# Damping of in-process measuring system through variable stiffness tunable vibration absorber

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## Abstract
In external cylindrical grinding machines, measuring systems are occasionally mounted on a gantry type frame. The modes of this structure are potentially harmful within the operation range of the grinding wheel in a standard machine configuration, since the rotation of the wheel can enter into resonance, thus avoiding a correct determination of the on-line measurement. The resonance problem can be successfully dealt with the use of a variable stiffness vibration absorber which autonomously adapts its stiffness to tune according to the mode to be damped, increasing the dynamic stiffness along the whole operation range of the wheel.

In this work, a variable stiffness self-tunable vibration absorber prototype has been built and a new tuning function has been derived in order to minimize the response at the frequency coincident with the rotating speed of the wheel. Finally, validation tests have been performed in a scale supporting structure and the vibration reduction improvement comparison with respect to a standard fixed tuning strategy has been evaluated.

## 1 Introduction

Growing demands on dimensional requirements in the manufacturing industry makes in-process control a fundamental tool for dimensional quality assurance. Nowadays, machine tool builders are increasingly implementing in-process measuring systems integrated in the own structure of the machine, with the purpose of cost-efficiently manufacturing products of complex geometries requiring a high degree of accuracy. The on-line measuring system allows an accurate determination of the dimensions of the workpiece, discarding the need of a final precision verification in an external measuring machine. Moreover, as these products usually require long manufacturing times, a single machining mistake can result in a faulty workpiece, which in turn brings along an enormous economic loss. Thus, the on-line measurement avoids the error or permits a fast correction, before it becomes too late.

This integrated machining & measuring solution provides several benefits but it is also subject to the negative collateral effects coming from the machining process, such as vibration or thermal variations. As the available space inside the machine to locate the measuring system support structure is usually very limited, it often results in a rather flexible structure. Therefore, the sensitivity to vibrations is particularly compromising.

In grinding processes, typical vibrations from the wheel unbalance can be transmitted through the machine structure to the measuring probe, affecting a proper assessment of the measured dimension. Although grinding wheels are usually equipped with very effective wheel balance systems, there is always a slight remaining unbalance that, when the measurement structure is not stiff enough, could be capable of misleading the dimensional measure. This is especially critical when the grinding wheel is rotating in resonance (or close to the resonance) with some of the modes of the measuring system.

On the other hand, it must be taken into account that the wheel rotating speed changes as the wheel wears out, since the grinding cutting speed must be kept constant while the wheel diameter changes. Therefore, the wheel rotation speed varies from the rotation speed corresponding to the larger diameter (new wheel) to the smallest diameter (worn-out wheel). Thus, the excited frequency range is broad and the vibration becomes critical when the wheel rotating speed crosses a resonance of the measuring system.

Different measuring structure designs have been implemented in grinding machines for on-line measurement. The typical arrangement consists of an additional carriage supporting the measuring unit and moving in parallel to the workpiece axis. The measurement can be performed either by a laser/optical measuring system or by a touch probe. Due to the classical architecture of external cylindrical grinding machines, the addition of this measuring structure must be carried out between the workpiece an the door of the machine, which can be detrimental for the operator to access the workpiece, especially when loading and unloading the workpiece. It is for this reason that in external cylindrical grinding machines, measuring systems are alternatively located on a gantry type frame (see Figure 1).

