Self-tuning semi-active tuned-mass damper for machine tool chatter suppression

G. Aguirre[1], M. Gorostiaga[1], T. Porchez[2], J. Muñoa[1]
[1] IK4-IDEKO
Arriaga 2, E-20870 Elgoibar, Spain
e-mail: **gaguirre@ideko.es**

[2] CEDRAT TECHNOLOGIES
15, Chemin de Malacher – Inovallée, 38246 Meylan Cedex, France

Abstract

Tuned mass dampers are simple and efficient devices for suppression of machine tool chatter, which is one of the principal effects limiting productivity in many machining processes. However, their effectiveness depends on a proper tuning of the damper dynamics to the dynamics of the machine. This involves the dynamic characterisation of the machining process, in order to identify the critical resonance frequency, and the possibility of matching the resonance frequency of the damper to frequency. The difficulty of meeting these two requirements has been limiting the use of tuned mass dampers in industrial applications.

An improved damper tuning device is presented here trying to overcome these limitations, and its feasibility is demonstrated with simulation and experimental results. On the one hand, the damper incorporates accelerometers and a control strategy that can detect in real time whether chatter is occurring and at which frequency in real time during the machining process. On the other hand, the damper has a variable stiffness spring, actuated by a motor, which allows automatic control of the stiffness, and thus the resonance frequency of the machine. Damping is generated in the damper by eddy currents, altogether providing a quite linear and predictable behaviour.

1 Introduction

The main dynamic problem limiting productivity in milling and other manufacturing processes is a self-excited vibration named chatter, which results from a waviness regeneration mechanism in the generation of chip thickness during the machining operation. In practice, chatter appears above a certain cutting depth, and produces high machine vibration levels that lead to problems such as high surface roughness and tool breakage. Much research effort is still dedicated to the prediction of chatter appearance and to broaden the range of stable cutting conditions. The latter will be the focus of this paper, by proposing a self-tuning semi-active tuned-mass damper that can detect when chatter is being generated and can tune itself to add damping to the critical resonance mode and avoid chatter, and thus allow more productive cutting conditions.

The current trend in machine tool design aims at stiffer machines with lower influence of friction, leading to faster and more precise machine tools [1–3]. However, this is at the expense of reducing the machine damping, which is mainly produced by friction, and thus increasing the risk of chatter appearance. Therefore, chatter, being a long-studied phenomenon, is still gaining relevance as the main limiting factor towards higher machining productivities, requiring improved methods for adding damping to machine tools without reducing the stiffness or increasing friction.

Tuned-mass dampers are a relatively simple and low cost solution to add damping to machine tools, by adding an inertial mass to the machine structure, attached to it with certain stiffness and damping [2–5]. A proper tuning of the resonance of the damper to the dominant structural mode of the machine increases the damping of this mode and enables higher productivity rates.

This principle has been demonstrated already for a long time, but its impact in industry is still small. The frequency of the dominant structural mode changes during the machining process, due to the dependency of the dynamics of the machine with the position of the axis, and between machining processes, due to change of tools or workpiece materials. Conventional tuned-mass dampers cannot maintain the tuning for all these conditions, thus losing their effectiveness. Therefore, there is a clear industrial need for solutions that can extend the functionalities of tuned mass dampers to add self-tuning capabilities, thus identifying in real time during the machining process the dominant resonant frequency and tuning the damper to follow it.

Semi-active dampers (i.e. dampers with variable stiffness and/or damping by means of low power action) [6–8] can offer a good trade-off between performance and design complexity and cost, compared to active solutions, but effective and robust implementation solutions suitable for industrial use are still missing.

A new semi-active tuned mass damper concept will be presented here, which combines three novelties:

- Automatic in-process detection of the main vibration frequency to be damped with accelerometers embedded in the damper.
- Automatic in-process tuning of the resonance frequency of the damper to the main vibration frequency of the machine by means of a variable stiffness spring controlled by a motor, ensuring repeatable and linear behaviour.
- Damping is produced by eddy current effect, generated by the vibration of conductor plates within a magnetic field, providing a predictable and linear damping behaviour.

Such a damper prototype has been designed and built, with performance, robustness, low cost and compactness as main goals, in order to achieve a system that can be used in real industrial applications. With external dimensions of 200x150x150 mm and an inertial mass of 7 Kg, the damper can detect chatter conditions and tune itself automatically to frequencies between 66 and 105 Hz, with a damping of 700 Ns/m. These values have been selected to fit the requirements of a SORALUCE milling machine at IK4-IDEKO's facilities, which has been used for experimental validation. Since online chatter detection and tuning function is still in implementation phase, in the machining tests presented here the damper has been tuned offline based on a previous experimental dynamic analysis of the machine.

In the following, in Section 2 the general design requirements for such a damper are presented. In Section 3, the design of the main functions of the damper proposed here is presented. In Section 4, experimental results demonstrating the improvement in the machining conditions are shown. In Section 5, the online chatter detection and damper tuning function which is being developed is described. Finally, conclusions are drawn in Section 6.

2 Tuned mass damper design requirements

The effectiveness of tuned-mass dampers in chatter suppression relies on a proper choice of the location of the damper, the moving mass and a proper tuning of the stiffness and damping with which the mass is connected to the machine. The choice of the optimal values for these parameters relies on a previous identification of the critical vibration mode leading to chatter. The theory behind this is explained next. The implications for the design of the dampers are discussed afterwards.

2.1 Tuning theory for machine tool inertial dampers

The working principle of inertial dampers is the same when used for structural damping in civil engineering or when applied to chatter suppression. However, the optimal tuning frequency is calculated in a different way due to the specifics of chatter generation.

