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*damping, parting-off, model-based identification,
stability, operational damping ratio,
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DESIGN AND DYNAMIC CHARACTERIZATION OF COMPOSITE MATERIAL DAMPERS FOR PARTING-OFF TOOLS

This paper introduces a novel design for parting-off tools and the method to characterize their performance. 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 tool has been characterized by the dynamic characteristics criterion, i.e. frequency and damping ratio, of the machining system, as well as the roughness of the machined surface criterion. This paper demonstrates a new model-based method for characterizing the machining system dynamic properties, applied, in this study, to parting-off operations. The presented mathematical model of the machining system is based on the data recorded by a microphone during operational conditions. In this way, a step beyond the classical method of analyzing the dynamics of a machining system, which separately identifies the structural and process parameters, is taken. The analyses together with the experimental results proved that the parting-off tool was able to machine over a wide range of cutting parameters. It was found that the limiting factor for increasing cutting parameters is not the damping capability of the tool but the tool clamping system stiffness and the workholding system dynamic properties. This implies that, in order to further optimize the machining performance, it is vital to take in consideration not only the tool-clamp-turret system but the whole machining system.

1. 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.

Parting-off operations are quite sensitive towards cutting instability, the problem lies both in the geometry of the tool and the peculiar disposition of the cutting force. The parting-off tool has a slender protrusion, like a blade, where the cutting edge resides. The longer and thinner this protrusion the easier is to get unstable cutting conditions. The dimension of the tool is though constrained by the feature it has to create on the workpiece, this means that most of the time the tool geometry is not negotiable. The other major issue in parting-off or grooving operations is the disposition of the cutting forces.

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The feed direction is radial, therefore the longitudinal component of the force is null and the radial is significantly high, while in longitudinal turning the radial component is almost negligible if compared to the longitudinal one (feed force).

This means that the major force component is applied along the direction of the workpiece minimum stiffness. Thus vibration can be generated by the deflection of the workpiece as well, especially if the cutting operation is carried out far away from the chuck, see Fig. 1.

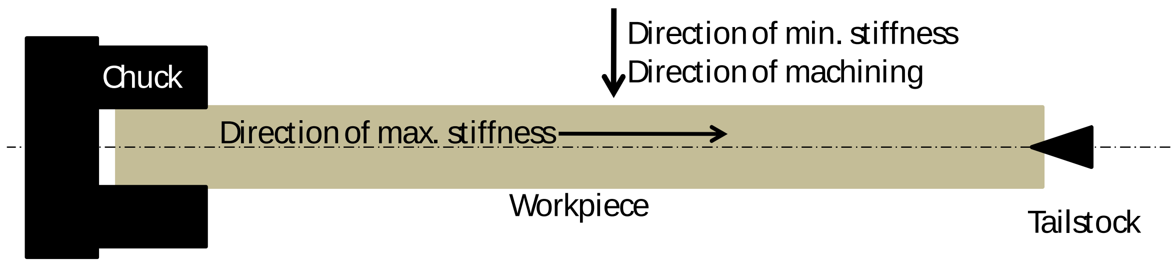

Fig. 1. Illustration of the directions of minimum and maximum stiffness and machining direction in a generic long and slender workpiece.

In the case of grooving camshafts, vibration could also be induced by the rotational unbalance of the workpiece. This paper will introduce a composite material damper for parting-off and grooving operations. Such approach has already been proved to be effective in applications such as boring bars for internal turning (DAGHINI, L. et al., 2008). The capability of the newly designed tool holder in machining camshafts is then quantified. To accomplish this, the acoustic signal emitted during machining and the produced workpiece roughness have been recorded, analysed and ultimately correlated to each other and to the cutting parameters. The machining system dynamic properties of the parting-off operation are analysed applying a novel model-based identification method. The mathematical model of the machining system presented is based on the previously mentioned data obtained during operational conditions. In this way, a step beyond the classical method of analyzing the dynamics of a machining system, which separately identifies the structural and process parameters, is taken.

