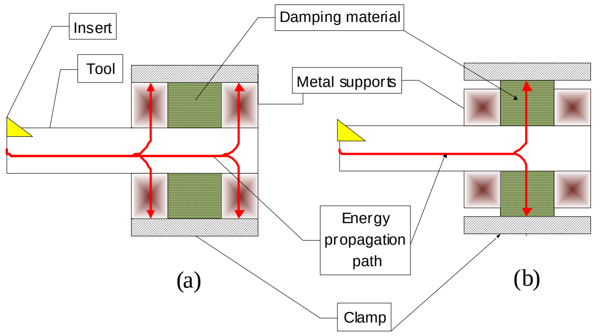

Fig. 3 Energy propagation paths. (a) Bypass through metal-to-metal contact. (b) No bypass, energy flows through damping material

B. Final Design

The boring bar used in this work has a diameter 25 mm with milled facets for clamping on conventional VDI adapter with screws. The damped tool has been produced by placing damping rings on the tool shaft (see Fig. 4(a)). These rings are made of VE composite material composed of a 0.26 mm thick aluminium foil and a 0.12 mm thick viscoelastic material layer.

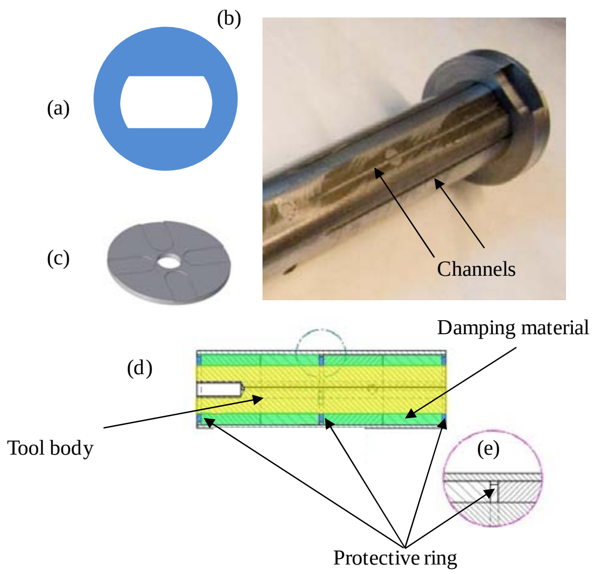

Fig. 4 Mechanical components of the tool: (a) damping ring shape; (b) channels for the glue flow; (c) specially shaped ring for allowing the coolant flow through the tool; (d) section of the damping structure; (e) the protective discs are designed to not touch the external collet

Three protective steel rings, to prevent the deformation of the damping rings, are also placed on the shaft. Two of the protective rings are positioned at the extremities of the damping structure while the third is in the middle (see Fig. 4(d)) The damping assembly has been glued on the tool, where small channels have been milled in order to let the adhesive cover the area homogeneously (see Fig. 4(b)). To protect the damping assembly from the coolant, a specially shaped ring has been fitted to the bottom of the tool where channels have been milled to direct the coolant flow through the tool (see Fig. 4(c)). The connecting screw is provided with a coolant channel as well. A collet with an external diameter of 42 mm is glued on top of the whole structure. This design allows for the clamping to be completely isolated from the tool, i.e. all the energy generated during the cutting process flows through the damping material [21] (see Fig. 4(e)). The complete tool assembly is shown in Fig. 5

Fig. 5 CAD model of the boring bar

IV. Performance Evaluation

A. Analysis method

The evaluation of the novel design compared to the conventional tool has been carried out in two steps: at first EMA has been employed to extract modal parameters for the tool and clamping. Then sound recorded from machining tests has been processed by LMS Test.Lab to extract the dynamic characteristics of the process-machine interaction. The sound recorded by the microphone can be shown to have good correlation to the vibration generated during machining [10] and also have the practical benefit of not interfering in the working zone. The surface roughness has been measured after every test with a Mitutoyo Surftest SJ-301, and correlated to the vibration signals. The model-based identification procedure used to identify the dynamic characteristics of the machining system is performed as follows: first the model's parameters are estimated. The frequency and total damping ratio (combined elastic structure and machining process) are then statistically computed from the model parameters and used as discrimination features. The desired mathematical model of the machining system is based on the data obtained during normal operational conditions. By this, a step beyond the classical way to analyze the dynamics of a machining system by separately identifying the structural and process parameters respectively is taken. In the identification procedure, the estimation of physical parameters, angular frequency, ω, and damping ratio, ξ, can be used for dynamic stability evaluation [2].

