A simplified vibration compensation through magnetostrictive actuators

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Abstract: This paper addresses the vibration reduction of a simple structure, simulating a workpiece carrier plate, through a cancellation technique based on the use of smart compensation actuators. The proposed compensation strategy is implemented resorting to magnetostrictive actuators acting far from their resonant frequencies. Specifically, the procedure has been tested on two test rigs designed with particular reference to machine tools applications: they require the use of one and three magnetostrictive actuators for 1D and 3D vibrations compensation.

The parameters of the compensation strategy are identified from the direct characterization of the actuators embedded in the structure. The resulting reduction of the lateral vibrations of 1D and 3D plates in the frequency range up to 400 Hz proves to be very satisfactory.

Keywords: magnetostrictive actuator, characterisation, vibration compensation

1. INTRODUCTION

Vibration compensation is a perennial problem in applications requiring high accuracy in positioning or in mechanical measurement. For many applications there are usually efficient passive suppression systems, especially when the disturbance is predictable or does not change continuously versus time, or the system does not changes its natural frequencies; otherwise vibration reduction is obtained by active systems. Common examples of vibration reduction are reported in aerospace, automotive and general industrial applications Kamesh et al. (2010), Vaillon and Philippe (1999), Wu at al. (2000), Li et al. (2012).

In the present paper the authors focus on the vibration reduction in industrial machine tools applications, especially milling. In this application the machining accuracy is limited by the vibrations of the spindle and of the machine structure. Compensation would ensure a better processing and diminish chatter. To reduce the effect of the natural frequencies of the system, masses and dampers tuned to natural frequencies of the structure can be added. A further possibility consists in changing the spindle speed depending on the type of processing: this solution is quite cumbersome and sensitive to structure changes. Alternative or complementary systems for the vibration reduction are the active vibration compensators, using the action of one or more smart actuators -Rashid and Nicolescu (2006), Shaw (2001)- where the latter could also produce energy (energy harvesting) -Davino et al. (2011).

The goal of this work is to use smart actuators as counter-phase shakers, to obtain vibration reduction: the actuators task is to create mechanical forces opposing the movement of the vibrating structure. The compensation in this case is particularly effective when the vibration to be compensated exhibits a predominant harmonic component. This happens exactly in milling operations, where the frequency of the disturbance is related to the rotation speed of the tool and to its number of teeth. The extension of such technique to a system in motion, as the spindle, is really challenging, despite some noticeable examples reported in literature –Dohner et al. (2004), Denkena et al. (2007). In this paper a fixed workpiece carrier plate is considered, which constitutes an inertial system in the kinematic chain of the machine. The paper is organized as follows: in section 2 the choice of the magnetostrictive actuators is motivated and their magneto-mechanical characterization is presented, while section 3 is devoted to the analysis of the proposed cancellation algorithm. Section 4 contains the application of the cancellation procedure to the 1D test rig while in sections 5 and 6 the algorithm is applied to the 3D case, i.e., to an archetype of a modular workpiece carrier plate. The experimental results are discussed in section 7.

2. COMPENSATION SHAKER

The active vibration compensation or cancellation is usually achieved through smart structures, which integrate components based on functional materials and are able to reach frequencies and amplitudes related to mechanical vibrations. A comparison between the existing and emerging technologies can be found in Pons (2005). To compensate the vibrations of a machine tool, forces of hundreds (or thousands) newton are typically necessary while the stroke varies from a few to tens of microns; the frequency reaches some hundreds hertz, depending on the number of cutting edges of the tool. These conditions lead towards two technologies: piezoelectric and magnetostrictive. In the present paper the second technology has been investigated. An overview of magnetostriction and magnetostrictive actuators can be found in Ekreem et al. (2007), Olabi and Grunvald (2008). Early studies concerning the vibration compensation through magnetostrictive actuators can be found in Hiller et al (1989), Geng and Haynes (1994) and May et al (2003).

Using an actuator as a compensation shaker can be a simple and effective approach if attention is focused only on the amplitude and phase to be generated. The amplitude is herein intended as the peak of the displacement whilst the phase is the delay between the input command (voltage or current) and the resulting displacement. By driving the actuator with an harmonic command at a given frequency, amplitude and phase are repeatable although the behaviour of the actuator is nonlinear or hysteretic. Above proposition is not true in any condition so that, for a given type of smart actuator, it is necessary to define the conditions under which the above statement is valid. In this work, we used a few commercial actuators based on Terfenol-D ($Tb_{0.3}Dy_{0.7}Fe_{1.92}$), and the following consideration will clarify the statement.