2. TOOL HOLDER DESIGN

2.1. PASSIVE CONTROL OF VIBRATION IN TOOLING SYSTEMS

The principle of passive control is to enhance the damping capability of the component by converting the vibrating energy into other forms. A common approach is the use viscoelastic materials to dissipate the energy that causes vibration, by converting it into heat. This sort of materials has been used in other fields of application, such as automotive and aeronautics (RAO, M. D., 2003). Viscoelastic 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) (RAO, M. D., 2003). Examples of TVD are the dynamic vibration absorbers (DVA) with inertia mass. The basic principle of this technique is to add a mass residing on a spring and a viscous damper at the point of maximum displacement. The viscous damper is usually implemented using viscoelastic materials. This additional single degree of freedom (SDOF) system must have approximately the same natural frequency of the component in order to obtain large relative displacements, and if the viscous damper is properly designed it will dissipate the mechanical energy (RÜDINGER, F., 2006). A solution in this direction is proposed by Rivin et al. (RIVIN, E. I. and Kang, H., 1992) where the inertia weight is integrated in a boring bar, suspended on rubber rings. The absorber is tuned by changing the stiffness of the additional system. Another example of application of DVA principle is proposed by Lee et al. (LEE, E. C. et al., 2001), the DVA is, in this work, tuned by changing the inertia mass. This technique has been presented even for milling operations by Rashid (RASHID, A. and Nicolescu, C. M., 2008), who integrated the DVA into the workholding system. DVA technique is already successfully used in several successful commercial products. Rashid (RASHID, A., 2005) 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 success of the design to properly locate the viscoelastic layer in the structure since viscoelastic materials give the best results through shear deformation (BUTLER, N. D. and Ojadiji, S.O., 2007).

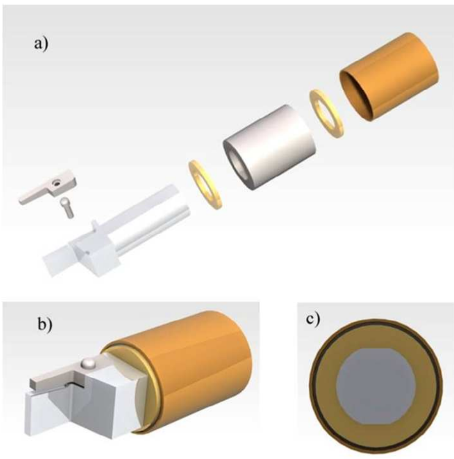

Fig. 2. CAD model of the parting-off tool. (a) Exploded view. (b) view of the assembled tool. (c) View of the back of the tool.

2.2. DESIGN CONCEPT

The concept followed in the design of the parting-off tool holder is by implementation of composite material layers in the tool structure. This principle had been previously proved to be effective in other tooling applications (DAGHINI, L. et al., 2009). The tool holder is clamped to the turret by hydrostatic adapters, in order to guarantee homogeneous clamping force and high static stiffness (DAGHINI, L. and Nicolescu, C. M., 2008). The damping interface is positioned as near as possible the cutting edge, creating an interface between the tool holder and the hydrostatic clamping system (Fig. 2).

3. CHARACTERIZATION METHOD

In order to understand the principle for design of efficient damping systems it is necessary to understand the dynamic behaviour of machining systems. Machining systems may be represented by a closed loop system comprising the machine tool elastic structure, i.e. the machine tool structure including tool, tool holder, workpiece etc., and the cutting process, i.e. turning, milling etc (Fig. 3) (ARCHENTI, A., 2008).