For the sake of simplicity, we will consider that the machine resonance mode leading to chatter (the one we need to damp) does not have other relevant modes at nearby frequencies with which it could interact. In this case, we can model it as a single degree of freedom mass-spring system, represented by modal mass M and stiffness K in Figure 1, which is affected by an external force F_0 that represents the machining forces. If we want to reduce the vibration amplitude around resonance, we can add a second vibrating system attached to M, represented by m_1, k_1 and c_1. These elements represent the tuned-mass damper.

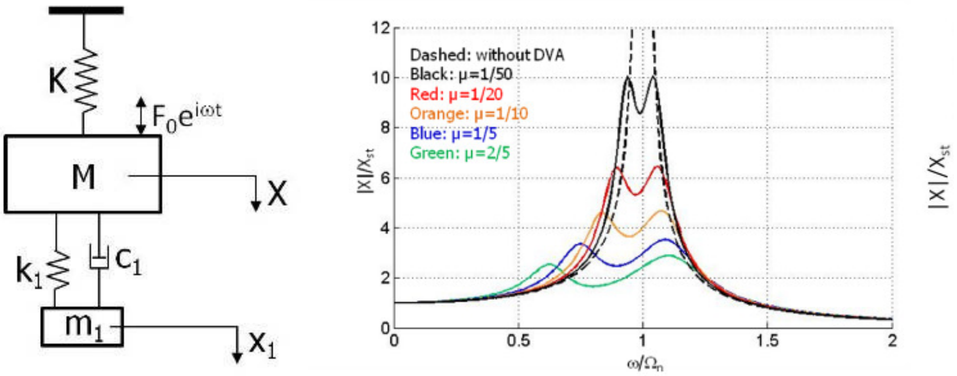

Figure 1: a) Simplified model of machine and damper b) Influence of mass ratio μ on FRF

The key to the efficiency of the damper is the right choice of the parameters m_1, k_1 and c_1. The goal is to tune the resonance of the damper to the machine resonance, so that the force generated by the vibration of m_1 counteracts the external force F_0, reducing the vibration of the machine M. The main dynamic parameters defining the system are represented in Table 1.

TMD				Structure	
Mass ratio	Damping ratio	Frequency ratio	Natural frequency	Frequency ratio	Natural frequency
$\mu_1 = \frac{m_1}{M}$	$\xi_1 = \frac{c_1}{2\sqrt{k_1 m_1}}$	$f_1 = \frac{\omega_1}{\Omega}$	$\omega_1 = \sqrt{k_1/m_1}$	$\beta = \frac{\omega}{\Omega}$	$\Omega = \sqrt{K/M}$

Table 1: Non-dimensional dynamic parameters

The first step in the tuning of the damper is the selection of the mass m_1 of the damper. In theory, m_1 should be as large as possible, but this is limited by practical issues limiting available space, and thus mass. In practice, m_1 is typically selected as 5 to 10% of the modal mass M. This mass ratio is called μ and its effect can be observed in Figure 1. Once the mass ratio is defined the optimal tuning of the damper resonance frequency needs to be calculated. Here two different strategies need to be mentioned, depending on the application, and their optimal tuning values are given in Table 2.

- The general theory of tuning inertial dampers was established by Den Hartog [4], and is based on choosing the damper frequency so that the absolute value of the FRF shows two peaks of the same amplitude, as in Figure 1. This is of general use for damping structural vibrations.
- The particular properties of chatter generation led Sims [2] to the derivation of a different tuning strategy that provides a broader range of stable working conditions. In this case, the optimal tuning frequency is the one that minimizes the real part of the FRF around the resonance frequency.

	Frequency ratio	Damping ratio
Den Hartog ([4])	$f=\dfrac{1}{1+\mu}$	$\xi=\sqrt{\dfrac{3\mu}{8(1+\mu)^3}}$
Sims [2] (α is positive)	$f=\sqrt{\dfrac{\mu+2+\sqrt{2\mu+\mu^2}}{2(1+\mu)^2}}$	$\xi=\sqrt{\dfrac{3\mu}{8(1+\mu)}}$

Table 2: Optimal tuning analytical expressions

2.2 Damper requirements

The importance of a proper selection of the dynamic characteristics of the damper has been established in the previous section. In practical terms, this brings the following requirements:

- Identification of vibration mode leading to chatter: An experimental modal analysis of the machining is required, which combined with the simulation of the cutting process, leads to stability diagrams that can help in detecting the critical mode.
- Choice of location and mass of the damper on the machine. Looking at the mode shape of the damper, the damper should be located at a point, within feasible ones, where the vibration amplitude is maximum, so that the modal mass at this point is minimum, requiring thus less inertial mass on the damper for the same efficiency. The inertial mass is typically set to 5-10% of the equivalent modal mass.
- Identification of the optimal tuning frequency (see Table 2): the optimal tuning frequency is the frequency of the mode that leads to chatter. This frequency can be detected at the modal analysis mentioned in point one, but with the drawback that this frequency can change during the cutting process, for different axis positions within the workspace, or due to the variation of the mass of the workpiece due to material removal. Active inertial dampers have a relevant advantage here, since their acting force can be used for automatic frequency identification during operation. With passive or semi-active dampers, no force can be generated, and thus the only possibility is to measure the chatter frequency when it is being generated, since this frequency is typically close to the resonance frequency.
- Once the optimal tuning frequency has been detected, the next step is to tune the damper to this frequency. Semi-active dampers can be tuned in frequency, typically by varying their stiffness, having a constant mass. When the frequency is detected from an experimental modal analysis, it is sufficient to have the possibility of tuning the damper manually right after it. If automatic tuning is possible, it can be of interest to detect the chatter frequency during the machining process in order to continuously tune the damper.
- Finally, the damping of the inertial damper needs to be adjusted to the optimal value (see Table 2). The effectiveness in chatter reduction is less sensitive to the proper tuning of the damper, and therefore, even in semi-active solutions, the damping is usually fixed at a value similar to the optimum, and not modified afterwards.