Figures 1 and 2 show interpolations of the experimental characterization of a Terfenol-D sample using the approach based on the Classical Preisach Model (CPM) and described in Bottauscio et al. (2011). Figure 1a confirms that the mechanical deformation (displacement of the device) is largely dependent on preload. Moreover, with a fixed preload, the displacement presents a clear frequency dependence (Figure 1b).

With a fixed preload value and providing a constant magnetic bias, which is the rule in the actuators, the elongation at constant driving current (that is magnetic field excitation) is shown in figure 2. In this case the actuator acts as a compensative shaker and the elongation depends basically on both frequency and electrical (current) excitation.

In the paper the following conditions will be assumed:

- the dynamic load on the actuator is negligible with respect to preload;

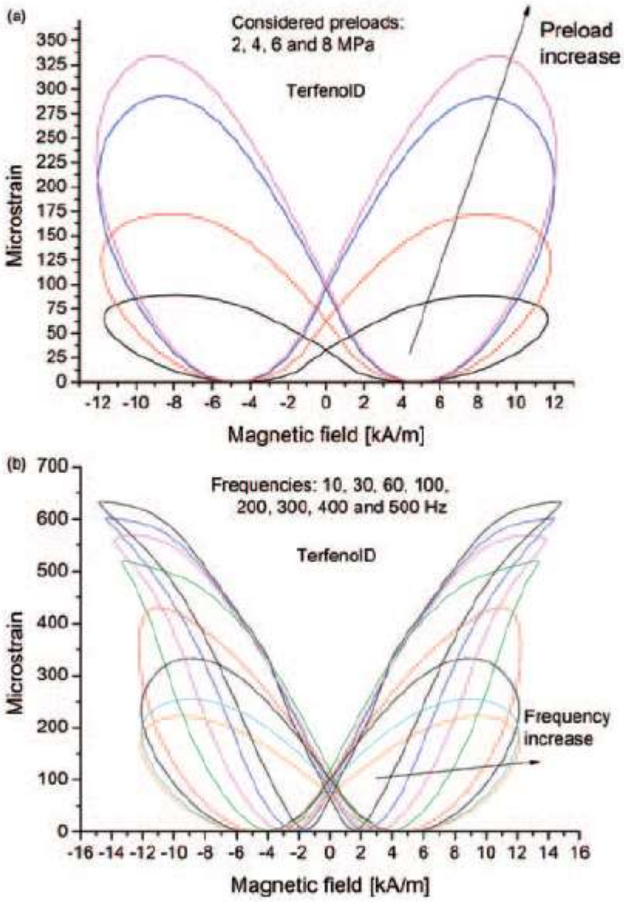

Figure I. (a and b)

- the mechanical load is high enough to neglect non-homogeneities of the active material;
- the frequency of the disturbance to be compensated is generally well determined because the vibrations are induced by a cutting tool spinning at a constant speed and the variation range is well known.

Under the above hypotheses, the non-linear behaviour of the actuator, and hence of the whole system, varies substantially as a function of two parameters namely the magnetic field (controlled by. the excitation current of the actuator) and its frequency. Recent insight concerning this non-linear behaviour are found also in Aljanaideh et al 2013. In such a case the vibration compensation has to be achieved through a non-linear controller, whose characteristic can be measured by means of the approach proposed in section 4.2.

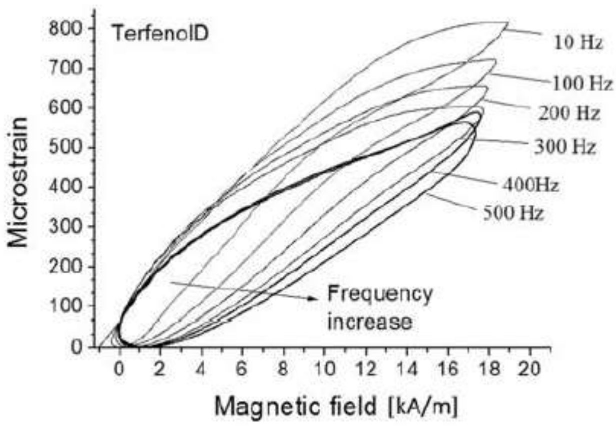

Figure 2.

A model based on the well known piezomagnetic equations IEEE (1991) would not be adequate because it should yet take into account the nonlinearities, as reported e.g. in Zhou et al. (2006)

When the processing is characterized by high velocity gradients of the tool or the mechanical stress significantly varies with respect to the preload, such a treatment is no longer valid. A comprehensive modelling approach would then be necessary: examples of vibration compensation are reported in Kumar et al. (2004), Moon et al. (2007). Nonlinear hysteretic models are employed in micro-positioning problems as well as in the "slow" active vibration compensation (few Hz) and, in this case, modelling can be based for example on the CPM – Bottauscio et al. (2010), Natale et al. (2001) – or on energy based (e.g. Armstrong model) and phenomenological models – Smith et al. (2003), Bashash et al. (2011), Nealis and Smith (2007).

