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DEVELOPMENT OF A PASSIVE DAMPER FOR MILLING TOOLS BASED ON SMART MATERIALS

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Abstract. *Poor quality of machined surfaces is mainly caused by chatter vibration. This kind of vibration is usually nominated as self-excited because forces related to the machining process end up promoting vibrations on the natural frequencies of the milling system (or on some of their harmonics). So, it is typically different from natural or forced vibrations. In order to avoid this effect, this project intends to develop a dynamic model regarding a tool holder for milling based on shape memory alloys (SMA) to act as a passive vibration damper. To predict the effectiveness of this idea, a numerical simulation was developed in order to do a comparison between models regarding a tool holder filled with SMA and a hollow tool holder. Results show significant difference on the behavior of these models when it comes to high spindle rotations. So, it is expected that a SMA-based tool holder may be useful on vibration damping for higher cutting speeds.*

Keywords: *Machining, chatter vibration, shape memory alloys, modeling.*

1. INTRODUCTION

The currently state of art at process fabrication addresses many solutions that contribute to increase the industrial production. However the increased of the cutting speed can lead to some problems, such as the chatter. The chatter is a self excited vibration that can cause poorly quality surface, loud noise, damage at the cutting tool and others da Silva *et al.* (2015).

The prediction of chatter is one of the most effective ways to avoid it. The prediction can describe the ideal conditions of machining to the process, and it can provide a stability lobe diagram that allows one to better select machining parameters. To predict chatter, many authors propose different models to the machining process. Siddhpura and Paurobally (2012) have published a review about the chatter control at turning process and shows the influence of the regeneration of chip thickness.

In the case of processes which the tool holder has a high ratio between length and diameter, such as internal turning and drilling long holes, are more susceptible to suffer chatter due to its low stiffness. Aiming to reduce the vibration at the internal turning process Suyama *et al.* (2016) have studied three types of different tool holders, a standard tool holder, a hollow one and a hollow tool holder filled with little spheres. He conclude that the use of this spheres increases the damping capacity of the process. Thomas *et al.* (2019) have used the same idea of filling a tool holder for a internal turning, but instead of use small spheres, the study uses larger spheres with the diameter close to the internal diameter of tool holder which allowed a larger increase at the tool holder overhang.

Following the same line of thought, Galarza *et al.* (2020) have studied the use of a tool holder filled with bearing spheres at the ball nose end mill process, and obtained a notable decrease at the surface roughness of the work piece.

It has been shown in theses studies that the variation of some parameter, for example, the tool holder stiffness, may cause a decrease of chatter, which lead to a better quality surface of the work piece. One way to change the stiffness of the tool holder is changing it geometry and composition. Following the above cited authors, the use of a hollow tool holder filled with some material is an interesting way to try to avoid the chatter.

One of the advantages of use a numerical simulation is a controlled environment which allows to study a wide variation of parameters.

In order to study chatter analytically, Altintas (2011) has developed a two degrees of freedom (2 DOF) model of milling process, considering the dynamics of the cutting and feed forces acting at the multiple teeth of the tool holder. Based at the model proposed by Altintas, Budak (2006) has shown analytical methods to avoid the chatter at the milling process, without loss of productivity, based on a 2 DOF model of face milling.

Li and Liu (2008) have applied the model of milling with 2 DOF developed by Altintas at a numerical simulation of a face milling of a workpiece made by Al7075. The author have compared the simulation to an experimental test, considering the time and frequency responses. The simulation presented results similar the experimental ones, evidencing the efficiency of an analytical model allied of a numerical simulation.

The smart memory alloys (SMA), which are known for their unique properties, have been used by automotive industry, aerospace, biomedical and robotic (Mohd Jani *et al.*, 2014). One of its main properties is the pseudo elasticity and the capacity of return, what provides a high damping capacity, in way that it can be used as passive, semi-active or active damper. This property allows to use SMA with the strategy of filling the tool holder, as mentioned above.

To study the chatter of a face milling process using two different tool holders, this work uses an adaptation of the 2 DOF model proposed by Altintas to obtain the time and frequency response from which can be implemented a numerical simulation of the process. This work presents the simulation of two different tool holders: a hollow tool holder and a tool holder filled with a cylinder made of smart material alloys. Future tests will be executed in order to evaluate the results obtained by the simulations.

2. MATERIALS AND METHODS

2.1 System modeling

This work intends to model and simulate the dynamic behavior of an specific face milling process, using a hollow tool holder and one tool holder filled with smart memory alloys.