The design of a semi-active damper with self-tuning capabilities is proposed next.

3 Design of a new semi-active tuned-mass damper

A new semi-active tuned mass damper for machine tool chatter suppression has been developed. The main goal has been to obtain a system that can tune itself automatically to changing working conditions. Therefore, the main challenges have been the development a stiffness tuning system with wide working range and high repeatability and linearity, and the integration in the damper of an automatic tuning control strategy. The design of the damper is discussed in detail in the following.

3.1 Stiffness tuning

The optimal stiffness for the inertial damper is defined by the frequency of the vibration mode leading to chatter, and the inertial mass of the damper. Since the optimal frequency is variable within a machining process, and in order to cover a range of machines and processes with the same damper design, it is of interest to have the possibility of changing the stiffness of the damper.

A rotary element with variable stiffness in function of the angular direction has been selected, with suitability for automatic control and linearity, predictability and repeatability as main advantages. A similar concept for variable stiffness was presented in [5], but it has been here improved to enable an easier design and automation. The general design and working principle can be seen in Figure 2a. The thickness *a* is used to tune the lower stiffness of the spring, and the thickness *b* for the higher stiffness of the spring.

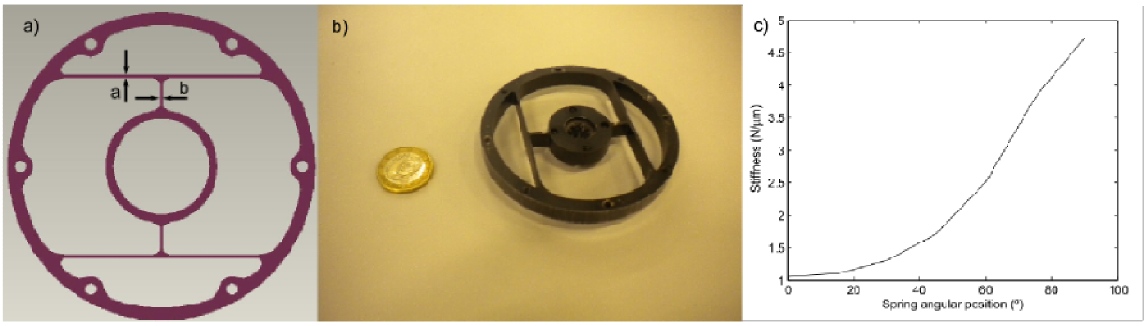

Figure 2: Rotary spring with variable stiffness (a) design (b) prototype (c) simulated stiffness

Finite Element (FE) calculations have been performed to design the spring according to the requirements defined in Section 4. The results of the obtained stiffness in function of the angular position are presented in Figure 2c

This rotary spring connects the moving mass to the machine. Automatic stiffness control is implemented by controlling the rotary position of the spring with a motor. The integration in the damper is shown in Section 3.3

3.2 Eddy current damping

Damping produced by energy losses related to eddy currents is proposed here as an optimal method for generating the damping needed in inertial dampers, since the damping they produce is very close to ideal viscous behaviour and it is generated without contact between parts, avoiding, for example, the non-linear effects produced by friction.

According to Faraday's law, when a time varying magnetic field is applied to a conductor, eddy currents are induced in it. Similarly, if a conductor moves through static but non-uniform magnetic field, eddy currents are induced on it. These eddy currents generate a magnetic field opposed to the variation of the magnetic field seen by the conductor, producing a force that acts against the variation of the magnetic field.

In the second case mentioned above, where a conductor moves through a static and non-uniform field, this force acts as a braking force, trying to slow down the relative motion of the conductor within the field.

Since this force is proportional to the velocity, it can be seen as a damping force. The energy dissipated is converted into heat as Joule's effect losses of the eddy currents on the conductor [9].

From the Lorentz's Law and Ohm's law,

$$\vec{E} = \vec{E}_c + \vec{v} \times \vec{B} \tag{1}$$

where v is the relative velocity between the conductor and the static field, B is the magnetic flux density and E_c is the electric field due to the induced Coulomb charges. Considering the electrical conductivity of the conductor σ, the density of the induced currents J in the conductor is:

$$\vec{J} = \sigma\left(\vec{E}_c + \vec{v} \times \vec{B}\right) \tag{2}$$

and finally the braking force can be calculated as:

$$\vec{F} = \int \vec{J} \times \vec{B} dV \tag{3}$$

3.2.1 Magnet and conductor configuration

Damping generation based on eddy currents has been proposed at research level for damping cantilever beams [10], [11]. In [12] eddy currents are used to damp a high precision positioning system. In [13] an improved magnet disposition for a damper was presented, in which the poles were arranged in an alternating pattern that increased significantly the damping force and coefficient.

Based on [13], and in order to optimize the damping force, but also the total volume, the configuration shown in Figure 3 was selected. This module design allows fixing the magnets easily, and then handling and mounting each module in the damper independently. The magnets used are made of Neodymium (magnetic remanence of B_r=*1.2T*) and their size is of 40x20x10mm. An air gap of 0.4 mm between the magnets and the copper plate has been defined.