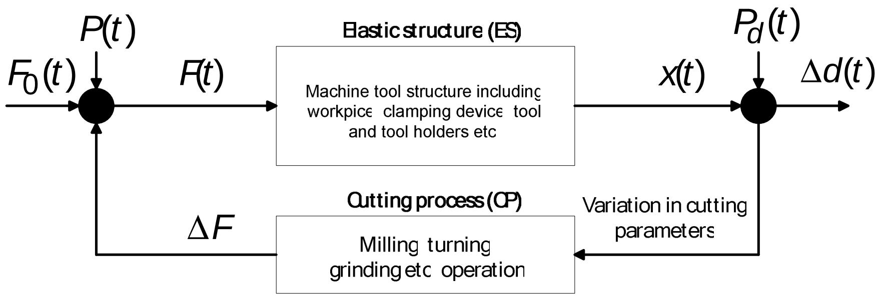

Fig. 3. A typical representation model of a machining system from a process-machine interaction point of view (ARCHENTI, A., 2008)

In Fig. 3 $F(t)$ 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. The interaction between the machine tool's elastic structure and the cutting process describes the behaviour of the machining system. This behaviour directly affects the process accuracy. Critical factors for optimization of design for damped structural components for machine tools are the modal parameters (frequency, mass 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 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.3

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 characterization of a new design for parting-off tools based on a novel model-based identification method, that takes into consideration the above mentioned interaction, will be introduced (ARCHENTI, A. and Nicolescu, C. M., 2008). 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.

3.1. MODEL-BASED IDENTIFICATION

The key concept of the identification procedure in this paper is to find, and discriminate, a feature of the measured random response, reflecting the operational conditions of the machining system (ARCHENTI, A., 2008). This is a second order qualitative identification since no provision for quantitative estimation is done on the measured signal (as opposed to non-parametric identification methods). The identification procedure used in this paper consists of four steps:

1. Data acquisitioning and data conditioning
2. Data reduction
3. Model identification
4. Feature extraction and analysis

Important factors that has to be considered in the first step is, choosing appropriate sensors and sampling techniques for data collection, removing trends and DC-levels from the sampled data. In some cases bandpass filtering techniques are used to reduce bandwidth of the studied frequency range.

One important feature when considering parametric models is that they can, with relatively short data lengths and few model parameters, qualitatively describe the dynamics of a machining system (ARCHENTI, A., 2008). In the second step this feature makes data reduction possible leading to fast identification of the system behaviour.

In the third step model structure and model order is selected.

In the fourth step the physical parameters, describing the interaction between the machine tool elastic structure and the cutting process, are statistically computed from the model parameters and used as the discrimination features.

Physical parameter identification

The parametric models, used in this study, are based on stochastic processes. A special class within the family of stochastic processes are defined by autoregressive moving average or ARMA models (BROCKWELL, P.J., Davis, R.A., 1987), (PANDIT, S.M., Wu, S.M., 1983). In the model-based identification procedure, estimation of the physical parameters can be used for the control of dynamic stability. By physical parameters, it is denoted, operational frequencies, ω_{op}, and operational damping ratios, ξ_{op}, which is the result of the structural vibration modes and process vibration modes during machining (ARCHENTI, A. and Nicolescu, C. M., 2009). 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 certain confidence intervals. The model for an ARMA process can be expressed as

$$Y(z) = H(z)U(z) \tag{1}$$

where $Y(z)$, $U(z)$ and $H(z)$ are the z-transforms of the output sequence, input sequence and the system impulse response (transfer function), respectively, and

$$H(z) = \frac{c_0 + c_1 z^{-1} + c_2 z^{-2} + \ldots + c_q z^{-q}}{1 - a_0 - a_1 z^{-1} - \ldots - a_p z^{-p}} \tag{2}$$

The motion of an n degree-of-freedom system excited by a random excitation $f(t) \in \Re^{n\times n}$ can be represented by a system of second-order differential equation

$$M\ddot{y}(t) + C\dot{y}(t) + Ky(t) = f(t) \tag{3}$$

$[y_1(t), y_2(t), \ldots, y_j(t), \ldots, y_n(t)] \in \Re^{n\times n}$ is the vector of n displacements of the system. $y_j(t)$ is the displacement of the mass j. M, C and $K \in \Re^{n\times n}$ are mass-, damping- and stiffness matrices. A dot over a time function denotes the derivative with respect to time.