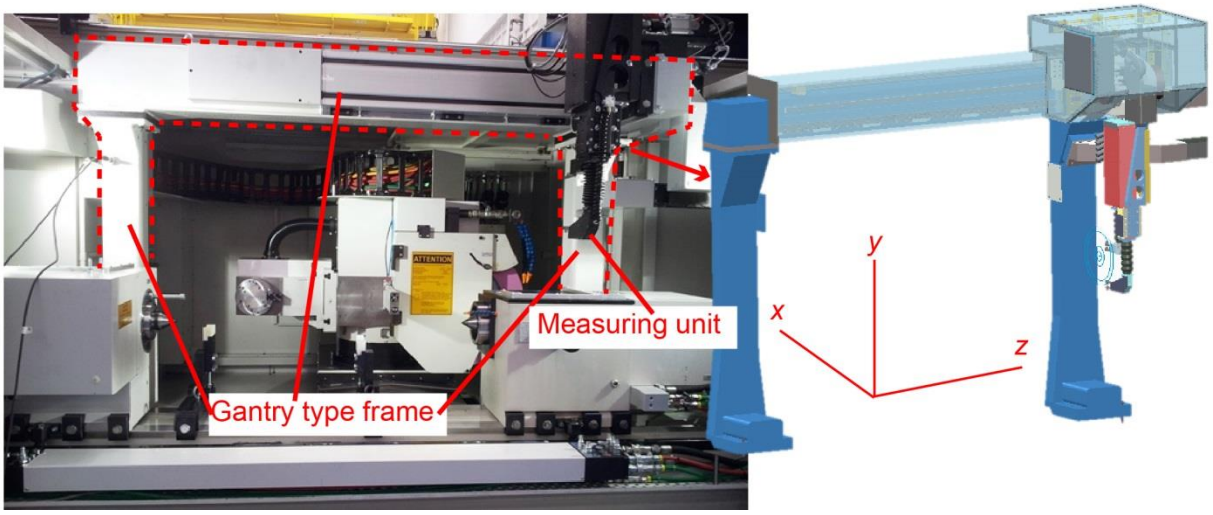



Figure 1: HG72 grinding machine with in-process measuring system

In this measuring system configuration, two bending modes and a rotation mode may be typically located within the working range of the grinding wheel.

## 2 Vibration mitigation strategies in measuring structures

There are different solutions to tackle vibrations on the measuring system. The most direct one would consist of redesigning the gantry-type structure using numerical models like **Finite Element Method (FEM)** models. The purpose would be to make it stiffer in order to shift the natural frequencies over the maximum excitation frequency (highest rotation speed of the wheel). However, this solution is many times unfeasible, due to the limited space to add new bracings and reinforcements to the gantry-type frame. The introduction of materials with high Young's modulus in zones where the strain energy of the modes is high can increase the stiffness of the critical mode without any increase in the required volume. Considering that steel and cast iron are the standard materials in machine structures, only costly materials like carbide and special metal alloys and composites can offer substantially larger Young's modulus [1], which makes the material variation not affordable for this kind of large gantry-type structures. Theoretically, the modifications to increase the stiffness could be combined with the reduction of weight in points with high critical modal displacement, with the purpose of increasing the natural frequencies of the system. However, the mass concentration at the critical locations is related to the own mass of the measurement device and, therefore, it is not easy to achieve important mass reductions. Moreover, the

frequency range limited by the wheel rotating frequency range can be very broad in some cases, which makes the FEM based design to move modes out of this range extremely complex.

The introduction of **flexible supports** in the base of the gantry structure to isolate the vibrations coming from the machine can be also regarded as a possible solution. However, the grinding wheel rotation frequencies can be relatively low, which requires the introduction of very flexible supports for an effective passive isolation. The presence of moving masses and gravity centers of the measuring system out of the gantry-type frame makes the application of these compliant isolators on the base of the gantry not suitable.

For this reason, solutions seeking to add damping to the system reducing the impact of the resonance are more suitable to solve this problem. **Active damping** is usually an effective solution for any kind of broadband vibration problem like self-exited vibrations [2,3]. However, active damping strategy requires the integration of sensors, actuators and a control system, which usually makes the solution too costly [4,5]. Thus, **passive damping** techniques, which are usually more affordable, are selected as more efficient solutions for this particular problem.

The introduction of **special materials, coatings and foams** with high internal damping has been proposed by several authors [6,7]. These materials should be placed at the locations where the modes have the highest strain energy. However, the main source of damping is at the interfaces and joints of the system, where damping is around one order higher than the structural damping of metallic parts [8]. This damping difference can be even higher in the case of low frequency modes. Some authors tried to increase the damping at joints by creating highly damped interfaces [9,10].

