Self-tuning dynamic vibration absorber for machine tool chatter suppression

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SELF-TUNING DYNAMIC VIBRATION ABSORBER FOR MACHINE TOOL CHATTER SUPPRESSION

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INTRODUCTION

The current trend in machine tool design aims at stiffer machines with lower influence of friction, leading to faster and more precise machines. However, this is at the expense of reducing the machine damping, which is mainly produced by friction, and thus increasing the risk of suffering from a self-excited vibration named chatter, which limits the productivity of the process. Dynamic vibration absorbers (DVAs) offer a relatively simple and low cost solution to reduce chatter appearance risk by adding damping to the critical vibration modes. The proper tuning of the dynamic characteristics of the damper to the machine/process dynamics is the key for successful implementation.

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

- Automatic in-process chatter detection and optimal tuning frequency calculation by means of dedicated hard/software and embedded accelerometers.
- Automatic in-process tuning of the resonance frequency of the damper to the chatter frequency by means of a variable stiffness spring controlled by a motor.
- Damping is produced by eddy current effect, generated by the vibration of conductor plates within a magnetic field.

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. This damper has been designed to fit the requirements of a SORALUCE milling machine at IK4-IDEKO's facilities, which has been used for experimental validation.

The main design and working principles of this damper and the improvement of the machining conditions allowed by the damper will be demonstrated by real milling experiments.

DVA REQUIREMENTS

Dynamic vibration absorbers consist of a mass connected to the machine with a certain stiffness and damping, so that its resonance frequency is tuned to the frequency of the machine mode leading to chatter, by adding damping to it and allowing higher cutting depths [1].

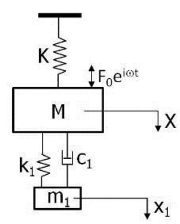

FIGURE 1. DVA concept, m_1 mass connected to machine M with k_1 and c_1

Tunable DVAs are needed because the effectiveness of the damper depends greatly on an accurate tuning of the damper frequency. Furthermore, even within the same machine, chatter frequency can vary with the position in the workspace (due to the variation in the machine stiffness), with the process parameters and with the workpiece mass [2].

A self-tuning DVA is thus of advantage for ensuring optimal damping in a wider range of working conditions. Such damper needs two main new functions:

- Online chatter monitoring and identification of optimal damper tuning frequency
- Automatic control of the damper resonance frequency to match the monitored one.

DAMPER DESIGN

Following the requirements described above, a new self-tuning DVA concept is presented here. The main novelties of this DVA are (i) online detection of optimal tuning frequency, (ii) automatic control its resonance frequency and (iii) decoupled damping and stiffness generation with linear characteristics. The main design aspects enabling these characteristics are described next.

Stiffness control

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.

This is commonly achieved by means of elastomers, which provide required stiffness and damping. Stiffness is then controlled by applying different preloads to the material. An evolution of this principle is the use of magnetorheological elastomers, which change in stiffness when a magnetic field is applied on them [3]. These technical solutions are relatively simple to implement, but are difficult to tune due to the nonlinear behavior of the elastomers. Other authors propose pre-stressable leaf-springs, which offer a more predictable behavior [4].

In this project, a rotary element with variable stiffness in function of the angular direction has been selected. A rotary stepper motor controls the orientation, and thus, the stiffness and frequency. Linearity, predictability and repeatability are the 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 1. 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

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 behavior and it is generated without contact between parts, avoiding, for example, the non-linear effects produced by friction.

When a conductor moves through a static and non-uniform field, eddy currents are induced on it, and they generate a counteracting field which generates in interaction with the original field 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 [6].

The magnet configuration shown in Figure 3 has been selected, in which the poles are arranged in an alternating pattern that increases significantly the damping force and coefficient, and minimizing the required volume. This module design allows fixing the neodymium magnets easily, and then handling and mounting each module in the damper independently.