The problem is to calculate the n operational frequencies, $(\omega_{op})_j$ and the n operational damping ratios, $(\xi_{op})_j$, j=1...n. Let $y_j(k\Delta T)$, k = 0, 1,2 … be the discrete samples of the displacement of the j-mass. ΔT is the sampling interval. Then the observations $y(k\Delta T)$ can be represented by an ARMA model

$$\sum_{i=0}^{p} a_i y(t-i) = \sum_{i=0}^{q} b_i x(t-i), a_0 = 1 \tag{4}$$

The AR characteristic equation of (4) can be written

$$\sum_{i=0}^{p} a_i y(t-i) = \prod_{j=1}^{n} (\mu - \mu_j) \cdot (\mu - \mu_j^*) \tag{5}$$

where

$$\begin{aligned} \mu_j &= exp\left[-(\omega_{op})_j (\xi_{op})_j \Delta T + i(\omega_{op})_j \sqrt{1 - (\xi_{op})_j^2} \Delta T \right] \\ \mu_j^* &= exp\left[-(\omega_{op})_j (\xi_{op})_j \Delta T - i(\omega_{op})_j \sqrt{1 - (\xi_{op})_j^2} \Delta T \right] \end{aligned} \tag{6}$$

μ^*_j is the complex conjugate of μ_j and $i = \sqrt{-1}$ .By deriving the physical parameters from the estimated AR (autoregressive) parameters, the operational frequencies $(\omega_{op})_j$ and operational damping ratios $(\xi_{op})_j$ can be determined (ARCHENTI, A. and Nicolescu, C. M., 2009)

$$\left(\xi_{op}\right)_j = \frac{ln\left(\mu\mu_j^*\right)}{\sqrt{\left\{\left[ln\left(\mu\mu_j^*\right)\right]^2 - 4\left[tan^{-1}\left(\frac{\mu-\mu_j^*}{\mu+\mu_j^*}\right)\right]^2\right\}}} \tag{7}$$

$$\left(\omega_{op}\right)_j = -\frac{1}{2\Delta T}\sqrt{\left\{\left[ln\left(\mu\mu_j^*\right)\right]^2 - 4\left[tan^{-1}\left(\frac{\mu-\mu_j^*}{\mu+\mu_j^*}\right)\right]^2\right\}} \tag{8}$$

Theoretically, dynamic stability can be defined in terms of positive damping. A system is dynamically stable if the damping is positive and unstable when damping approaches zero. In machining, as we are interested in avoiding instabilities like chatter, when damping start to decrease towards zero it is a proof that the system approaches the stability threshold. Therefore, monitoring damping in an on-line identification scheme can give good indication about the dynamical state of the system.

3.2. CHARACTERIZATION OF THE NOVEL DESIGN

The characterization of the novel design has been carried out in two steps: at first EMA has been employed to extract modal parameters for the tool and clamping. Then the sound recorded from machining tests has been processed 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 (DELIO, T. et al., 1992) and also have the practical benefit of not interfering in the working zone. The surface roughness has been measured after every test using Wyko® replicas. The workpiece used in the experiments was a forged steel camshaft. Finite element analysis (FEA) has been employed to estimate natural frequencies and mode shapes of the workpiece. The cutting parameters used in the machining tests were combined and alternated according to a matrix generated by the design of experiment procedure (DOE) with screening objective. These were chosen starting from the parameters suggested for such operation (v_c = 100 m/min and f = 0.1 mm/rev) and, to explore the tool capability, it was chosen to run tests with parameters up to as high as 5 times the ones conventionally used. The factors for this investigation were cutting speed (v_c), feed (f), and geometry of the cutting insert (Tab. 1 and Fig. 4), the responses were roughness (R_a), operational damping and operational frequency. The results have been analysed with DOE dedicated software MODDE®.