3. COMPENSATION STRATEGY

The scheme used for the compensation is shown in figure 3 where the vibration reduction is obtained by using the smart actuator as a compensation shaker. The core of the system is the controller which drives the amplifier of the magnetostrictive actuator and therefore the counteractive force. The controller processes the frequency and amplitude of the acceleration (1st harmonic) and provides the modulus and phase of the electrical current feeding the actuator amplifier (eqns (4) and (5) in the following). In order to correctly compensate the external disturbance, the characterization of the amplifier-actuator system has to be accomplished with the actuator embedded in the smart structure.

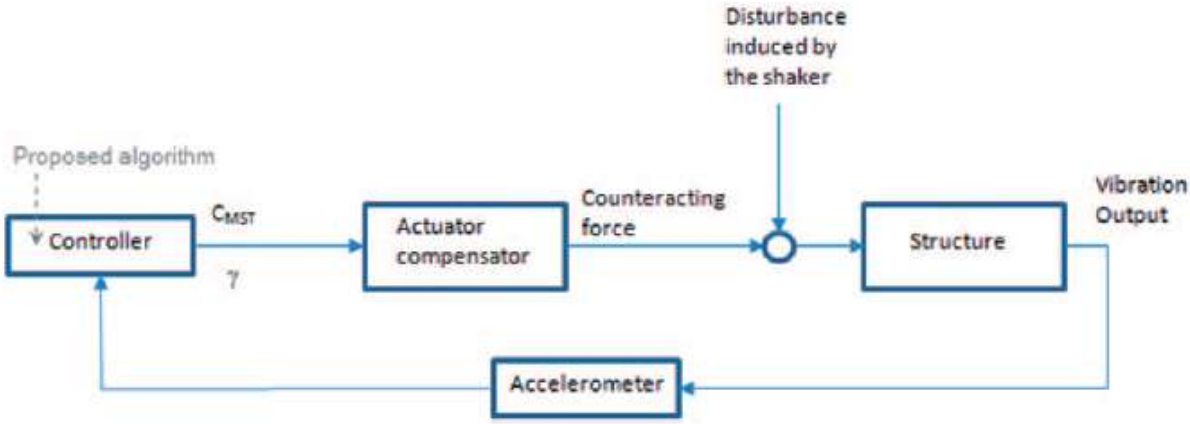

Figure 3.

3.1 Single actuator compensation algorithm

The aim of the proposed compensation algorithm is the vibration cancellation at a specific point of a structure or at least a significant reduction of the acceleration. Compensation is accomplished far from the actuator resonant frequencies and is based on the first harmonic component of the acceleration signal.

Through experimental measurements, i.e., by exciting a single actuator with a sinusoidal current at different commands values and frequencies, one determines (figure 4a) the function Γ which fits the measured acceleration a_{meas}

$$a_{meas} \cong \Gamma(C_{MST}, f) = \sum_{i=0}^{N} \sum_{j=0}^{N} p_{ij}\, C_{MST}{}^{(i-1)}\, f^{(j-1)} \qquad (1)$$

where C_{MST} is the peak value of the driving current of the magnetostrictive actuator (*command* in the following), f is the frequency, p_{ij} are scalar coefficients identified by interpolation and N is the degree of the polynomial interpolating surface (N=3 in this paper).