The motion of an n degree-of-freedom system can be represented by a system of second-order differential equations:

$$M\ddot{x}(t) + C\dot{x}(t) + Kx(t) = f(t) \tag{5}$$

where x is the displacement function of time and $f(t)$ is the excitation force. If the model of an ARMA process is represented by (5) then the autoregressive (AR) parameters are related to M (mass matrix), C (damping matrix) and K (stiffness matrix) through the characteristic equation of the form:

$$\sum_{t=0}^{2m} a_i \mu^{2m-i} = \prod_{i=1}^{m}\left(\mu - \exp\left\{\xi_j \omega_{nj} \Delta t + i\omega_{nj}\sqrt{1-\xi_j^2}\,\Delta t\right\}\right)\times\ldots$$
$$\ldots\times\prod_{i=1}^{m}\left(\mu - \exp\left\{-\xi_j \omega_{nj} \Delta t + i\omega_{nj}\sqrt{1-\xi_j^2}\,\Delta t\right\}\right) \quad (6)$$

where $\xi_j \omega_{nj}$ and $\pm i\omega_{nj}\sqrt{1-\xi_j^2}$, j=1… n are the eigenvalues of (5). The structural parameters ξ_j and ω_{nj}, j=1... 2n, may be determined from the AR parameters estimates

$$\xi_j \omega_{nj} \Delta t = -\frac{1}{2}\ln(x_j x_j^*)$$
$$\omega_{nj}\sqrt{1-\xi_j^2}\,\Delta t = \tan^{-1}\left|\frac{x_j - x_j^*}{x_j + x_j^*}\right| \quad (7)$$

where x^* denotes the complex conjugate of x. It is important to stress that in the context of stochastic modelling, the estimated physical parameters are meaningful only from a statistical point of view, i.e. they are properly significant within a certain confidence interval. The mapping process from ARMA parameter domain to the $\omega_{ch} - \xi$ domain gives the advantage of a robust chatter identification criterion. Theoretically, dynamic instability can be defined in terms of negative damping. When damping approaches zero the system reaches stability threshold. Model identification algorithms were implemented in Matlab/Simulink.

B. Experiments and Results

1) EMA

The experimental modal analysis has been carried out in two different setups: with the tools clamped in a conventional adapter (with screws) and with a hydrostatic clamp. The conventional and the damped boring bars show almost the same natural frequency when clamped on conventional adapter with screws. The conventional tool has a higher compliance magnitude (lower dynamic stiffness, see Fig. 6).In addition Fig. 6 shows that the damped tool has the largest dynamic stiffness and damping ratio (the curve is blunt at the top) when clamped in a hydrostatic clamp. The conventional tool improves the dynamic stiffness (but not the damping ratio) when clamped in a hydrostatic clamp. It is also noticeable from Fig. 6, that the damped tool has a relatively lower static stiffness which is compensated by much larger damping ratio. As it can be noticed, the damping ratio has been enhanced by approximately ten times.