Table 1. DOE matrix of factors. MP stands for positive square edge, MB for full-radius round edge

Run #	v_c [m/min]	f [mm/rev]	Insert geometry
1	100	0,1	MP
2	133	0,5	MP
3	150	0,3	MB
4	150	0,3	MB
5	200	0,5	MB
6	200	0,1	MB
7	200	0,37	MP
8	100	0,37	MB
9	200	0,1	MP
10	100	0,1	MB
11	133	0,1	MB
12	150	0,3	MB
13	100	0,5	MP
14	100	0,23	MP
15	167	0,1	MP
16	200	0,5	MP
17	100	0,5	MB

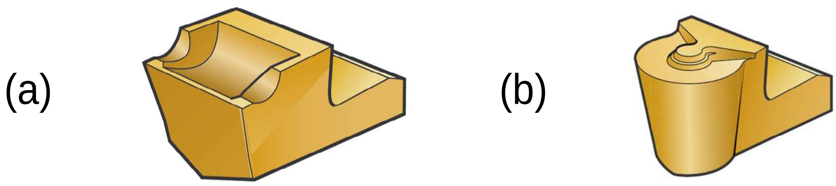

Fig. 4 Cutting insert geometry: (a) MP, positive square edge. (b) MB, full-radius edge

The machine tool employed for these tests was a SMT Swedturn® 300. The cutting insert grade was TNC150 (CVD TiN/TiCN/TiN coating) for both geometries suitable for machining P35 type material. Both inserts had a width of cut of 4 mm and positive rake angle.

4. RESULTS AND DISCUSSION

The parting-off tool has been characterized by analysing the tool itself by experimental modal analysis, and by running cutting tests as previously described.

4.1. MODAL ANALYSIS OF TOOL AND WORKPIECE

The EMA carried out on the tool clamped in the turret identified that the relevant natural frequencies were $f_{1T} \approx 1100$ Hz, $f_{2T} \approx 2200$ Hz and $f_{3T} \approx 3400$ Hz.

Fig. 5 illustrates their respective mode shapes and the positions of the accelerometers on the tool-clamp-turret system. The first mode (f_{1T}) is the natural frequency related to the tool-clamp-turret system as it can be observed in Fig. 5 (b). The second and the third modes (f_{2T} and f_{3T}) are natural frequencies related solely to the tool.

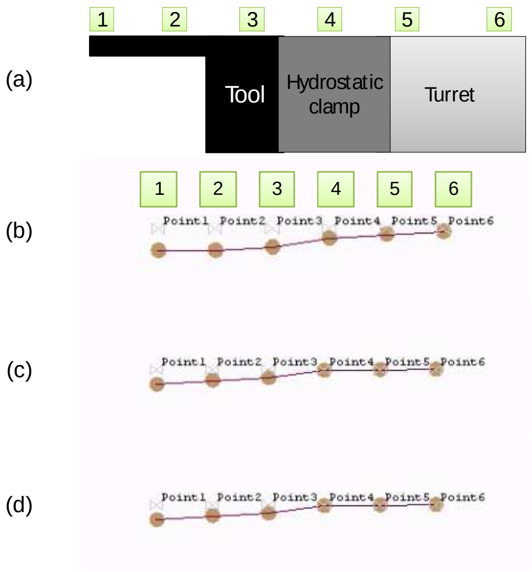

Fig. 5. Tool mode shapes: (a) Positions of the accelerometers. (b) Mode shape for f_{1T}≈1100Hz. (c) Mode shape for f_{2T}≈2200Hz. (d) Mode shape for f_{3T}≈3400Hz