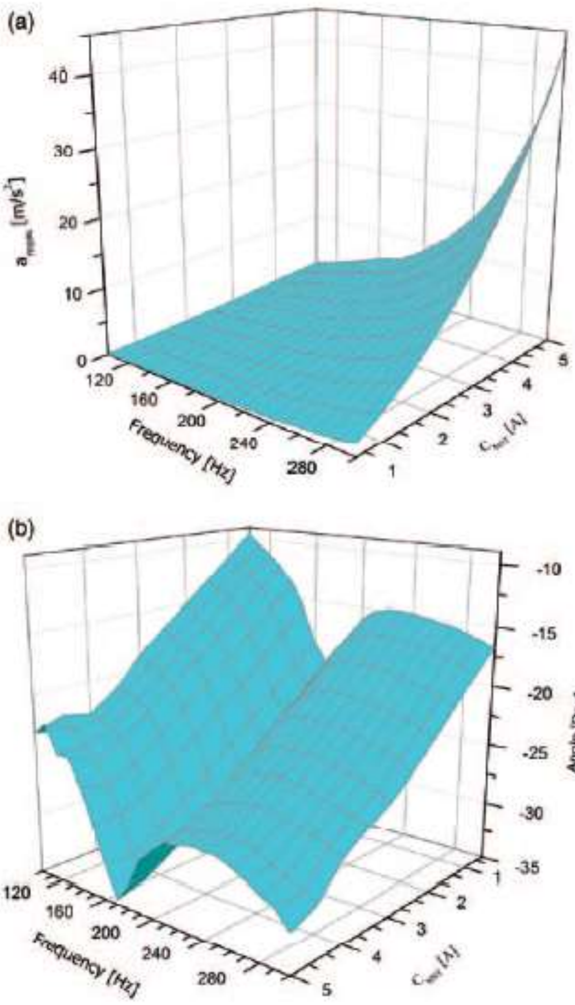

Figure 4. (a and b)

Through the same measurement one defines also the function

$$\beta = \beta\ (C_{MST},\ f) \qquad (2)$$

which gives the delay angle (figure 4b) between the command C_{MST} to the magnetostrictive actuator and the corresponding acceleration signal (first harmonic).

The compensation starts by measuring the acceleration a_{meas}^{*} and the frequency f^{*} in the point of interest which, according to eq. (1), correspond to a suitable compensation command C_{MST}*.

In the following two special additional specifications of the above procedure are required:

1) the compensation is not possible when the measured frequency is outside the range of the actuator;

2) when the acceleration is higher than the admissible one, the solution procedure drawn in figure 5 is proposed. Specifically, with the pair (a", f ") the command $C^{''*}$ is outside the domain, i.e., the actuator cannot generate the required acceleration at that frequency; in such a situation the maximum command $C^{''} = C_{max}$ is assigned to the actuator but the compensation will be incomplete.

The actual acceleration a_{act} is recorded by an acquisition card as measured acceleration a_{meas}, so that $|a_{act}| = |a_{meas}|$ but the following phase shift is introduced:

$$\delta = \delta(f_s, f) \tag{3}$$

which is related to the acquisition card and depends on both the frequency of the acceleration signal f and the sampling frequency f_s.

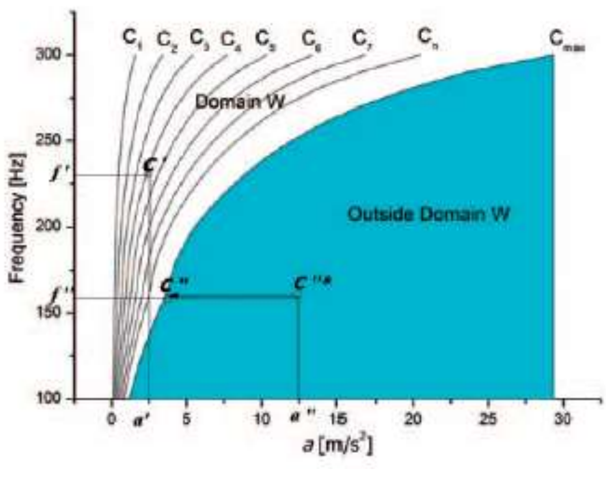

Figure 5.

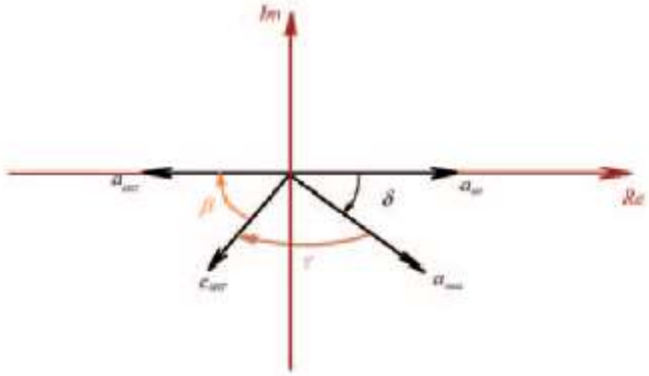

Figure 6.

The cancellation scheme is presented in figure 6. The basic idea is to generate a first harmonic acceleration a_{MST} equal and opposite to a_{act} so that three conditions are verified:

1. the generation of the magnetostrictive actuator control is synchronous with the measured acceleration;
2. the command magnitude can be computed by inverting (1). Except for the two above mentioned special cases one has

$$C_{MST} = \varphi(a_{meas}, f) = \Gamma^{-1}(C_{MST}, f) \tag{4}$$

3. the command must be given with a delay γ

$$\gamma\left(\left|a_{meas}\right|, f\right) = -\pi - \beta\left(C_{MST}, f\right) - \delta(f_s, f) \tag{5}$$

Correctly setting the delay γ is compulsory to achieve a satisfactory compensation Zhao et al. (2011).