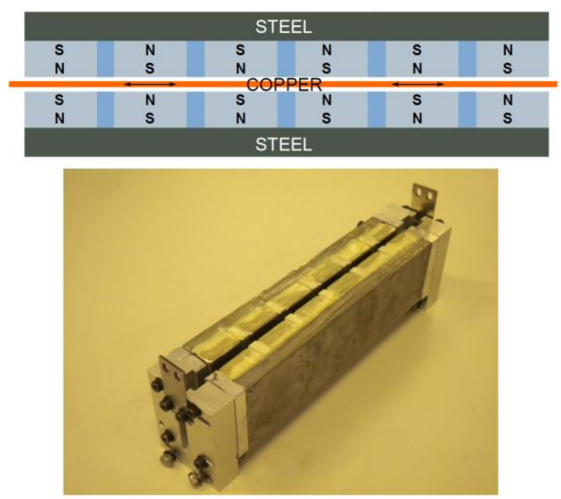

FIGURE 3. Magnet configuration

Self-tuning strategy

The self-tuning requirement aims at ensuring that the damper is always optimally tuned in an automatic way. The first step is to detect whether chatter is occurring or not during the machining process. If it is not occurring, no action is needed. If chatter is detected, the damper needs to be tuned to the right frequency.

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 it 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 proposed chatter detection and suppression algorithm is presented in the next figure.

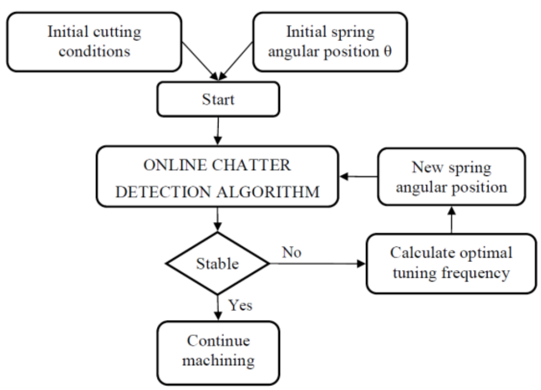

FIGURE 4. Online chatter detection algorithm

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 5. 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.

In Figure 5, 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.

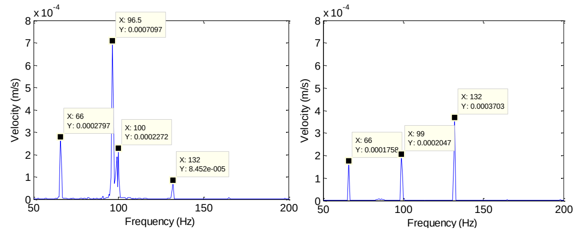

FIGURE 5. Spectrum of measured vibration (a) chatter is detected, 96.5 Hz (b) no chatter is detected

EXPERIMENTAL SETUP

Milling machine

Machining tests have been carried out on a SORALUCE milling machine, mounting the workpiece on a flexible fixture for development and evaluation purposes (see Figure 6). 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 evaluation of the performance of the semi-active damper presented here. Anyway, this is still a realistic test case comparable to many industrial cases.

DVA prototype

A DVA prototype has been built to meet the requirements of this test bench, which shows a critical mode at 94 Hz and 150 kg modal mass. With a moving mass of 7 kg, the DVA can change its main resonance frequency between 65 Hz and 105 Hz, providing an estimated 800 Ns/m damping, values which are in range with the optimal. As it can be seen on Figure 4, the moving mass is formed by the four magnet racks, with the copper plates fixed to the frame, providing thus a very compact system. An accelerometer is placed on the frame to measure machine vibration.

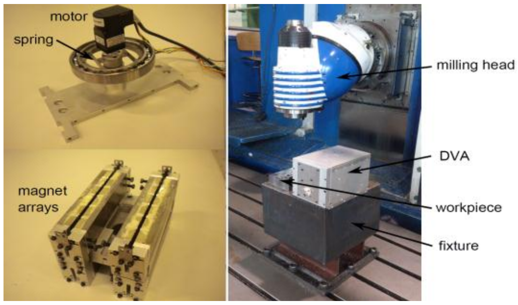

FIGURE 6. Machine tool and DVA setup

EXPERIMENTAL RESULTS

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 7a shows the variation of the FRF of the damper for different angular positions of the spring. A clear dominant mode is observed, with its amplitude and frequency changes with the spring position. In Figure 7b, 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.

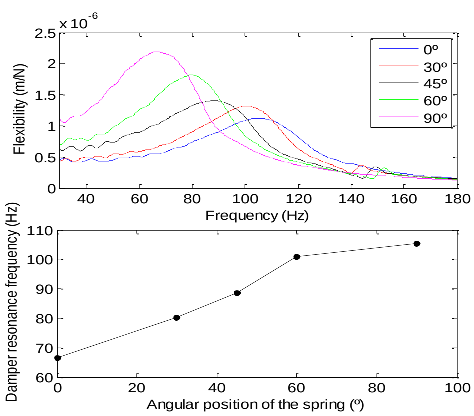

FIGURE 7. Dynamic response of the damper, experimentally measured: a) FRFs for different angular positions b) variation of the resonance frequency with the angular position of spring

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.