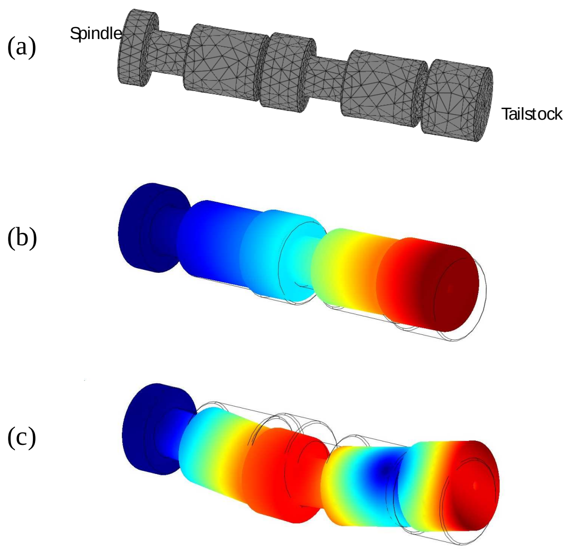

Fig. 6. FE-model of workpiece. (a) Mesh view.(b) Mode shape for $f_{1W} \approx 200$ Hz. (c) Mode shape for $f_{2W} \approx 900$Hz

A FEA of the workpiece has been carried out using COMSOL Multiphysics® 3.5 in order to identify the natural frequencies and the mode shapes of the workpiece clamped in the machine tool. The spindle chuck was modelled as a 3D constraint on the corresponding surface of the workpiece (left side in Fig. 6(a)). The tailstock was modelled as a one dimensional constraint, in the direction of the rotational axis of the workpiece. The first two natural frequencies resulted being $f_{1W} \approx 200$ Hz and $f_{2W} \approx 900$ Hz. The respective mode shapes are shown in Fig. 6 (b) and (c). From these analyses it can be observed that tool and workpiece are characterized by distinct dynamic properties and that the lowest natural frequency of the system belongs to the workpiece ($f_{1W} \approx 200$ Hz).

4.2. MACHINING TESTS

The acquired acoustic signal was studied together with its power spectral density (PSD). When analysing the signal, it could be seen that the identified frequencies (around 200 Hz) were very near the natural frequency of the workpiece and were therefore not correlated to the tool (see Fig. 7).

The next step in the analysis was to determine each cutting operations margin to instability. By applying ARMA models, of an order determined by the Akaike´s Informatics Criterion (AIC) (AKAIKE, H., 1981), and then fit them to the filtered signals, by help of Gauss-Newton algorithm (LJUNG, L., 2006), a parametric representation of the system can be achieved. From the estimated model parameters, dynamic characteristics of the machining system are then calculated. In this identification approach, the response signal is fitted into different models, evenly distributed, by batch technique, over the acquired data.

The lowest value for the operational damping ratio, for each cutting cycle, was calculated, see Table 2. The lowest operational damping ratio could be seen for the run 5 (ξ_{op}=0.0005) at $f_{op} \approx 205$ Hz. With 95 % probability, the operational damping ratio parameter is in the interval

$$0.000451 \leq \xi_{op} \leq 0.000549 \tag{9}$$

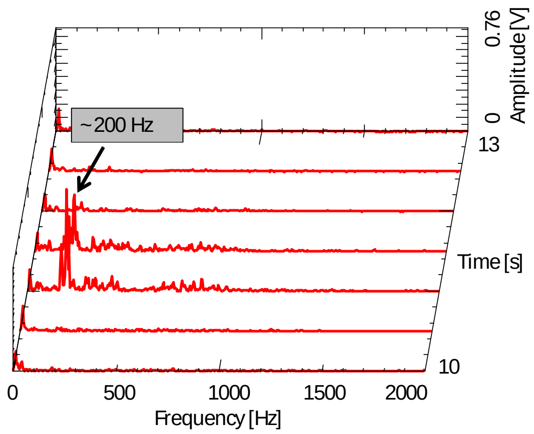

Fig. 7. Waterfall representation of PSD relative to test run 7

As it can be noticed in Tab. 2 the identified operational damping ratios were not excessively fluctuating reflecting good model stability.