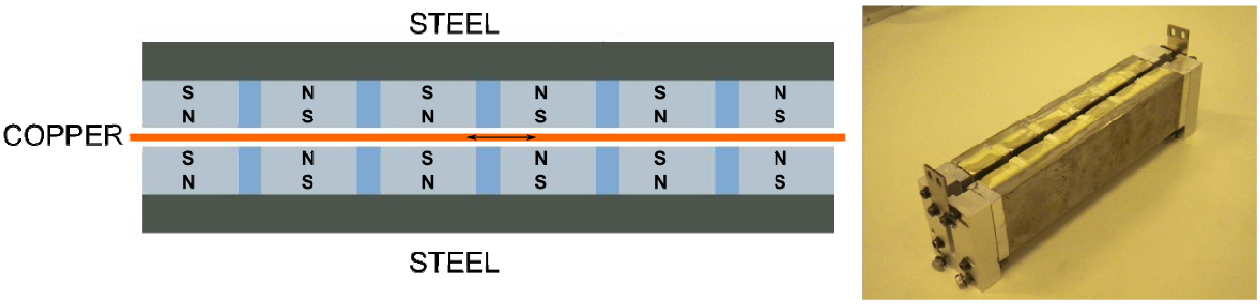

Figure 3: Magnet configuration and orientation, design and prototype

In Figure 4, the flux stream lines and the flux density values along the copper conductor calculated with FLUX Finite Element software are represented. The value of the magnetic flux density through the conductor is plotted, where the maximum value of 0.9T can be seen. It can be noticed that the stream lines close through the circuit avoiding the air, thus reducing the system's reluctance and increasing the magnetic flux density through the conductor.

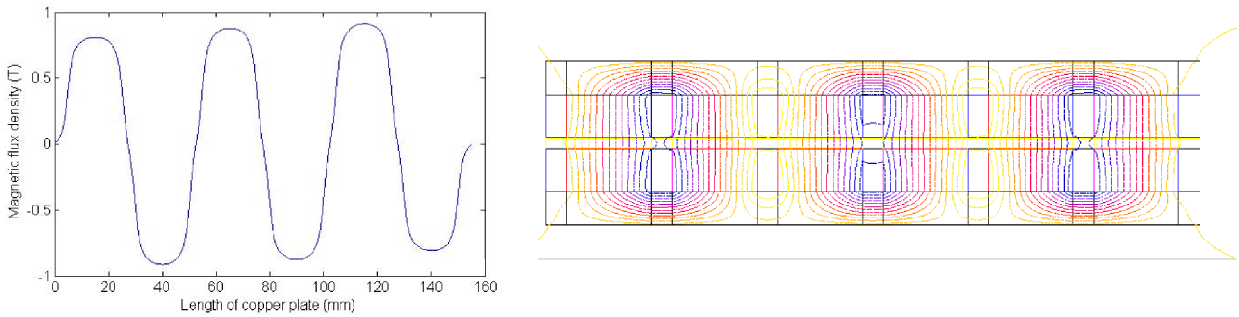

Figure 4: Distribution of the magnetic field

3.2.2 Conductor and air gap thickness optimization

After defining the magnet configuration, the next step is to define the conductor plate's thickness. A design trade-off needs to be taken here: thin plates are of interest because they allow shorter distance between magnets, and thus stronger magnetic field, but the generated currents, and thus the force are then limited by the resistance of the conductor plate, which decreases with the thickness, up to the limit where skin effect is relevant, since then the currents tend to concentrate on the surface of the conductor.

Copper is chosen as material for the conductor plate, due to its low resistivity, and its optimal thickness is calculated with FE simulation. The skin effect depth at 100Hz is 6.5 mm. In Figure 5, the dependence of the damping coefficient generated by a magnet-conductor block is plotted for different copper thicknesses, assuming an air gap of 0.4 mm at both sides of the copper. In this case, a maximum damping coefficient is observed for 2mm plate thickness.

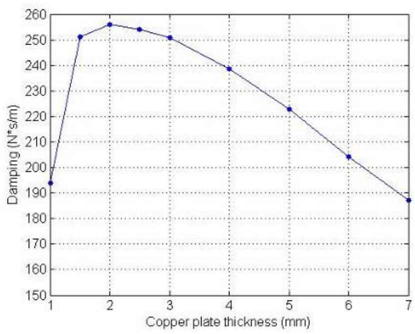

Figure 5: Influence of the thickness of the copper plate on the damping

3.3 Prototype

A first prototype of semi-active tuned-mass damper has been built following the principles described above. With external dimensions of 200x150x150 mm and an inertial mass of 7 Kg, the damper can detect chatter conditions and tune itself automatically to frequencies between 66 and 105 Hz, with a damping of 700 Ns/m. These values have been selected to fit the requirements of a SORALUCE milling machine at IK4-IDEKO's facilities

The prototype and its assembly process can be seen in Figure 6. The arrow indicates the direction of vibration of its main mode, which must be aligned with the machine mode that needs to be damped.

The integration of the main functions in the prototype is presented next. In a first step, the location of the rotary spring is presented in Figure 6a. The inner part of the spring is connected to the support (and the machine) through a rigid column, and the inertial mass is mounted on the external part of the spring. The stepper motor is mounted on the structure and it can rotate the spring.

The four magnet blocks are part of the moving mass, and are mounted symmetrically in the direction of motion, as shown in Figure 6b. At this point the moving mass is only connected to the rotary spring. In order to ensure that the first vibration mode of the damper is the expected one, eight guiding elements are added between the moving mass and the fixed structure

Finally, in Figure 6c, the four copper plates are added, fixed to the structure, so that a relative motion with the magnets in the moving part is produced by the vibration.