The process studied considers a tool holder with a single tooth, milling an H13 steel work piece.

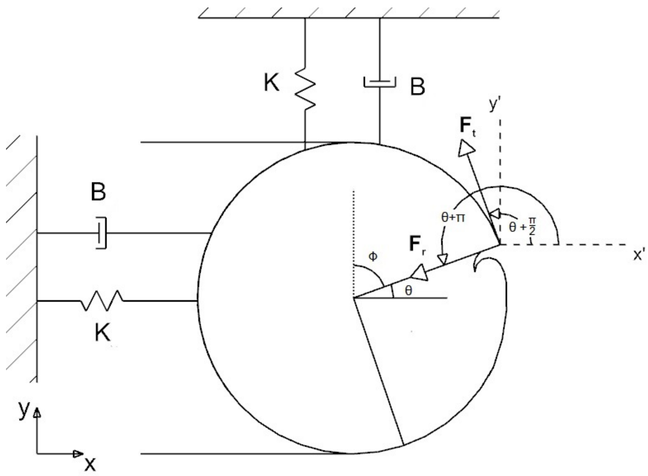

Figure 1. System modelling.

The face milling process can be modeled as a 2 DOF system, excited by two forces, the tangential (F_t) and radial (F_r) forces, as shown by the Fig. 1 The instantaneous chip angle of immersion ϕ can be directly related with the modulus of the spindle rotation (ω) as shown by the Eq. (1).

$$\phi(t) = \frac{\pi}{2} - \omega t \tag{1}$$

This way the system is ruled by the differential Eq. (3) and Eq. (4) rules at the time domain. The constants m_{eq}, B, and K are respectively the equivalent mass of the tool holder, the damping at the horizontal and vertical axis, the damping at the horizontal and vertical axis.

Modelling the tool holder as an cantilever, the equivalent mass m_{eq} can be calculated by the Eq. (2), where m is the real mass of the tool holder.

$$m_{eq} = \frac{33}{140} m \tag{2}$$

The instantaneous chip angle of immersion ϕ can be directly related with the cutting speed as shown by the Eq. (1).

$$m_{eq}\ddot{x}(t) + B\dot{x}(t) + Kx(t) + F_r(t)\cos(-\omega t + \pi) + F_t(t)\cos(-\omega t + \frac{\pi}{2}) = 0 \tag{3}$$

$$m_{eq}\ddot{y}(t) + B\dot{y}(t) + Ky(t) + F_r(t)\sin(-\omega t + \pi) + F_t(t)\sin(-\omega t + \frac{\pi}{2}) = 0 \tag{4}$$

2.2 Modeling of a hollow tool holder

In order to apply this model at a face milling process using a hollow tool holder, an impact test was made at a CoroMill 490-020A16-08L tool holder. The system test included an PCB 086C03 test hammer with a 2224,8 N sensor. The tool holder was instrumented with two PCB 333B30 with a sensitivity of 100mV/g and a frequency range between 0,5 - 3000 Hz. A cDAQ-9178 from National Instruments with 4 channels, 51,2kHz/channel and a 24 bits resolution was used for the data acquisition.

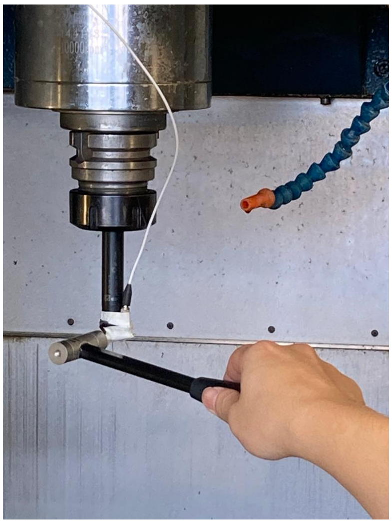

Figure 2. Hammer impact test.

With the impact testing data, a $H-1$ spectral estimator was used to estimate the frequency response function (FRF) relating the force applied in the tip of the cutting tool and the acceleration in the same place. With the FRF it is possible to obtain the natural bending frequency and the damping ratio of the corresponding vibration mode.

$$K = \omega_n^2 m_{eq} \tag{5}$$

Where ω_n and m_{eq} are respectively the natural frequency of the tool holder and it equivalent mass, considering it a cantilever beam.