The introduction of passive auxiliary systems, such as **Tuned Mass Dampers (TMD)** [11], **friction** [12] and **impact** [13] **dampers**, is another alternative to increase the damping by means of different physical mechanisms. Tuned mass dampers have been used for vibration control in many civil and industrial applications such as buildings [14], wind turbines [15] and machine tools [11], among many others. A TMD is an inertial mass added to the system to damp via a linear spring and a dashpot. These elements are tuned to damp the critical mode of the original system. By matching the natural frequency of the highly damped TMD with the critical frequency of the system, both modes can be coupled to increase the damping. Consequently, the original mode is split into two modes with higher dynamic stiffness. The TMDs have to be tuned accurately to the targeted frequency, and their positive effect is limited to a certain frequency range [16, 17]. Therefore, this system could only damp one mode of the gantry-type structure of the measurement device inside the grinding wheel rotation range. In this aspect the impact and friction dampers have the advantage to damp more than one mode, but they are usually applied in high frequency applications and are difficult to design and control.

Some authors have tried to overcome the limitations of the TMD by implementing **tunable TMDs**. For instance, Slavicek and Bollinger [19] proposed a damper for which the natural frequency and the damping of the system can be varied in process. These semi-active dampers require some energy to be tuned but after that the system behaves like a passive damper. The energy consumption is low compared with pure active devices and a small battery is providing sufficient energy for that purpose [20]. The semi-active concepts have been proposed for chatter mitigation [18-21]. As chatter onset frequency strongly depends on the excited mode and the rotating speed, the capability of varying the stiffness of the tuned mass damper in order to tune the frequency at the desired value is highly useful in order to achieve an optimum performance [20, 21].

In this work, this tuning capability is applied for an effective vibration reduction in the measuring unit. Thus, instead of two or three different tuned mass dampers which damp each different mode, a variable stiffness vibration absorber which autonomously adapts its stiffness to tune according to the mode to be damped is used.

In order to tune the absorber to its optimum point, a tuning function has been derived in order to minimize the magnitude of the frequency response function (FRF) at the frequency coincident with the rotating speed of the wheel. Thus, an improved performance with respect to a fixed standard tuning criterion is achieved.

Finally, validation tests have been performed in a scale supporting structure and a real machine and the vibration reduction improvement comparison with respect to a standard fixed tuning strategy is evaluated.

# 3 Optimal tuning of the vibration absorber

In order to tune the absorber to its optimum point, an investigation about the optimum tuning has been carried out. As mentioned earlier, in an external cylindrical grinding machine, the main excitation is produced by the unbalance of the wheel, which has a frequency coincident with its rotation frequency. Therefore, the objective is to derive a tuning function that minimizes the dynamic response at the frequency coincident with the rotating speed of the wheel.

The advantage of the variable stiffness tunable absorber is that its stiffness and, therefore, its frequency can be varied along a specific range. Thus, an improved performance with respect to a fixed standard tuning criterion is achieved.

The vibration absorber must be tuned so that the lowest response is obtained in the measuring unit at each wheel rotating speed. Computing the achieved response reduction at every rotating speed, a pseudo FRF which describes the dynamic response at each individual frequency line can be obtained. This pseudo FRF is only valid to assess pure sinusoidal excitation.

Previously, Den Hartog [16] defined an optimal tuning for a tuned mass damper with constant stiffness in order to minimize the amplitude and the real part respectively. In this case, as the tuning frequency will be modified for different wheel speeds, a new derivation will be sought.