4. EXPERIMENTAL RESULTS ON A 1D TEST RIG

4.1 Free behaviour of the actuator

Sonic actuators 050-LLAS produced by Etrema Inc. have been utilized for this work. The actuators with their amplifiers were characterized in a clamped-free condition (the moving pin was not connected to the compensation structure), in a frequency range between 50 Hz and 1000 Hz. The results in figure 7 show a fairly quadratic acceleration behaviour up to 500 Hz: in this range the displacement is then proportional to the command and is thus of interest for compensation.

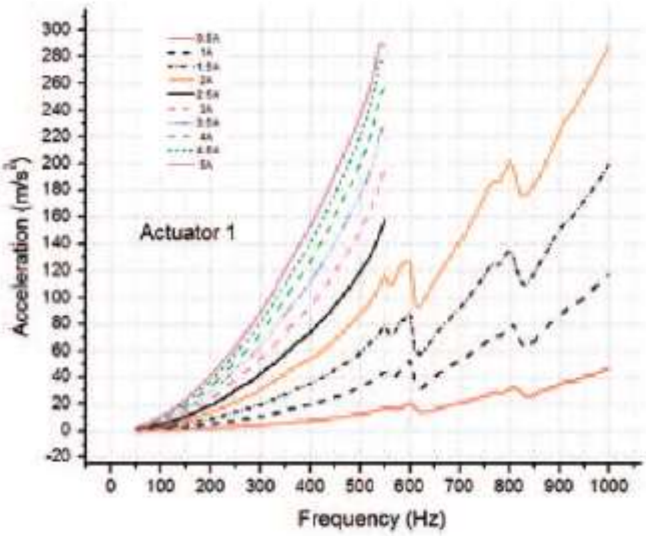

Figure 7.

4.2 Self tuning on a beam-like structure

The algorithm was tested on a first setup - details in Raffa et al (2011) - whose first resonant frequency is about 460 Hz. The external vibration source is an electrodynamic shaker, anchored to the ceiling (figure 8) and acting on the mid point of a thin plate.

The control algorithm was implemented on a PXI system and managed through a Labview program.

The first step for the algorithm validation was the characterization of the smart actuator inside the setup. Since such a characterization can be fully automated and repeated when changing a structure component, it is actually equivalent to a system self-tuning, which requires:

- to define the compensation point (accelerometer position);
- to define the command span;
- to define the frequency span;
- to decide a minimum frequency and a step increase;

- to decide a minimum command and a step increase;

1. For each command value, sweep the frequency span, step by step.
2. Increase the command and go to step 1.

In the characterization phase the functions Γ (eq. 1) and β (eq. 2) are built and both the compensation amplitude (4) and the phase (5) are computed. It has to be emphasized that the structural configuration of the system must be the actual one: for instance, the shaker remains connected to the structure.

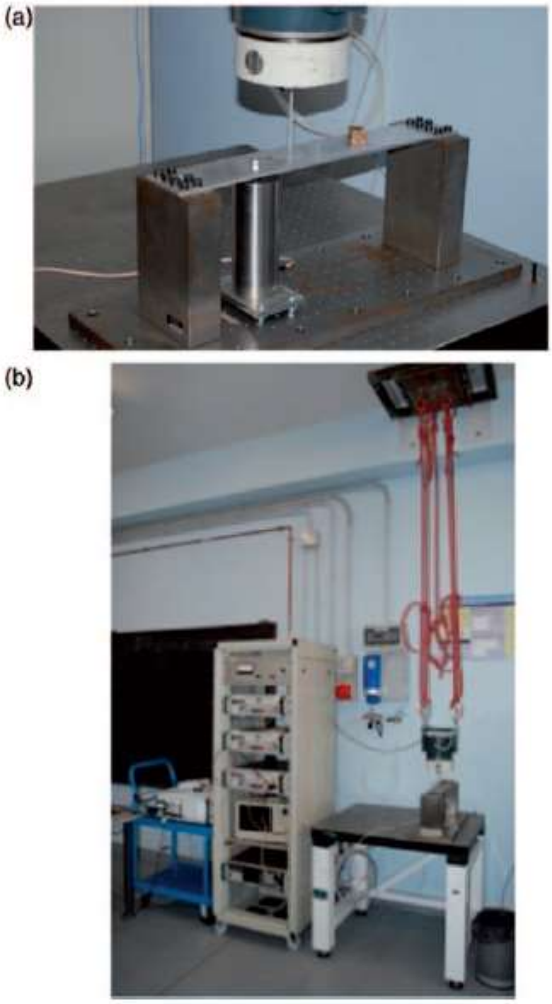

Figure 8.