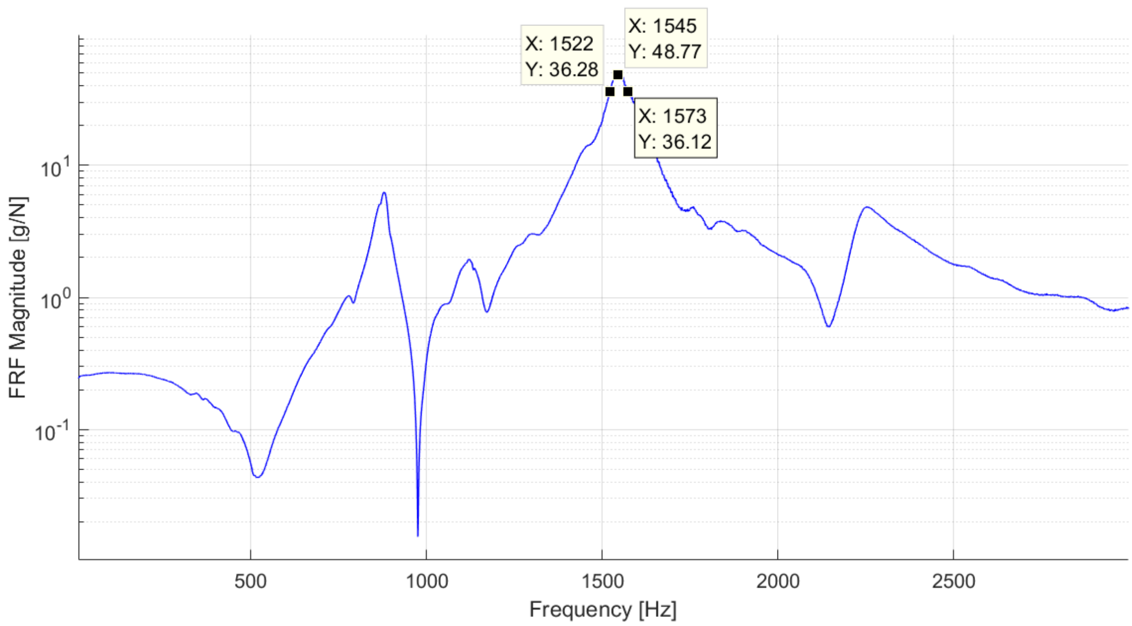

Figure 3. Hollow tool holder frequency function response.

The damping ratio (ζ) of the system was calculated using the half-power bandwidth method. The Eq. (6) and the

Fig. (3) shows the application of the method. The Tab. 1. shows all the values found.

$$\zeta = \frac{f_b - f_a}{2f_n} \tag{6}$$

The frequencies f_a and f_b are respectively the frequency which provides an amplitude equals to the natural frequency amplitude divide by square root of 2 (half-power frequencies).

The constant of damping B then can be calculated by:

$$B = 2m\omega_n\zeta \tag{7}$$

Table 1. Hammer impact test results.

Properties	Value
m_{eq}, kg	0.0269
ω_n, $rad.s^{-1}$	9707.24
K, $N.m^{-1}$	2.531e6
f_n, Hz	1544
f_a, Hz	1522
f_b, Hz	1573
ζ	0.0165
B, $N.s.m^{-1}$	17.7317

2.3 Modeling of a tool holder filled with SMA

The previously hollow tool holder is intended to be filled with a cylinder of SMA, as shown by the Fig. (4).

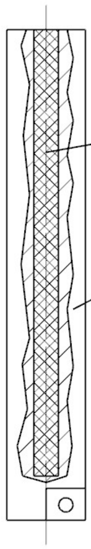

Figure 4. Tool holder filled with SMA.

The dynamic constants of this tool holder were defined by an analytical way. To determine the stiffness constant, it's necessary the elasticity modulus. Once the tool holder is composed by steel and it is filled with SMA, we can approximate it to a composite and apply the upper bound of the mixture rule to determine the elastic modulus E. The Eq. (8) shows the calculation of the elastic modulus at the transverse direction of the tool holder, the direction where the forces are applied.

$$E = (\frac{f}{E_{SMA}} + \frac{1-f}{E_{steel}})^{-1} \tag{8}$$

Where E_{steel}, E_{SMA} and f are respectively the steel and the SMA elasticity modulus and the volume fraction of SMA at the tool holder. This way the stiffness of the tool holder is given by the Eq. (9).

$$K = \frac{3EI}{L^3} \tag{9}$$

Where E, I and L are respectively the tool holder elasticity modulus, the cross section area moment of inertia and the tool holder length.