A set of dimensionless parameters have been defined in order to simplify the subsequent equations:

| Parameter | Definition |
|---|---|
| Mass ratio | $\mu = \frac{m_2}{m_1}$ |
| Tuning | $f = \frac{f_2}{f_1}$ |
| Dimensionless frequency | $g = \frac{\omega}{\omega_1}$ |
| Damping ratio of the original system | $\xi_1 = \frac{c_1}{2m_1\omega_1}$ |
| Damping ratio of the vibration absorber with respect to the critical damping by Dan Hartog [16] | $\chi_2 = \frac{c_2}{2m_2\omega_1}$ |

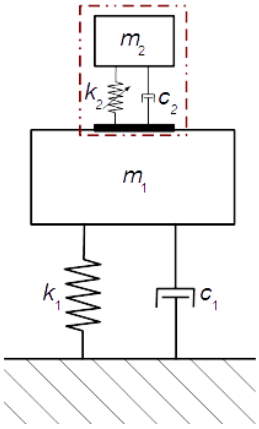



Table 1: Dimensionless parameters

The magnitude of the frequency response function of the system with the added vibration absorber can be expressed as [16]:

$$|H(g)| = \frac{1}{k_1}|M(g)|, \tag{1}$$

where $k_1$ is the modal stiffness of the original system and $|M(g)|$ is the magnification function defined as:

$$|M(g)| = \sqrt{\frac{(f^2 - g^2)^2 + 4g^2\chi_2^2}{4g^2(f^2\xi_1 + \chi_2 - g^2(\xi_1 + \chi_2 + \mu\chi_2))^2 + (f^2(-1 + g^2(1+\mu)) + g(g - g^3 + 4g\xi_1\chi_2))^2}}. \tag{2}$$

The optimum tuning is achieved with the minimization of $|M(g)|$. Equation (2) is derived with respect to $f$ and the following solution is obtained:

$$f_{opt}(g) = \sqrt{\frac{g(g^3(2+\mu) - 2g(1+4\xi_1\chi_2) + \sqrt{g^6\mu^2 + 16g^4\mu\xi_1\chi_2 + 16((-1+g^2(1+\mu))^2 + 4g^2\xi_1^2)\chi_2^2})}{2(g^2(1+\mu) - 1)}}. \tag{2}$$

If the damping of the original system is equal to zero $c_1$=0, the magnification function yields:

$$|M(f,g)| = \sqrt{\frac{(f^2-g^2)^2+4g^2\chi_2^2}{(g^2-g^4+f^2(-1+g^2(1+\mu)))^2+4g^2(-1+g^2(1+\mu))^2\chi_2^2}}\,. \tag{3}$$

For both cases $c_1$=0 and $c_1$≠0, if the magnification function $|M(g)|$ is derived with respect to $\chi_2$, it turns out that the minimization of the function is achieved when the damping of the added tuned mass damper $c_2$ tends toward 0. Therefore, in order to achieve the lowest possible response, the vibration absorber will need to have the lowest possible damping. This is the case of a vibration absorber, which is similar to a tuned mass damper but without adding damping to the moving mass ($c_2$=0).

From equation (3), it is possible to derive that, when $c_2$=0, the optimal tuning is achieved when $f$=$g$, where the magnification function would be $|M(g)|$ = 0. This is not possible to achieve in practice due to the presence of damping in any mechanical system, but it will produce the lowest response.

$$f_{opt} \cong g\,. \tag{4}$$

Summarizing the optimum tuning of the vibration absorber for a measuring system in an external cylindrical grinding machine is achieved when $c_2$=0 and $f$=$g$. This tuning is only appropriate for a pure single frequency sinusoidal excitation. If a broader excitation was present, this strategy would not be suitable.

Next, the comparison of a one critical mode system tuned according to the developed optimal tuning $f_{opt}$, its approximation $f$=$g$ and an optimal fixed tuning are presented through a theoretical simulation.