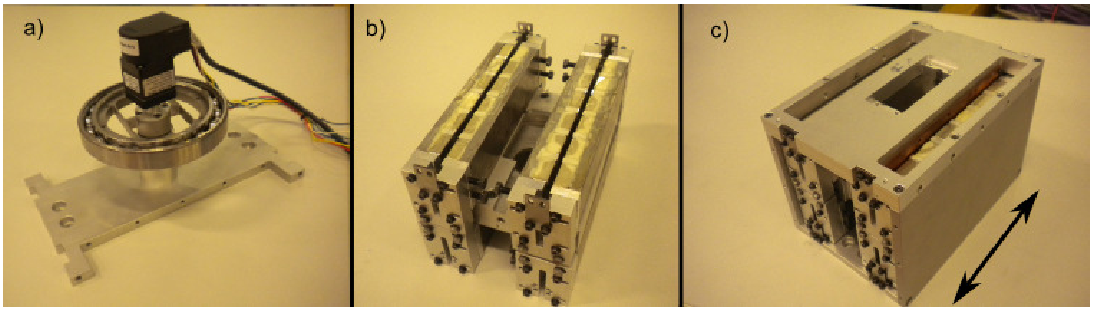

Figure 6: Assembly of damper a) rotary spring with motor b) magnet blocks and guiding elements c) damper prototype (protective enclosure missing)

4 Experimental validation of tuned-mass damper

In order to validate the approach presented here, some experimental tests have been carried out. First, the dynamic characteristics of the damper have been identified and compared to the expected values. Finally, machining tests have been carried out on a milling machine and the influence of the damper has been analysed.

4.1 Machine tool test bench

Machining tests have been performed to demonstrate the validity of the damper for chatter suppression. In this first research stage, the tests have been realized on a SORALUCE milling machine, mounting the workpiece on a flexible fixture. This fixture provides a dynamic response of the machine with a clear and isolated resonance mode, prone to suffer from chatter, and thus of help to avoid other disturbing effects, such us modes at similar frequencies, which would difficult the evaluation of the performance of the semi-active damper presented here. Anyway, this is still a realistic test case comparable to many industrial cases.

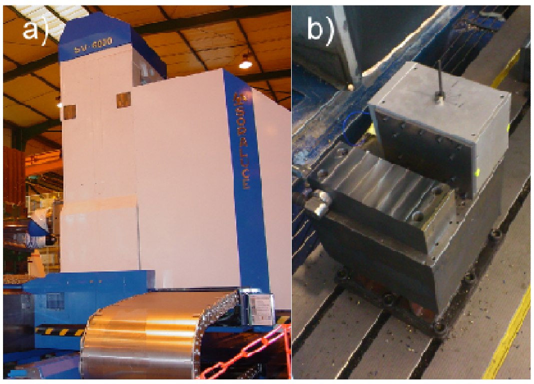

Figure 7: a) Soraluce milling machine b) workpiece fixture with damper prototype

4.2 Identification of damper characteristics

In a first step, the dynamic characteristics of the damper have been identified. An experimental modal analysis has been performed on the damper, using a hammer to excite the system and measuring the vibration at different points of the structure. The goal is to check that there is only one resonance mode in the frequency range of interest, and to see how its frequency changes with the angular position of the spring.

Figure 8a shows the variation of the FRF of the damper for different angular positions of the spring. A clear dominant mode is observed, which agrees well with the second order system that is expected from the theoretical analysis in Section 2.1. In Figure 8b, the variation of the resonance frequency of the damper in function of the angular position of the rotary spring is shown. The resonance frequency varies between 66 and 105 Hz.

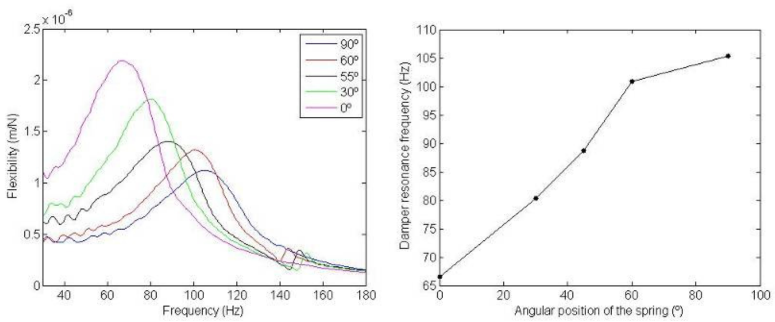

Figure 8: Dynamic response of the damper, experimentally measured: a) FRFs for different angular positions b) variation of the resonance frequency with the angular position of spring

These results demonstrate that the damper presented here fits well with the dynamic performance expected from a semi-active damper, with a dominant resonant mode in the direction and frequency range of interest. The results also demonstrate that the frequency tuning capability given by the angular rotation of the spring works well, and that the eddy currents can provide enough damping for this application.

4.3 Identification of the workpiece fixture

The next test has consisted in testing the influence of the inertial damper on the dynamic characteristics of the workpiece fixture, looking for the effect described in Section 2.1.

In Figure 9a, the effect of the damper on the magnitude of the flexibility FRF of the fixture is shown, where the strong reduction of the vibration amplitude due to the damper can be observed, together with its variation with the angular position of the spring.

In Figure 9b, the real part of the FRF is shown for the different angular positions of the spring. This figure is used to find the optimal damper tuning, by minimizing the negative real part of the FRF, as explained in Section 2.1 and [2].