Identification of fixture characteristics

The next test has consisted in testing the influence of the inertial damper on the dynamic characteristics of the workpiece fixture. In Figure 8a, 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 8b, 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 proposed for chatter suppression in manufacturing by [7].

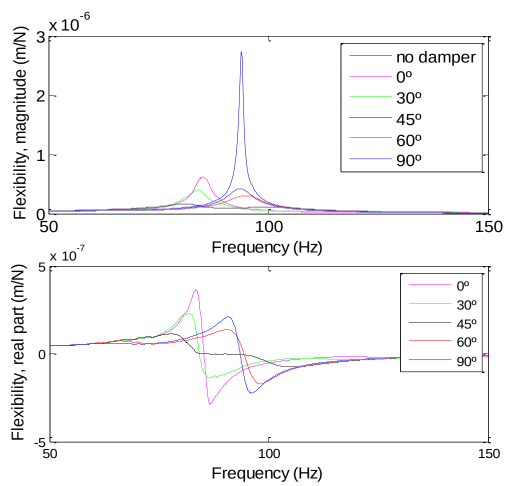

FIGURE 8. Flexibility FRF of the workpiece fixture (a) Magnitude (b) Real part

Machining results

The stability lobes [8]-[9] of the cutting process have been calculated by an experimental modal analysis, in order to show the maximum cutting depth that can be achieved without the damper. Machining tests show the validity of this prediction. The DVA was placed on the machine next, and it was tuned automatically during the cutting process, without using the information from the modal analysis. The process was found to be stable up to the maximum cutting depth defined by the tool (5 mm), compared to the unstable conditions with 1 mm depth without damper. A time simulation [10]-[11] of the cutting and tuning process predicts much higher stable cutting depths (see Figure 9), but they cannot be reached due to tool limitations.

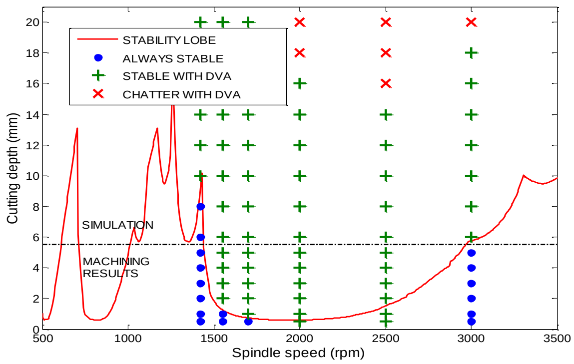

FIGURE 9. Machining stability test and simulation results, with and without DVA

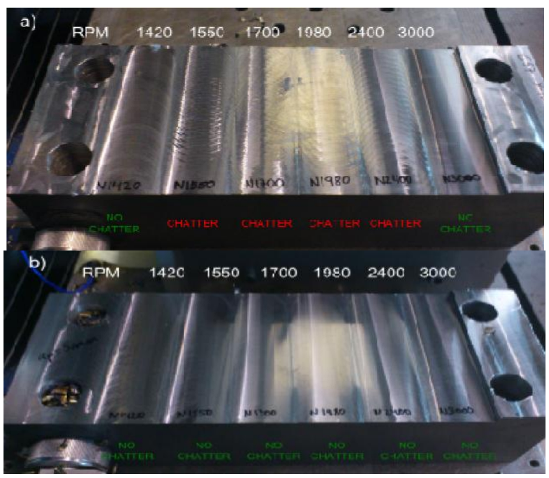

FIGURE 10. Workpiece surface after machining a) without damper, 3 mm depth b) with damper, 5 mm cutting depth

CONCLUSIONS

These results demonstrate the effectiveness of the self-tuning DVA principle presented here, which detects continuously during the machining process whether chatter is happening or not, and automatically tunes itself to the optimal position.

In real applications, productivity improvement will not be as high as the one obtained in this research test bench, but it will outperform existing DVAs by providing a low cost solution that does not require a previous experimental modal analysis and that works in close-to-optimal conditions even when process dynamics change during operation.

ACKNOWLEDGEMENTS

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