4.3 Vibration reduction results

Two time histories of the acceleration before and after compensation are shown in figure 9, at 300 Hz. Since the cancellation concerns the fundamental harmonic only, an evident noise appears on the residual waveform.

With a_{nc} and a_c the acceleration amplitude (peak) of the uncompensated and compensated vibration signals respectively, the vibration reduction R is defined as $R=100\left(a_{nc} - a_c\right)/a_{nc}$

Figure 10 shows the vibration reduction R for different vibration frequencies and excitation amplitudes. It ranges between 60% and 90% of the uncompensated acceleration. The excitation amplitude is defined as the ratio between the actual supply voltage of the excitation shaker and the voltage corresponding to an uncompensated acceleration of 50 m/s^2.

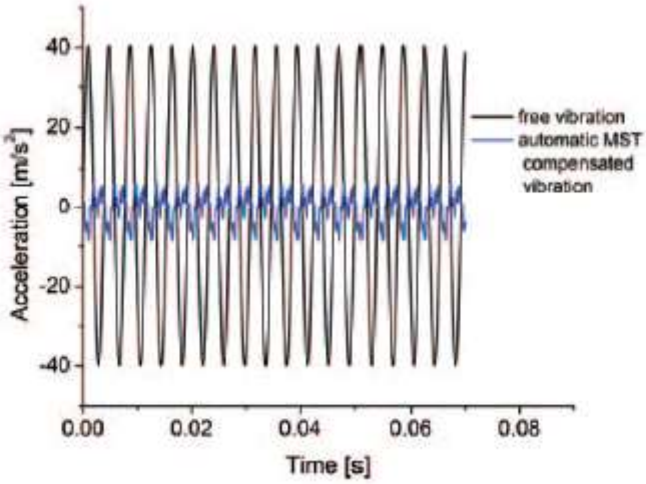

Figure 9.

Notice that the measurement of the vibration compensation was assessed for eight different values of the excitation amplitude and that the response of the controller is neither constant nor linear, but indeed effective.

5. DESIGN OF THE TEST RIG

In order to test the control algorithm in 3D conditions, a new test rig has been designed and manufactured. The aim was to obtain a simple structure, with a modular assembly, quick to modify, capable of hosting the magnetostrictive actuators and, with great simplicity, able to model the vibration of the tool table of a milling machine.

The result is represented in figure 11 where four vertical beams (square section) are clamped to a thick base plate and support a square plate on their other ends. The beam length depends on the magnetostrictive actuators (three cylinders) and their transverse section has been evaluated not only to prevent any relevant bending in the frequency range of the control (0-500 Hz).

The upper plate may be thick or thin: in the first case, flexible links with the vertical beams are necessary to respect the frequency range of interest; in the second case, reported in figure 11, an appropriate screw is tightened so that the motion is entirely due to the flexibility of the plate.

The position of the actuators is fixed while three different plates, with aspect ratios of 1, 1.5 and 2, can be mounted and tested.

The test rig was designed resorting to extensive static and dynamic finite elements (FE) analysis in order to correctly define the relevant dimensions, to simulate the behaviour of the actuators (with their power off), and to check the effectiveness of the aforementioned links for both thick and thin upper plates.