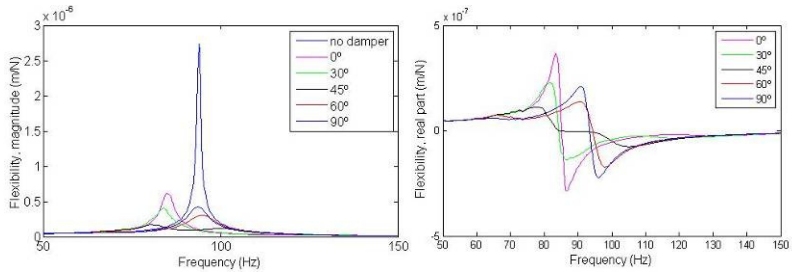

Figure 9: Flexibility FRF of the workpiece fixture (a) Magnitude (b) Real part

4.4 Machining results with manual tuning

The validation of the performance of the tuned-mass damper proposed here has been done by machining tests, comparing the results without and with damper, with the damper tuned manually to the optimal frequency as explained above. In each case, a single block of steel (F1140 (C45)) is machined with increasing cutting depths, therefore increasing the risk of chatter appearance, and using the same inserted cutter (SANDVIK 490-032A25-08M) and machine conditions.

Using the experimental modal analysis of the machine, the stability lobes of the cutting process with and without damper have been calculated following the method described by [14], as shown in Figure 10. The simulation parameters that have been used are shown in Table 3. The damper is expected to increase the chatter free area (above the lobe) by approximately a factor of ten.

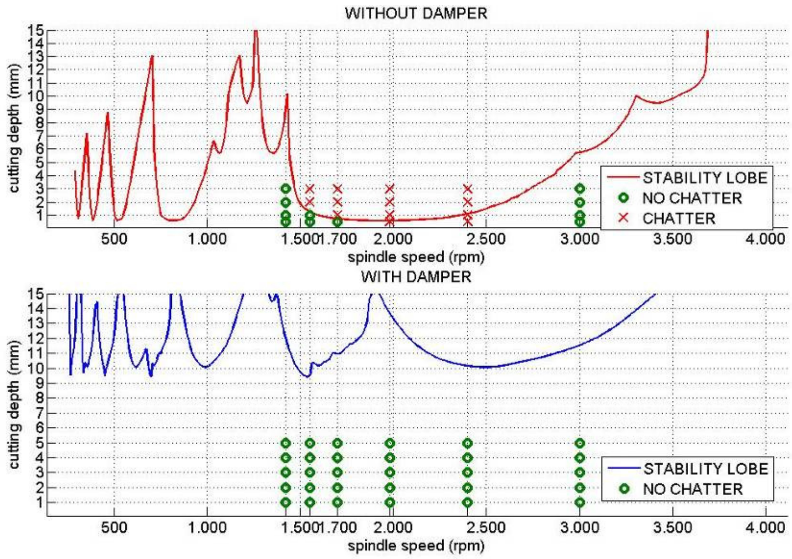

Figure 10: Machining test results with and without damper, showing predicted stability lobes (simulation) after modal analysis results, and chatter appearance at some cutting tests (experimental)

Tool				
Diameter, D (mm)	Number of flutes, Z	Helix angle, η (deg)	Lead angle, κ (deg)	
32	4	0	90	
Cutting conditions & coefficients				
K_t (N/mm^2)	K_r (N/mm^2)	Radial Immersion	Feed Direction (f)	
1459	257	30mm. Down Milling.	(1,0,0)	
Original Dynamic Parameters				
i	$\omega_{n,i}$ (Hz)	ξ_i (%)	k_i (N/µm)	Orientation (**v**)
1	94.0	0.66	58.4	(0,1,0)

Table 3: Simulation parameters

Some machining tests have been performed at different spindle speeds and cutting depths, showing good agreement with the stability lobes. Since the cutting tool only allowed up to 5 mm cutting depth, the stability limit using the damper has not been reached. The results are summarized in Figure 10. In Figure 11, the workpiece surface after machining is shown, with each column showing a different spindle speed. In a) chatter marks are observed in the central columns for 3 mm cutting depth, while in b) the no chatter marks are observed for a 5 mm depth.

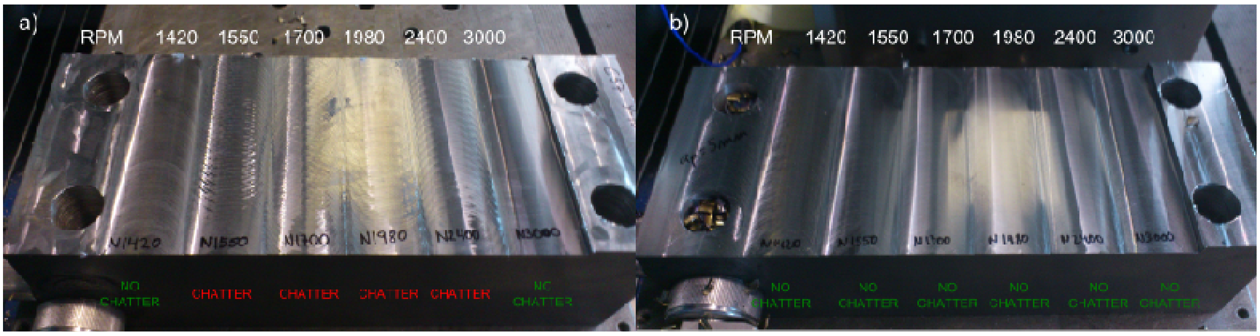

Figure 11: Workpiece surface after machining a) without damper 3 mm depth b) with damper 5 mm cutting depth

5 Control strategy for self-tuning damper

The tuned mass damper presented here needs a control strategy to enable self-tuning capabilities. The goal is to use the information coming from the accelerometers installed on the damper to detect online whether chatter is being generated, and in case it is, estimate the optimal tuning frequency tune the damper by commanding the stepper motor to the right damper tuning position, so that the process is stabilized chatter and machine vibration reduced. The control algorithm is presented here, but since it is still on implementation phase, no machining results can be provided at this point.