In order to calculate the damping, an estimate for the damping ratio of this tool holder were also made using the mixture rule, considering the values of SMA damping presented at Lagoudas *et al.* (2010) which executed a study over a Nitinol guide-BB-20 guidewire (55.8 wt % Ni 44.2 wt % Ti) at the pseudo-elastic regime. The damping ratio considered here is assumed constant, in order to avoid non-linearity at the damping behavior. Further experimental studies will analysis the non-linear damping behavior of the SMA acting as a passive damper at the milling process. All values calculated are shown at the Tab. 2.

Table 2. Tool holder filled with SMA properties.

Properties	**Value**
m_{eq}, kg	0.0293
E_{steel}, GPa	210
E_{SMA}, GPa	36
E, GPa	133
ζ_{steel}	0.0165
ζ_{SMA}	0.0670
ζ	0.017
$K, N.m^{-1}$	2.517e6
$B, N.s.m^{-1}$	17.7317

2.4 Cutting force model

The model adopted here is an adaptation of the model used by Altintas (2011). Altintas uses a linear model of force, where the machining force is directly depending of the uncut chip thickness ($h(\phi)$) as shows the Eq. (10), Eq. (11) and Eq. (12):

$$F_t = K_{tc} b h_c(\phi) + K_{te} b \tag{10}$$

$$F_r = K_{rc} b h_c(\phi) + K_{re} b \tag{11}$$

$$F_a = K_{ac} b h_c(\phi) + K_{ae} b \tag{12}$$

where K_{tc}, K_{rc} and K_{ac} are the cutting force coefficients, working respectively at the tangential, radial and axial direction. K_{te}, K_{re} and K_{ae} are the edge constants. The axial component can be considered null if the nose radius and the approach angle is assumed zero. The chip width is represented by b. Li and Liu (2008) have developed a numerical simulation using this model of force and they have obtained a representative model, as shown in the Sec. Introduction.

This work uses the Kienzle force model, where the cutting force can be expressed as a nonlinear function of the chip thickness, as shown by the Eq. (13):

$$F_c = K_{s1} b h_c^{1-z} \tag{13}$$

where K_{s1}, z are constants obtained experimentally. These values where taken from Machado (2011) and f_z is the feed rate. At the milling face process the instantaneous chip thickness can be wrote by the Eq. (14):

$$h_c = f_z \sin(\phi) \sin(\kappa_r) \tag{14}$$

where ϕ and κ_r are respectively the instantaneous immersion angle and the entering angle, which at this work is equal 90º. Considering the time dependence of the instantaneous immersion angle as shown by the Eq. (1) it is possible to wrote the cutting force as a function of time, as shown by the Eq. (15).

$$F_c = k_{s1} b (f_z \sin(\frac{\pi}{2} - \omega t))^{1-z} \tag{15}$$

At the proposed model, the cutting force F_c is equivalent to the tangential force F_t. The normal force can be expressed as a percentage of the tangential force, once it is equivalent to the feed force at its maximum.

2.5 Numerical simulation

The Figure (1) shows the model for the milling face process. However, it is evident that during half round of the process the cutting tool does not cut the work piece, turning the system into a second 2 DOF mass-spring-damper without excitation, only with initial conditions different from zero. In this way, the simulation has to alternate between the full model presented above and the model without cutting force.

The numerical simulation consists of approximate the time dt as Δt. The state-space representation allows the calculation of an variable as a function of it derivative. For the model mentioned above the acceleration of the Eq. (3) can be written as the Eq. (16).

$$\ddot{x}(t) = \frac{-(B\dot{x}(t) + Kx(t) + F_r(t)\cos(-\omega t + \pi) + F_t(t)\cos(-\omega t + \frac{\pi}{2}))}{m} \tag{16}$$

The software MatLAB R2018a was used to code the model and it was simulated applying the function ode45, which consist of an 4th order explicit Runge-Kutta method, to solve the differential equations.

The inputs for the simulation are the spindle rotation ω, the feed per tooth f_z and the depth of cut b and the outputs are the horizontal and vertical displacement of the tool holder at the time and frequency domain, and the forces at the time domain.

3. RESULTS AND DISCUSSION

The system model using both tool holders, the hollow and the filled one, were simulated. The tool holder displacement were simulated with an inertial referential located at the center of the tool holder, in way that the displacement represent directly the vibration of the process.