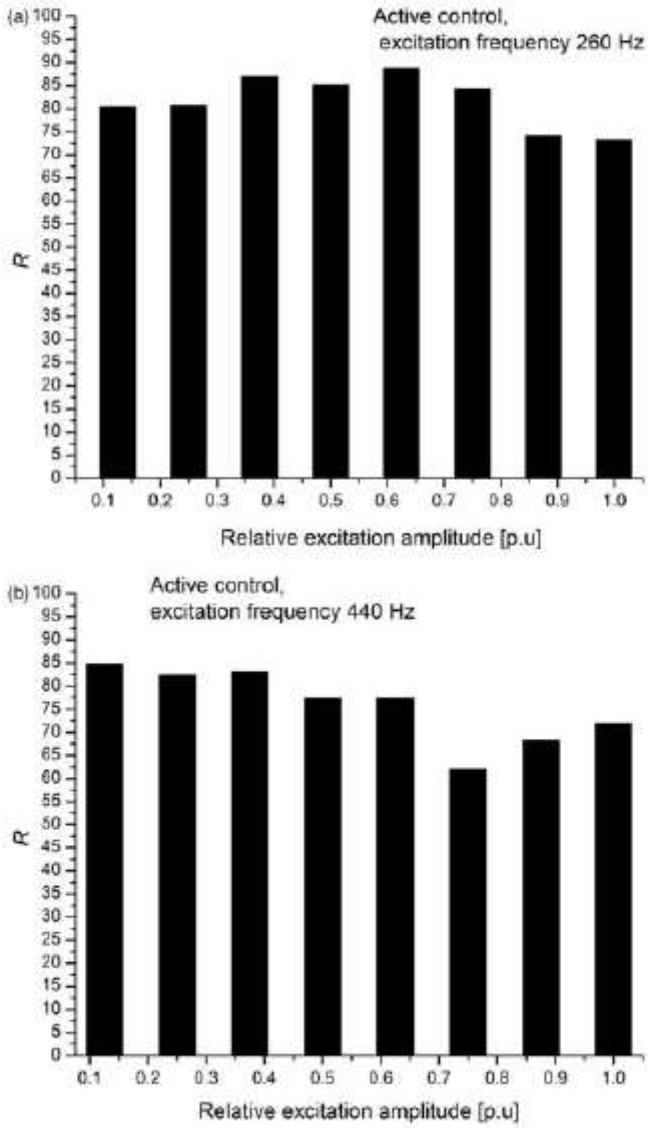

Figure 10. (a and b)

The analysis was focused on the dynamic behaviour of the structure. As an example the first mode shapes for the square and the rectangular plate (aspect ratio 2) are reported in figure 12. It may be worth noting that the mode shapes are largely influenced by the actuators, even when they do not receive an energy supply, because their stiffness is not negligible with respect to the upper thin plate (aluminium, 0.3x 0.3 x

0.003 m^3 or 0.3 x 0.6 x 0.003 m^3). The graphical representation of the mode shapes is useful when selecting the positions of both the input force and the accelerometer: in particular, the input force (external disturbance) has been applied in proximity of point of maximum displacement.

6. IMPLEMENTATION ON THE WHOLE STRUCTURE

The experimental set-up, including the acquisition system and the signal conditioning and control, is shown in figure 13.

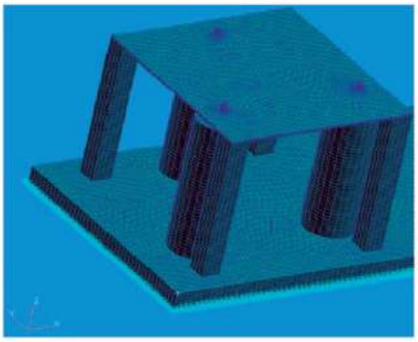

Figure 11.

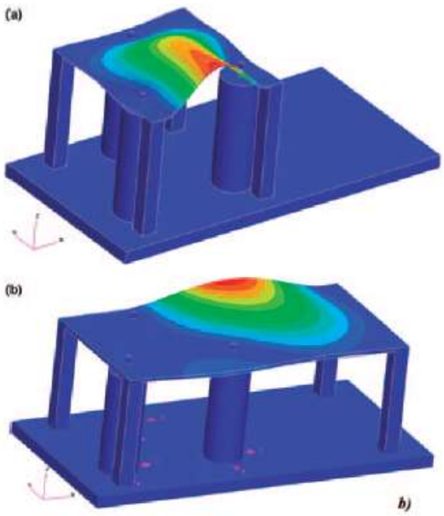

Figure 12. (a and b)

The compensation parameters have been obtained, as a function of the vertical acceleration, as described in section 4.2. In this case each actuator has singularly been characterized in the structure: the three of them were fixed to the structure but only the one under investigation was excited.

Each actuator has been operated with a command value and a proper gain. The gain for each actuator has been determined following a heuristic procedure based on a sequential tuning of the actuators, aiming at the minimum vibration amplitude.

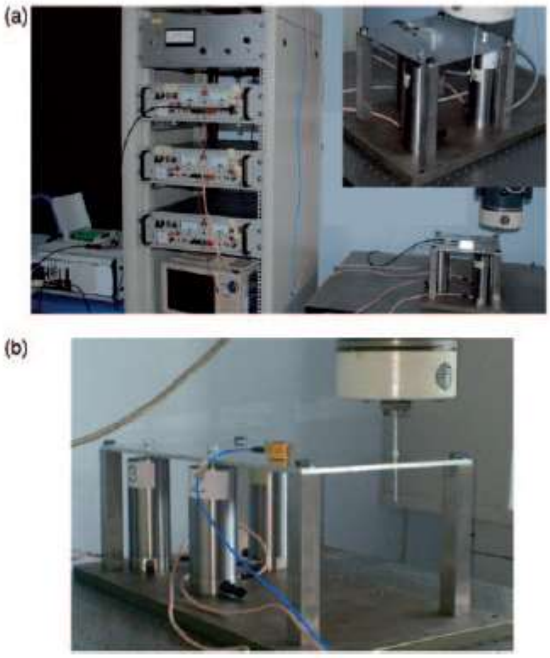

Figure 13. (a and b)

The heuristic approach can be applied in a few minutes, provides good results and in a given structure it is to be applied only once.