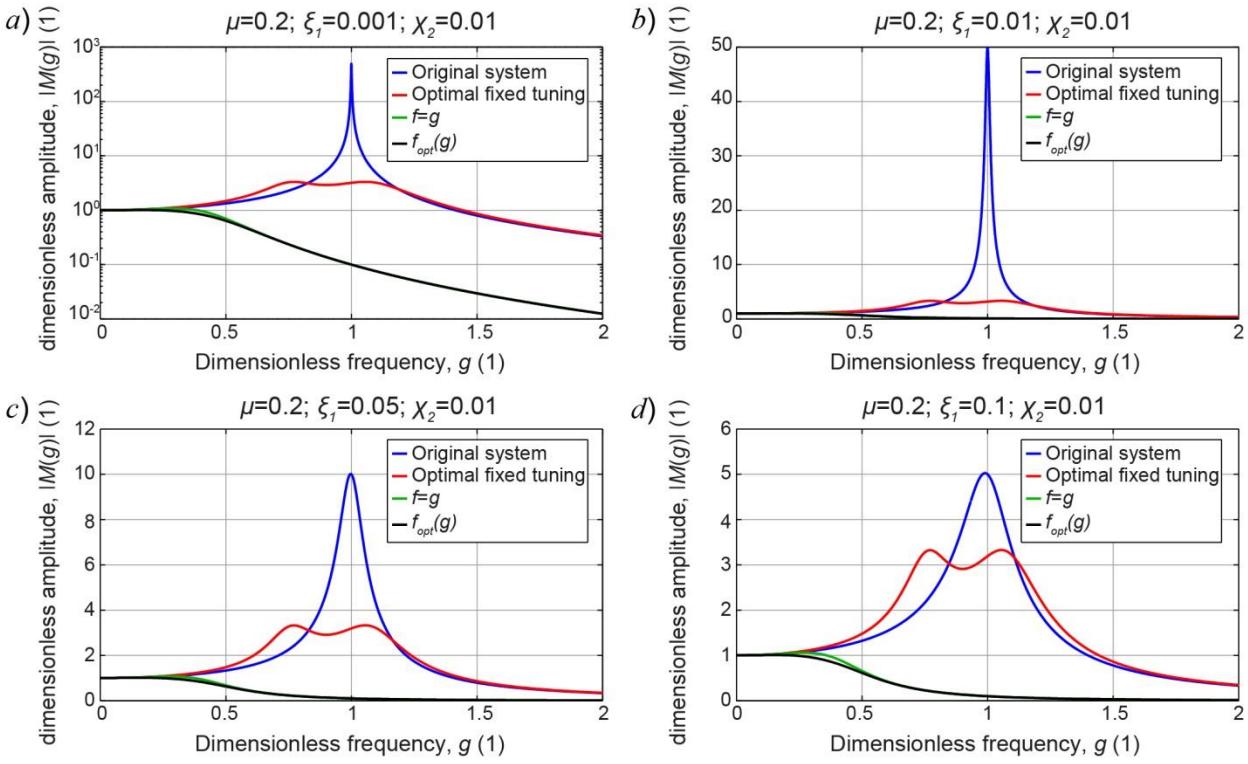



Figure 2: Comparison of tuning by optimal fixed tuning or through optimal variable stiffness vibration absorber. *a*) $\xi_1$=0.001, *b*) $\xi_1$=0.01, *c*) $\xi_1$=0.05, *d*) $\xi_1$=0.1

As it can be observed in Figure 2, although the fixed tuning decreases significantly the response of the system around the critical mode, the variable tuning through the vibration absorber improves the behavior to a much higher extent. It is also noticeable that the "simplified tuning" $f_{opt}$≅$g$ is virtually as good as the exact tuning. Therefore, this approximate optimum tuning will be applied henceforth.