5.1 Chatter detection and suppression algorithm

The first step towards chatter suppression is to detect whether chatter is occurring or not during the machining process. The vibration measured by the accelerometer installed on the structure of the damper is processed in order to find its main frequency components. Here it is important to distinguish between forced vibrations, induced directly by the cutting forces, and chatter, which is an unstable regenerative process generated only under certain working conditions.

Forced vibrations appear at harmonics of the tooth passing frequency, but are stable, and thus are usually not a problem for machining, except in finishing operations where surface roughness needs to be improved. Chatter appears at other frequencies than tooth passing frequencies, and is an unstable cutting process, meaning that the cutting forces and vibrations increase with time, leading to unacceptable machining conditions, since they produce very bad surface quality and can lead to damage in the machine.

The chatter detection and suppression algorithm is presented in Figure 12.

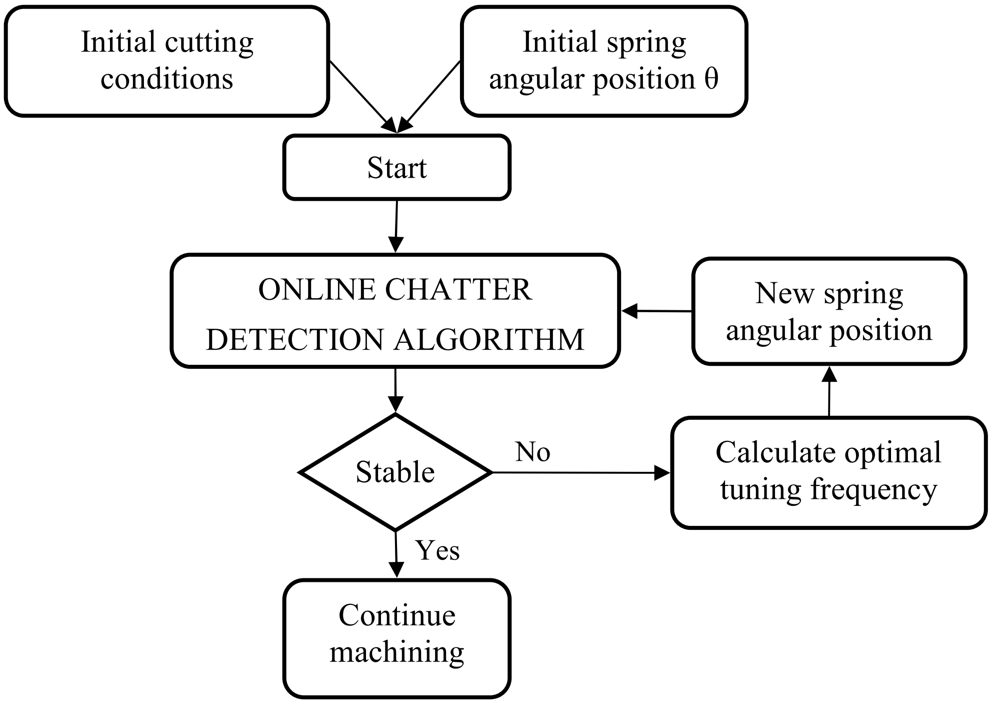

Figure 12: Chatter detection and suppression algorithm

This algorithm is implemented on a real-time controller. It is running continuously during the machining process, calculating the spectrum of the measured vibration of the machine, as shown in Figure 13. The algorithm detects the frequency of the maximum vibration peak, and compares it with the tooth passing frequency: if it is an integer multiple of the tooth passing frequency, it is considered a forced vibration, and no corrective action is taken. If it is not an integer multiple, it is considered to be chatter, and the angular position in the damper is modified in order to tune it with the chatter frequency.

It is very important to distinguish clearly chatter from forced vibrations, so that the damper is only tuned to chatter frequencies. Otherwise, once the damper is tuned to the chatter frequency, the vibration level at this frequency will drop, and the algorithm will detect a forced vibration as main frequency. If the damper is tuned to this new frequency, chatter generation could start again, so this needs to be avoided.

The detection algorithm first filters the acquired acceleration signal with a 500Hz low pass filter. Then The FFT of the filtered signal is calculated, finding the frequency of the highest peak. Then we apply another low pass filter to the frequency of the highest peak + 50 Hz, in order to increase the resolution in the spectrum.

When the highest peak's frequency does not match any harmonic of the spindle speed, chatter detection is considered positive. The criteria is that the absolute value of ((the peak frequency/spindle frequency) – harmonic of spindle frequency) > 0.1.

In Figure 13, the harmonics of the tooth passing frequency are 66, 100 and 132 Hz. In (a) the highest peak is detected at 96.5 Hz, and it is thus identified as chatter.

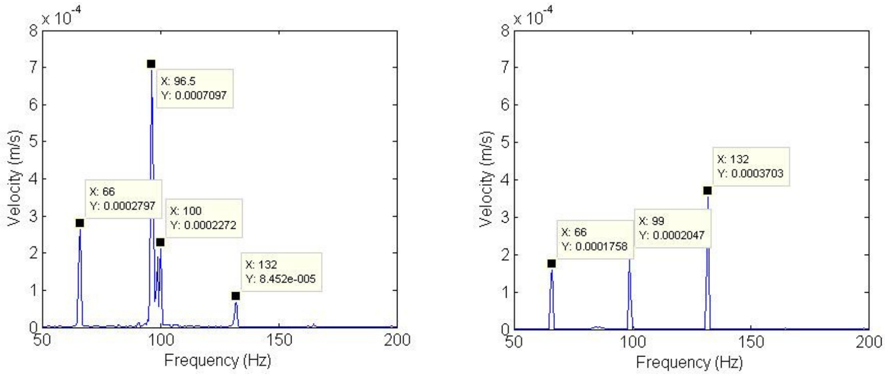

Figure 13: Spectrum of measured vibration (a) chatter is detected, 96.5 Hz (b) no chatter is detected

5.2 Simulation of chatter suppression algorithm

The stability of the chatter suppression algorithm has been assessed by simulations of the machining process (see [15]). In a first step, for certain cutting conditions, the algorithm detects whether chatter is being generated or not. In case it is, the damper is tuned to chatter frequency (which is not optimal the tuning frequency, according to [2]). The process starts again, simulating the stability of the process with the new damper dynamics. If chatter is not present, the damper tuning is not changed, and if chatter appears again, the new chatter frequency is detected and the damper is tuned to it.