The cutting forces shows exactly the cutting region of the process, that occurs at time between $0+2\pi k < t < \frac{\pi}{\omega}+2\pi k$, with $k = 0, 1, 2...$ as shown by the Fig. (5)

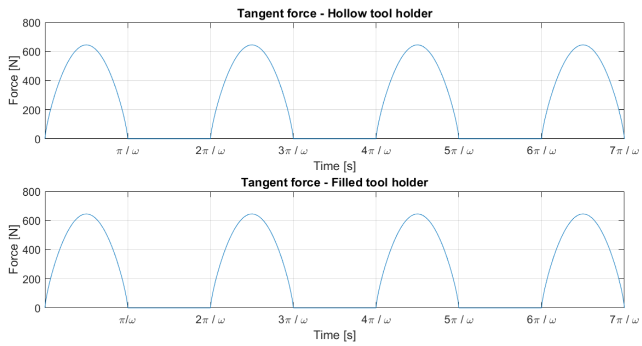

Figure 5. Tangent force time response results for ω = 15000 rpm.

As expected, the force is the same for both tool holder, once it does not depend directly from tool holder properties as depicted in Fig. 6.

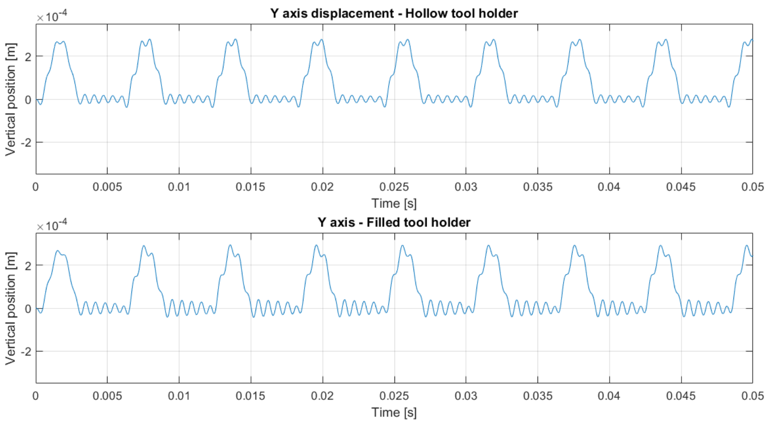

Figure 6. Time response results regarding Y displacement for ω = 10000 rpm.

To compare the chatter between both tool holders, orbital plots were used. The Figure (7) show the displacement for both tool holder at a low cutting speed, where the chatter is minimum. Note that the behavior of both tool holder is close at low speeds. This is expected since low speeds should lead to low vibration regardless the dynamic parameters.

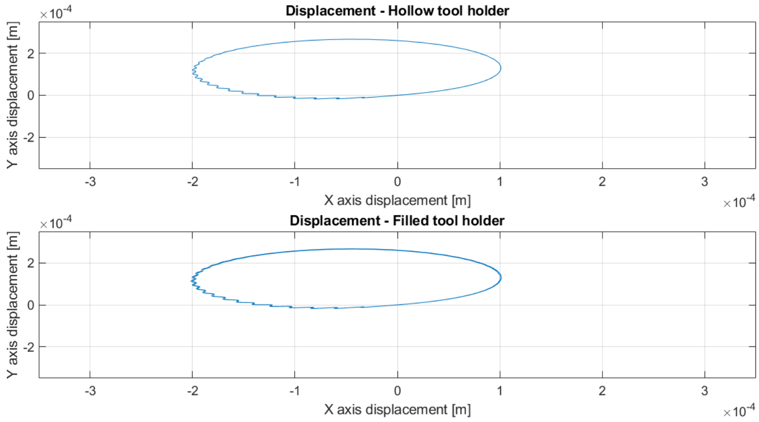

Figure 7. Displacement response results for ω = 1000 rpm.

When the spindle rotation is set up to 8000 rpm, the presence of chatter increase and it is possible to note less chatter at the hollow tool holder, what is unexpected, as show by the orbit plot at Fig. (8). This can be a critical configuration for the filled tool holder.