The surface roughness appears to be strongly correlated to feed and geometry (see example in Fig. 8) but not to the cutting speed Fig. 9 shows that the confidence interval for the cutting speed coefficient crosses the zero line, indicating that the coefficient is not significant. The theoretical roughness has been calculated according to the well known equation:

$$R_a = 32.1\frac{f^2}{r_\varepsilon} \tag{10}$$

where R_a represents the roughness expressed in micrometers, f the feed and r_ε the nose radius of the cutting insert. In most of the runs the experimental results confirmed the calculated value, indicating good machining stability, but the limits of (10) makes it inapplicable to this case, especially when the feed is very large or very small compared to the nose radius. In this case study, the roughness resulted being dependent on the feed and not its squared value, as equation (10) would suggest, and on the tool geometry. These results suggest a dependence on feed and geometry according to:

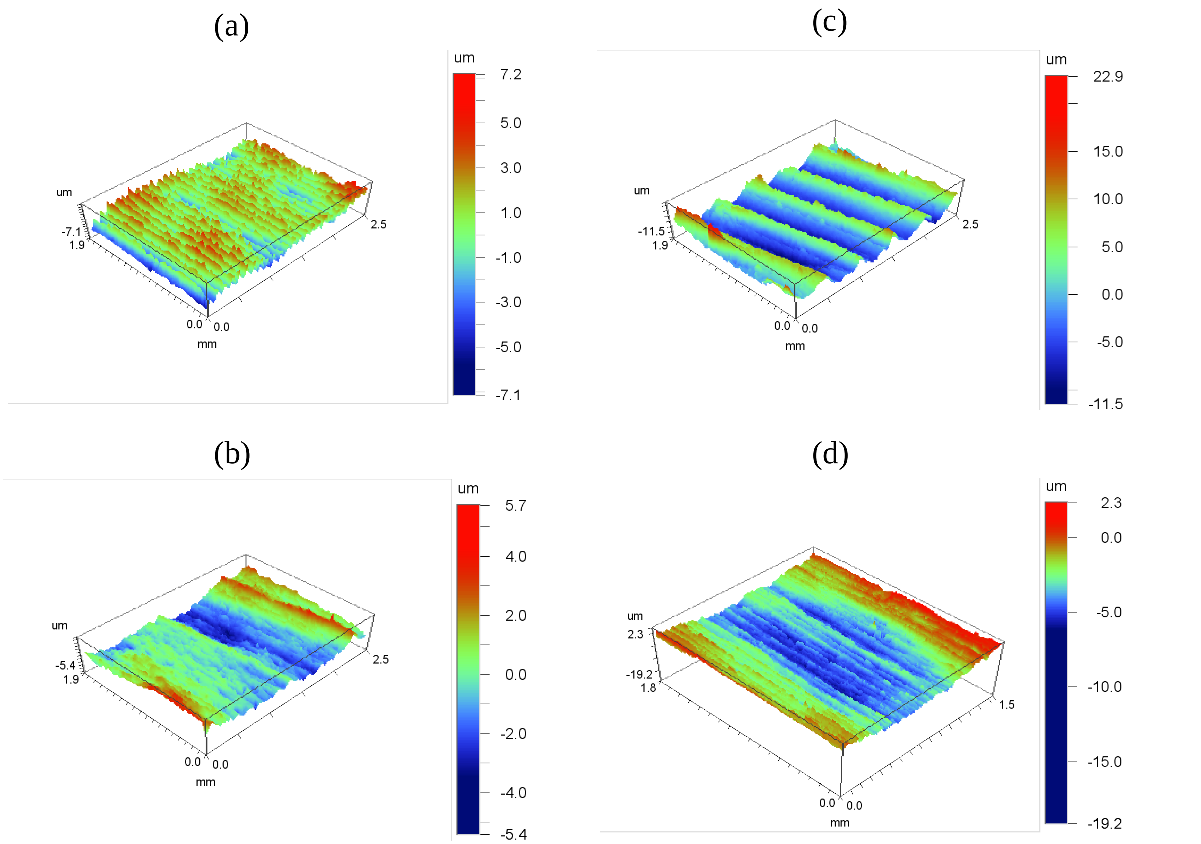

Fig. 8. Surface roughness comparison between (a) MP geometry and (b) MB geometry. Cutting speed and feed are unchanged. Surface roughness comparison between (c) f=0.5 mm/rev (d) f=0.1 mm/rev. Cutting speed and geometry (MB) are unchanged