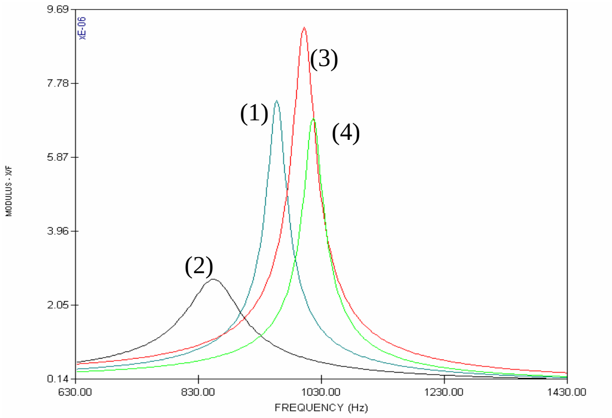

Fig. 6 Result of the modal analysis: compliance. (1) Damped boring bar in VDI adapter with traditional screw clamp. (2) Damped tool in a hydrostatic clamp. (3) Conventional tool in VDI adapter with traditional screw clamp. (4) Conventional tool in a hydrostatic clamp

2) Machining tests

The tools have been tested clamped in the same clamping configurations as modal analysis. Round workpieces made of TOOLOX® 44 and SS2541, respectively, with a diameter of 150 mm and a length of 170 mm were machined. The operation was internal turning and the starting inner diameter was 48 mm. The machining operations were carried out at three different depths of cut (a_p), 1 mm, 2 mm, 3 mm, keeping constant cutting speed (v_c) at 120 m/min and feed (f) at 0.15 mm/rev. The effect of the tool's damping ratio on the machining process is shown in Fig. 7 and Fig. 8, where the acoustic signals produced by machining with the conventional and the damped tool respectively, are compared. Fig. 7 shows the comparison when machining at a depth of cut (a_p) of 1 mm and Fig. 8 the comparison of the two signals when machining at a_p = 2 mm, keeping the other cutting parameters constant.

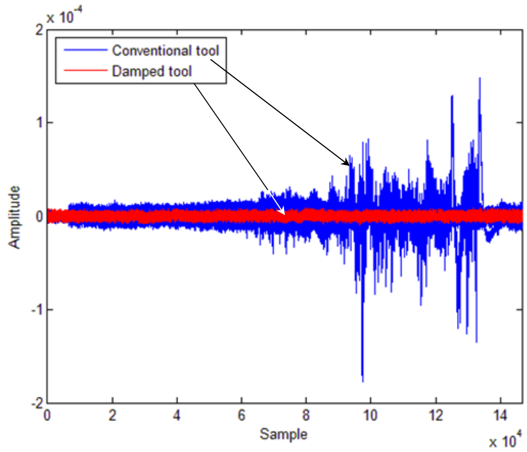

Fig. 7 Acoustic signal produced during machining of SS2541 with a_p = 1 mm, v_c = 120 m/min and f = 0.15 mm/rev. Comparison between the signal produced when machining with damped tool and with conventional tool, respectively

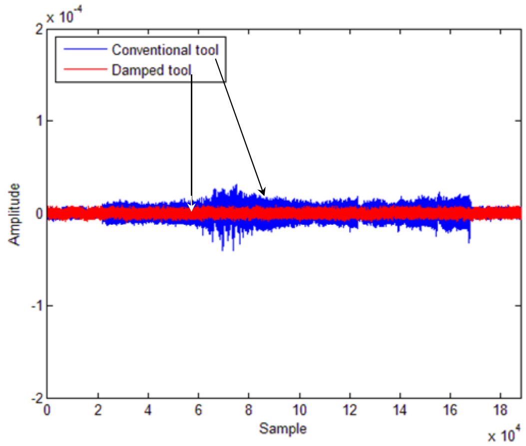

Fig. 8 Acoustic signal produced during machining of SS2541 with a_p = 2 mm, v_c = 120 m/min and f = 0.15 mm/rev. Comparison between the signal produced when machining with damped tool and with conventional tool, respectively

Machining with conventional tool showed the typical signs of instability, i.e. high and irregular amplitudes. The same conclusion can be drawn by looking at the power spectrum density (PSD) diagram of the signal. The PSD, represented by the waterfall diagram in Fig. 9, shows that the energy is concentrated at the tool's natural frequency (1060 Hz, as identified by EMA) and its harmonics, with non-significant participation of other frequencies [2].