The results of this simulation are shown in Figure 14. With the stability lobe of the process without damper as reference, the tuning steps needed for chatter suppression are marked in the figure. It can be observed that the simulation predicts a successful stabilization of the machining process in a wide range of cutting conditions that would have been unstable without the damper. It can be observed that the number of tuning steps needed increases with the cutting depth.

In Table 4, the evolution of the chatter frequency at the successive tuning steps is shown, for the case of 2400 rpm spindle speed. The convergence of the algorithm is shown, except for the highest cutting depths, where no stable solution is found.

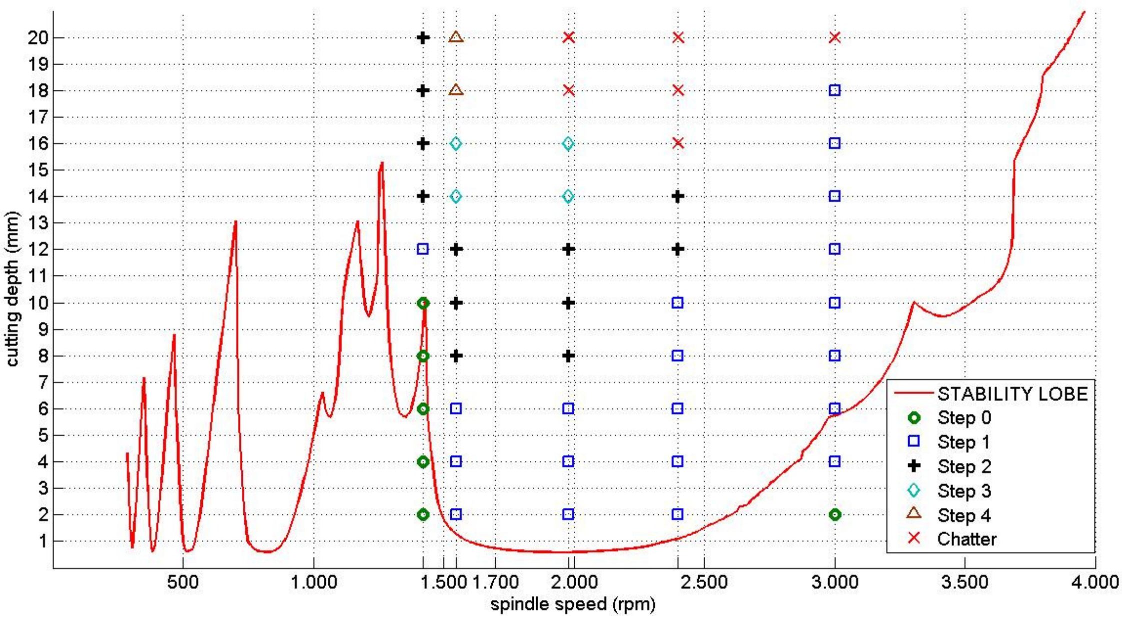

Figure 14: Simulation of machining results with chatter detection and suppression algorithm

Depth (mm)	2	4	6	8	10	12	14	16	18
Step 0	**96**	**98**	**99**	**101**	**102**	**104**	**106**	**109**	**111**
Step 1	**S**	**S**	**S**	**S**	**S**	**98**	**100**	**129**	**103**
Step 2	**S**	**S**	**S**	**S**	**S**	**S**	**S**	**115**	**117**
Step 3	**S**	**S**	**S**	**S**	**S**	**S**	**S**	**102**	**101**
Step 4	**S**	**S**	**S**	**S**	**S**	**S**	**S**	**117**	**115**

Table 4: Evolution of chatter frequency during tuning process (S=Stable; 2400 rpm)

5.3 Control hardware

A specific control unit is being developed by CEDRAT TECHNOLOGIES for the real time control of the damper proposed here. This controller reads the signals from the accelerometers, and processes them to detect whether chatter is being generated, and commands the stepper motor to tune the damper to the optimal damper frequency to eliminate chatter. The control algorithm, still in implementation phase, is described in Section 5.1.

Figure 15: Real-time controller developed by CEDRAT TECHNOLOGIES

6 Conclusions

A new design for a semi-active tuned-mass damper for machine tool chatter suppression has been presented. Having a fixed moving mass, the stiffness and damping are generated by different mechanisms so that they can be designed and selected independently.

The resonance frequency of this inertial damper can be varied in a range between 66 and 105 Hz by simply commanding a stepper motor that changes the orientation of a spring with variable stiffness. Damping is generated by energy dissipation produced by the eddy currents induced on conductor plates, by a magnetic field generated by magnets mounted on the moving mass.

Experimental machining results demonstrate the tuning capabilities of the damper, and its effectiveness in chatter suppression. Since the self-tuning control strategy is still in implementation phase, in these tests, the damper is tuned manually. However, simulations of the machining process have demonstrated the validity of the chatter detection and suppression strategy, which will be tested experimentally in the near future.

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