The gains are dependent on the geometrical position of the actuators and the compensation point; for example the following table summarizes the gains in the case of the two plates of figure 13 (square and rectangular, aspect ratio 2).

Table 1. Actuators gain.

Actuator	*Gain (square plate)*	*Gain (rectangular plate)*
1	*0.6*	*0.10*
2	*0.4*	*0.05*
3	*0.4*	*0.95*

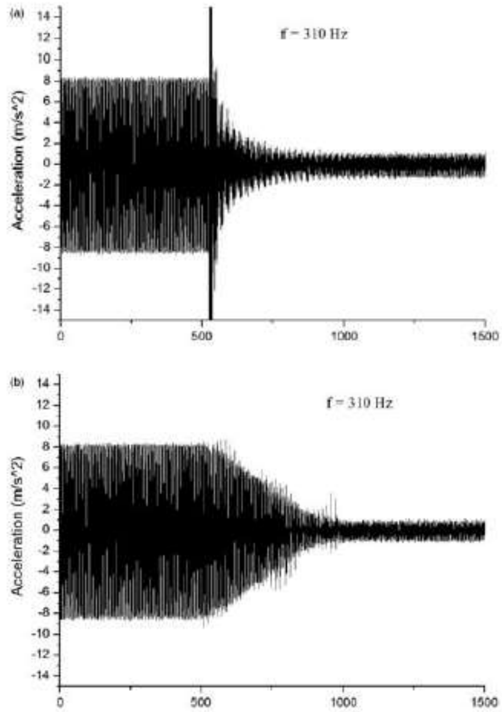

Figure 14. (a and b)

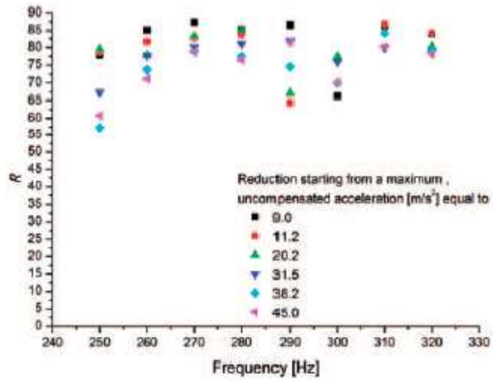

Figure 15.

The actuators are simultaneously driven and the actuation time can be extremely small, even a few milliseconds, but not surprisingly acceleration peaks appear as shown in figure 14a. To avoid this overshoot the actuators command is given linearly, from zero to its maximum, in a few hundreds of milliseconds, thus obtaining a smoother response (figure 14b).

The compensation procedure leads to satisfactory results and the vibration reduction, also in this case, spans between 60 and 90%.

The compensation is not constant but depends on factors such as working frequency, room temperature, heating of the actuators. In particular, for long running times of the actuator, its internal temperature may rise and its performances vary accordingly. This latter variation produces bands of repeatability similar to those obtained by varying the amplitude of the disturbance (figure 15) and can be related to the limits of the interpolating functions in the characterization phase.

7. CONCLUSIONS AND FUTURE PERSPECTIVES

The experimental analysis has shown the effectiveness of the proposed compensation strategy, which applies to those systems that meet the simplifying assumptions. However, it proved to be reliable, repeatable and easy to implement.

The compensation could also be achieved through approaches coupling material and device models (MDM), which require at least the characterization of the material, a demanding task compared to the characterization of the actuator in the system. Even the characteristics of the material are subject to tolerances and variations with temperature, exactly as it occurs for the behaviour of the actuator in the system. Furthermore, the MDM compensation is usually integrated in a control algorithm which depends on the configuration of the system. When the latter changes, the control needs to be reviewed. In the approach proposed in this paper it is sufficient to recalibrate the system and recalculate the compensation functions: a process that can easily be automated and quickly solved as a self tuning.

On the other hand, the proposed approach has the following drawback: replacing one of the actuators requires a new tuning of the system, while in the MDM compensation usually the component has only to comply with certain specifications. Notice however that, even in this case, a tuning of the system is required.

The presented procedure can be further developed. Future work should be devoted to various aspects, such as the experiments on the thick plates and the optimum positioning of the actuators and the compensation point. As regards the latter aspect we carried out a preliminary test on a rectangular plate, with the same position of the actuators and with both the excitation and compensation points laying outside the area bounded by the actuators: this first check shows that the compensation is promising although refinements are still required.

The self tuning has also to be ameliorated, to include, for example, the effects of the temperature of the actuators and the simultaneous presence of different harmonic components of the disturbance. The final goal should be the installation of the actuators on the actual workpiece carrier plate of a milling machine.

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