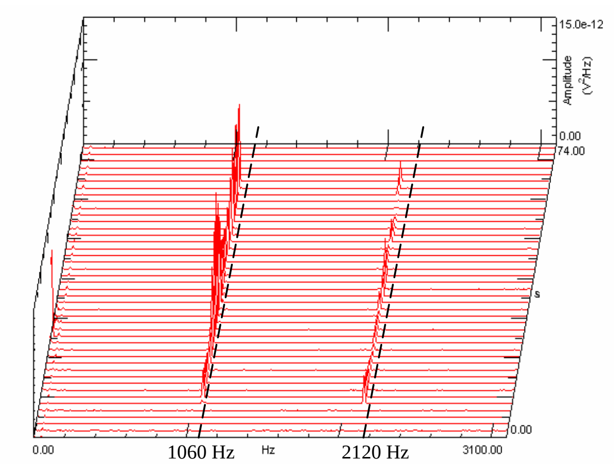

Fig. 9 PSD (waterfall representation) of the acoustic signal produced during machining with conventional tool at a_p = 1 mm

3) Model-based identification by parametric ARMA models

A quantification of the effectiveness of the damped tool during the cutting operations has been further derived by using model-based identification on the acoustic signals. From dynamics of machining system point of view, important physical characteristics are excitation frequencies and the overall damping of the total system, elastic structure-cutting process. ARMA models, of an order determined by minimizing Akaike´s information theoretic criterion (AIC) [22], are fitted to the acquired signals by help of a Gauss-Newton algorithm [23]. From the estimated ARMA model parameters, dynamic characteristics of the machining system are calculated. In the identification approach the response signal is fitted into ten different models evenly distributed over the acquired data. The model's window is set to 10000 samples to ensure a stable parameter estimation. The dynamic parameters, frequencies ω_n and damping ratio ξ_n, for each model (see Fig. 7 and Fig. 8) determined by solving the corresponding ARMA characteristic equation (6). Then, each pair of roots corresponding to excitation frequencies (1060 Hz for conventional tool and 805 Hz for the damped tool) and damping ratio, respectively, were identified. The most significant identified pairs of ARMA(3,2) models are presented in Table I and depicted in Fig. 10. Fig. 10 shows the damping ratio variations as the tool enters the workpiece (at the left end of the figure) and exit close to the chuck (at the right end of the picture). The damping ratio for the conventional tool (see Table I and Fig. 10) goes to zero during the machining of the whole length of the workpiece. The damped tool worked in stable conditions throughout the whole length of the workpiece with a tendency of stability increase as the tool approaches the chuck.

TABLE I
IDENTIFIED FREQUENCY AND DAMPING RATIOS

#	Damped tool a_p=1 mm freq.[Hz] /damping ratio	Conventional tool a_p=1 mm freq.[Hz] /damping ratio	Damped tool a_p=2 mm freq.[Hz] /damping ratio	Conventional tool a_p=2mm freq.[Hz] /damping ratio
1	804/0.2905	1065/0.0002	805/0.2456	1010/0.0025
2	804/0.1834	1064/0.0001	804/0.3315	1063/0.0002
3	805/0.2383	1065/0.0001	804/0.2791	1064/0.0002
4	804/0.2779	1065/0.0001	804/0.3297	1064/0.0001
5	804/0.2510	1066/0.0001	805/0.2758	1064/0.0001
6	804/0.2880	1066/0.0001	806/0.3995	1064/0.0001
7	805/0.3822	1065/0.0001	805/0.2682	1061/0.0002
8	806/0.2710	1065/0.0001	805/0.2682	1060/0.0002
9	804/0.3693	1064/0.0001	805/0.3298	1063/0.0001
10	804/0.4210	1065/0.0001	804/0.3616	1060/0.0001