The variable stiffness vibration absorber is even more advantageous than an optimal fixed tuning when more than one critical mode is present. In order to assess the improvement achieved in systems with more

than one dominant mode, a two-mode system with the characteristics described in Table 2 has been analyzed:

| | Dimensionless paramters | | Modal parameters | | | |
|---|---|---|---|---|---|---|
| | ***g*** | ***ξ*** | **Frequency *f*** | **Mass *m*** | **Stiffness *k*** | **Damping *c*** |
| Mode I | 1 | 0.01 | $f_1$ | $2m_1$ | $2k_1$ | $c_1$ |
| Mode II | 1.73 | 0.0173 | $1.73f_1$ | $2m_1$ | $6k_1$ | $3c_1$ |

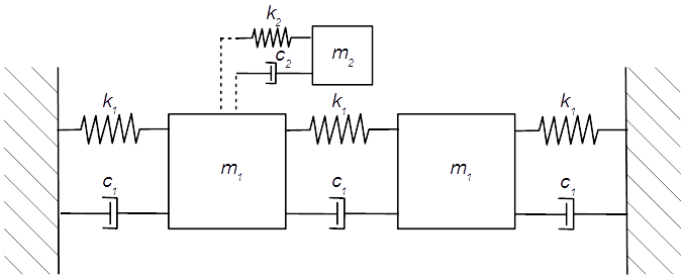



Table 2: Dynamic parameters for the simulation

The tuning parameters used for this simulation are indicated in Table 3:

| **Parameter** | **Fixed tuning at Mode I** | **Fixed tuning at Mode II** | **Variable *f=g* tuning** |
|---|---|---|---|
| Mass ratio ($\mu$) | 0.2 | 0.2 | 0.2 |
| Damping of the tuned mass damper ($\chi_2$) | Optimal (0.208) | Optimal (0.36) | 0.01 |
| Tuning ($f$) | 0.8 | 1.63 | $f=g$ |

Table 3: Tuning characteristics

In Figure 3, the two critical mode system is compared when damped through optimal fixed tuning or the optimum tuning developed for the variable stiffness vibration absorber.

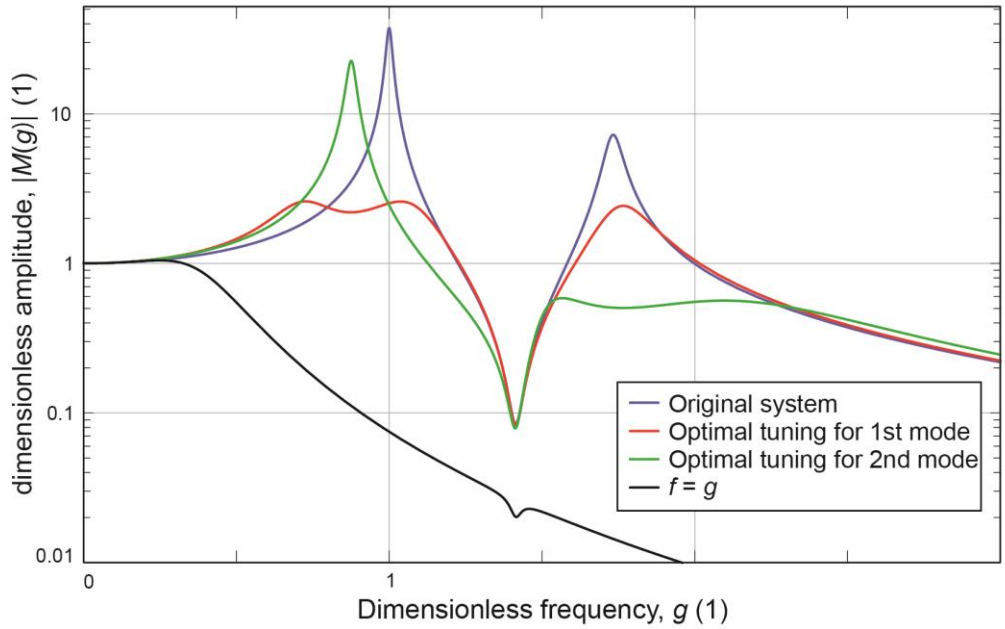



Figure 3: Comparison of optimal fixed tuning or through optimal variable stiffness vibration absorber in a two mode system.

Similarly to the one dominant mode case, the tuning trough the variable stiffness tunable vibration absorber outperforms the optimal fixed tuning.