$$R_a = -0.619668 + 17.6324 \cdot f + 0.722206 \cdot \begin{cases} -1 \text{ (if MB geometry)} \\ +1 \text{ (if MP geometry)} \end{cases} + \\ + 10.2114 \cdot f \cdot \begin{cases} -1 \text{ (if MB geometry)} \\ +1 \text{ (if MP geometry)} \end{cases} \tag{11}$$

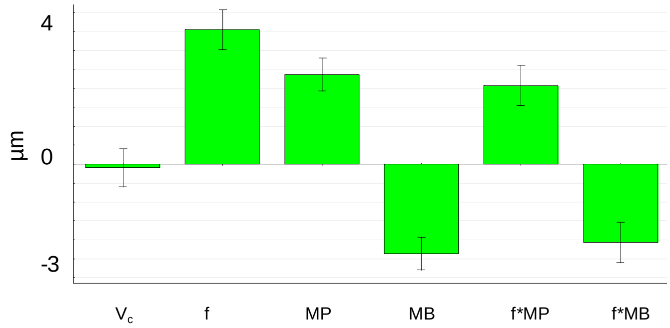

Fig. 9. Scaled and centered coefficients for R_a

Table 2. Summary of results

Run #	R_a [µm]	Calculated R_a [µm]	Op. Damping Ratio (ξ_{op})	Op. freq. (f_{op}) [Hz]
1	1.52	1.28	0.0018	205.0
2*	7.66	32.1	-----	-----
3	1.37	1.45	0.0011	205.5
4	1.75	1.45	0.0011	204.4
5	4.02	4.01	0.0005	207.2
6	1.22	0.16	0.0013	205.8
7	10.04	17.58	0.0011	204.2
8	3.34	2.14	0.0018	204.3
9	1.05	1.28	0.001	206.4
10	1.09	0.16	0.0007	207.0
11	1.11	0.16	0.001	208.2
12	1.75	1.41	0.0022	205.5
13*	21.75	32.1	-----	-----
14	5.12	6.79	0.0015	201.8
15	1.39	1.28	0.0009	205.7
16	11.81	32.1	0.0014	200.2
17	4.32	3.91	0.0014	205.0

Run 2 and 13 have been excluded from the investigation due to the poor quality of the measurements caused by unexpected tool holding failure. In this investigation no correlation between surface roughness and operational damping ratio could be found, this might be explained by a combination of two factors:

- The roughness has been measured along the feed direction and the tool's mode shapes are orthogonal to the feed direction (see Fig. 5).
- The dominant operational frequency identified, is not correlated to the tool-clamp-turret system, but to the workpiece-chuck-tailstock system.

5. CONCLUSIONS

The newly designed parting-off tool has undergone a complete dynamic characterization both by experimental modal analysis and through machining tests analysed with a model-based identification approach. The major implications of this characterization are:

- The tool operated in stable conditions over the whole range of cutting parameters. This has been demonstrated by the surface roughness (no signs of instability) and by the operational damping ratio that never reached zero.
- A linear dependency to feed and tool geometry could be found, this was developed into a mathematical model for predicting roughness value.
- The limiting factor for increasing cutting parameters is, in this study, not the damping capability of the tool but the tool clamping system stiffness and the workholding system dynamic properties. This implicates that, in order to further optimize the machining performance, it is vital to take in consideration not only the tool-clamp-turret system but the whole machining system.

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