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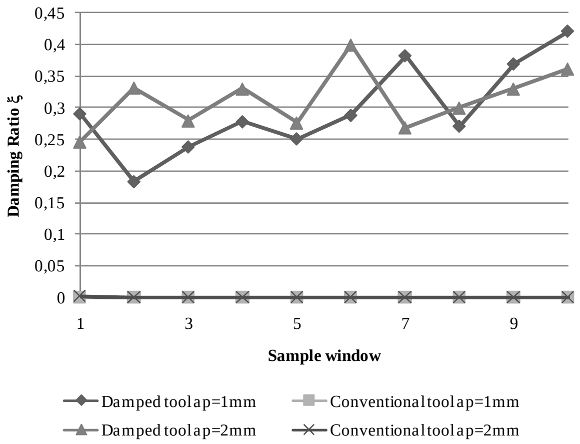

Fig. 10 Damping ratios identified with ARMA(3,2) model

4) Surface roughness

The surface finish produced by the conventional tool is of much lower quality if compared to the one produced by the damped tool, as also can be concluded from the model identification procedure. Fig. 11 shows the surface profile taken after machining at a_p = 1 mm; the conventional tool is not able to perform in stable conditions and therefore the surface profile is disturbed by the chatter marks. Fig. 12 shows the average surface roughness for the different depth of cut settings and tools. At a_p = 3 mm, due to severe chatter condition, the machining could not be accomplished with the conventional tool.

V. Conclusion

A passive vibration control approach has been used to design a boring bar for internal turning operations with enhanced damping capability. The design allowed a ten times enhancement of the damping ratio losing only a fraction of the static stiffness.

The design was validated by both EMA and machining tests. Model-based identification by parametric ARMA models has been used to investigate different cutting conditions effect on the frequency and total damping ratio of the machining system (combined elastic structure and machining process). The design allowed a ten times damping ratio enhancement compared to the conventional tool with a relatively low loss of static stiffness. The damped tool was able to perform stably at higher removal rates with a better surface finish than the conventional tool. The surface roughness produced by the damped tool was always about the half of the one produced by the conventional tool. Model-based identification offers a robust approach to estimate dynamic behavior during real cutting operations. The use of model-based identification for evaluation of dynamic stability gave the possibility to quantify the dynamic stability of a machining system in terms of damping capability.

The novel design using VE composite materials for boring bars has resulted in efficient tools that can be used to perform at high material removal rates in stable conditions over a relatively wide range of cutting conditions.

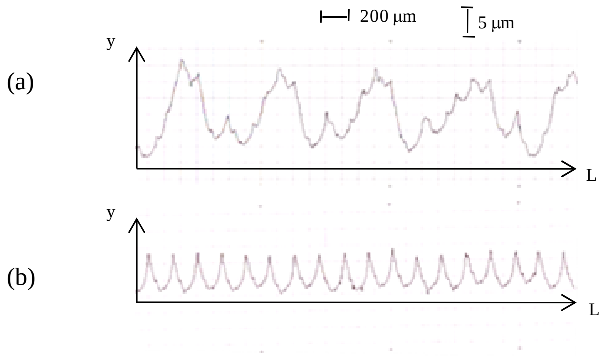

Fig. 11 Surface roughness scan. (a) Conventional tool, (b) damped tool; the scale is the same for both scans. The scans were performed after machining at v_c = 120 m/min, f = 0.15 mm/rev and a_p = 1 mm

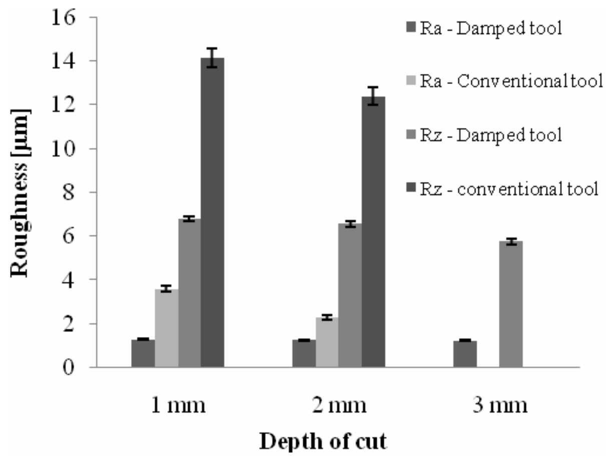

Fig. 12 Average surface roughness and standard deviation, R_a and R_z

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