# 4 Design of the vibration absorber

In this section, the variable stiffness DVA concept and tuning strategy developed in Section 3 will be implemented in a physical prototype capable of performing in practice the theoretical vibration reduction envisaged.

A vibration absorber prototype has been built. It consists of a unidirectional moving mass supported by a rotary spring located in the center of this moving mass. This rotary element, which is driven by an electric motor, is built according to a special shape that provides a variable stiffness as a function of its angular position (see Figure 4*a*). When the rotation angle is 0º, the rotary spring contributes to the system with its minimum stiffness, whereas when it is rotated at 90º, it provides the highest stiffness. An encoder coupled to the tuning motor controls the proper angular position in accordance with the rotating speed of the wheel. The design of the rotary spring must be fitted to cover the wheel rotating frequency range.

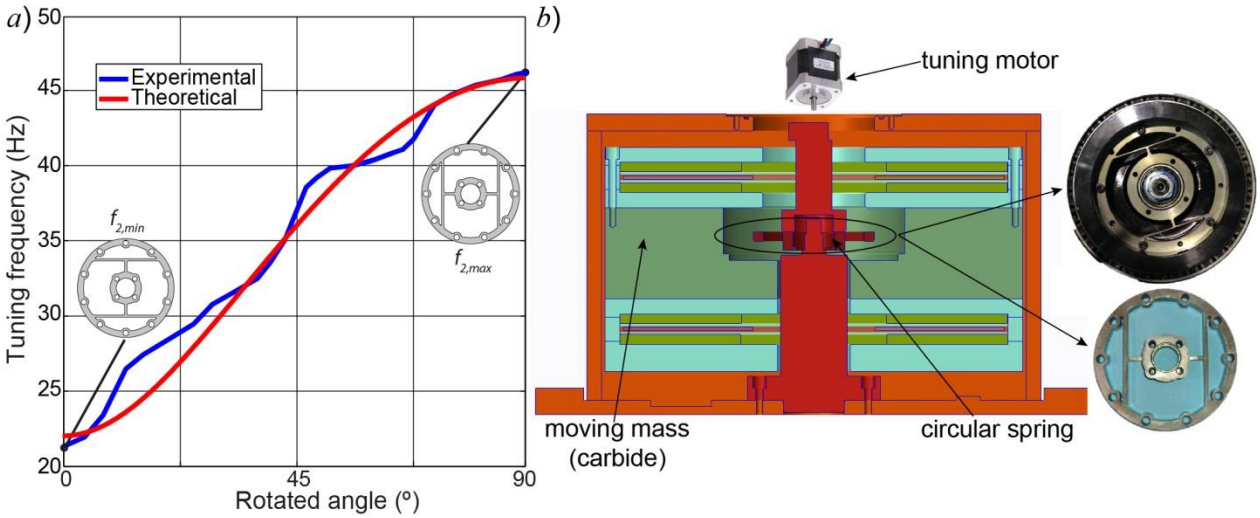



Figure 4: *a*) Variation of the tuning frequency with the rotation of the rotary spring. *b*) Sketch of variable stiffness vibration absorber. Red shades represent the fixed parts whereas green shapes represent the moving mass

Figure 4*b* presents a sketch of the design of the vibration absorber. It is located on top of the measuring unit support, oriented at 45º in the *xz* plane, in such a way that it is capable of acting in both directions with its unidirectional movement. The movement is guided through three flexural plates on the bottom of the moving mass. This assures a low damping of the DVA, which is the best scenario according to the findings in Section 3.

The damper has a weight of 37kg with a damping ratio ranging from 0.24 to 0.09, depending on the tuning frequency. The tuning frequency ranges from 22Hz to 46Hz and varies according to the stiffness variation of the rotary spring, which fulfils the following equation:

$$k = k_{\min} \cos^2 \varphi + k_{\max} \sin^2 \varphi \,, \tag{5}$$

where $\varphi$ is the rotation of the rotary spring from 0º to 90º.