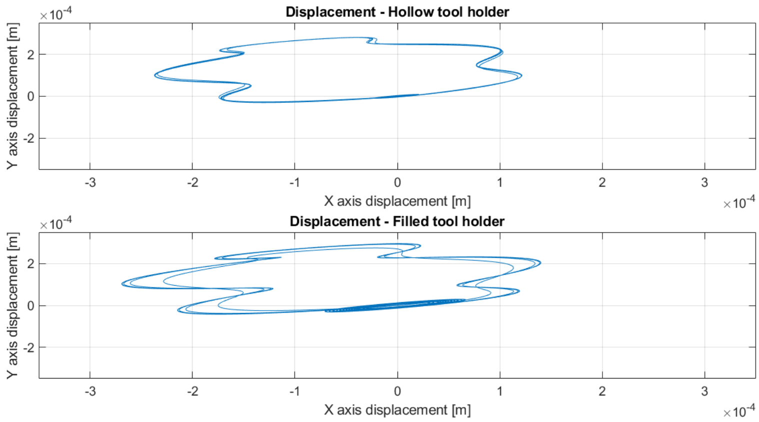

Figure 8. Displacement response results for $\omega = 8000$ rpm.

However, at 9000 rpm, the hollow holder chatter exceeds the chatter of the filled tool holder, as shown by the orbit plot at Fig. (9). This can be a critical configuration for the hollow tool holder.

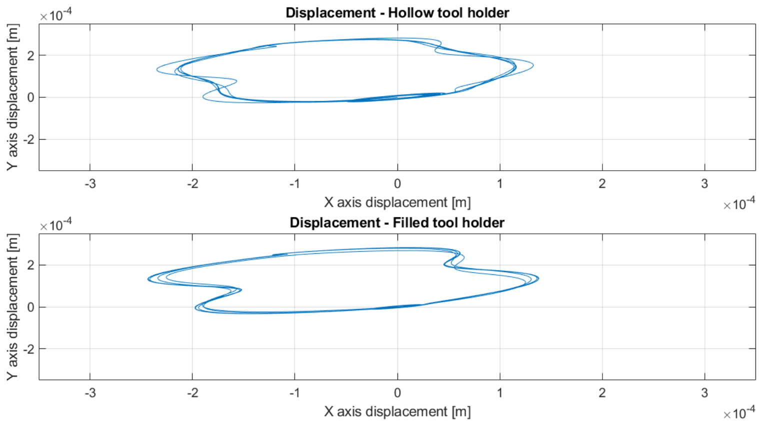

Figure 9. Displacement response results for $\omega = 9000$ rpm.

Once the different spindle speeds can benefit the hollow tool holder or the filled one, a set of simulation between 500 rpm and 10400 rpm with a pitch of 100 rpm with different cutting conditions. The difference between the hollow tool holder displacement and the filled one at the Y axis are shown at the Fig. (10). The regions where the blue curve assume positive values appoints the spindle value where is better to use the filled tool holder.

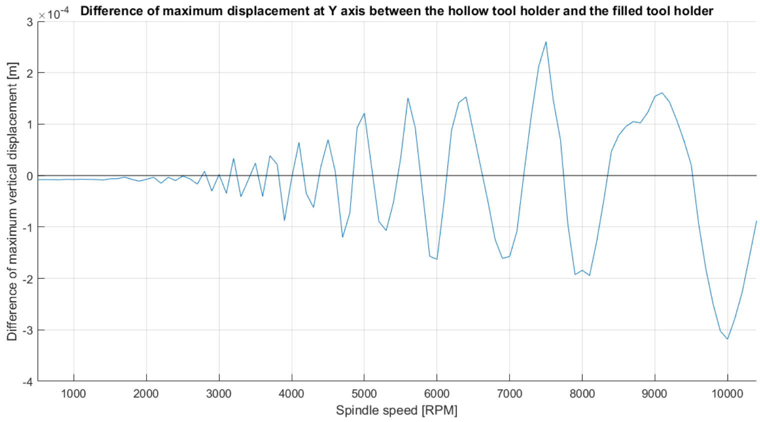

Figure 10. Difference between the maximum vertical displacement of both tool holder.

The simulations were performed using the Kienzle force model and they presented stable and critical configuration for both tool holders. The results demonstrate that these models can be used to extract information about both holders and compare them.

4. CONCLUSIONS

The models developed in this work provide a comparison between a hollow tool holder and a tool holder filled with SMA on face milling. Although their behavior is quite similar at low spindle speeds, they are significantly different for specific higher rotations, such as between 6000 RPM and 7000 RPM and between 8000 RPM and 9000 RPM, as shown at Fig. (10).

Since the models are based on Kienzle force model which is nonlinear, it is possible that they will be more representative than linear models found in the literature. Moreover, based on the results presented, it can be said that the use o SMA inside the tool holder may be useful on damping tool vibration specially at higher spindle rotations.

Further experimental tests may confirm these hypothesis.

5. ACKNOWLEDGEMENTS

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