The dynamic vibration absorber (DVA) is controlled through the CNC of the machine, which commands the rotary spring according to the wheel rotating speed through a tuning motor on real time. Therefore the developed optimal tuning $f_{opt}=g$ is achieved.

# 5 Experimental validation

## 5.1 Experimental setup

In order to validate the developed system, a scale prototype of the measuring unit mounted on the gantry type frame has been set up (see Figure 5). The system has been downsized, maintaining the proportion of the original system. The dynamic vibration absorber has been located on top of the gantry structure, at a corner, in the equivalent position where it will be later tested on the real structure.

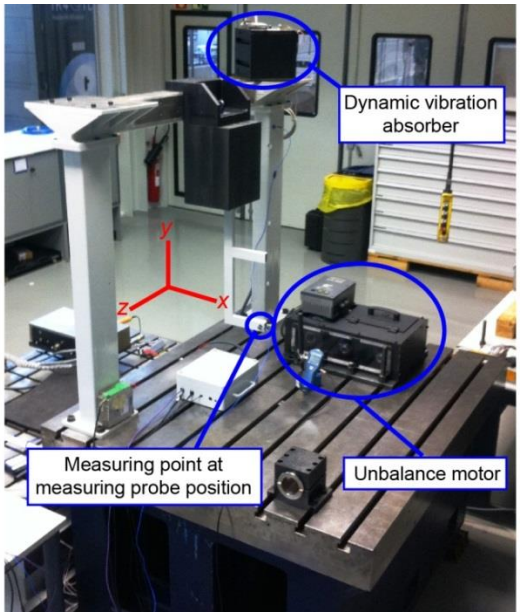



Figure 5: Measuring unit scale prototype

The DVA location and its orientation at 45º allows the operation against the main modes of the system.

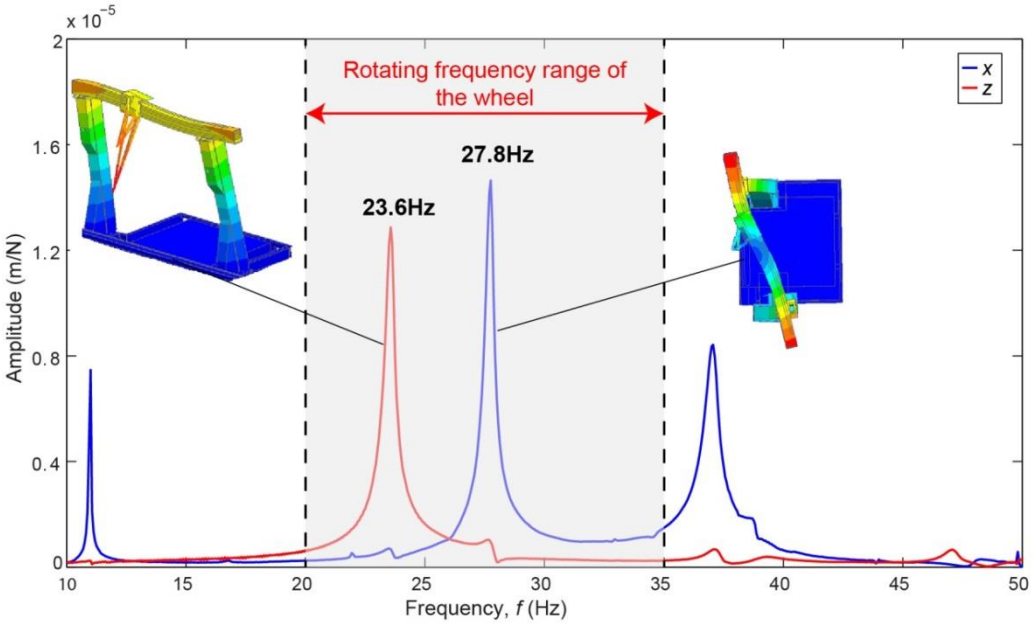



Figure 6: Modes of the measuring system located on the wheel rotating frequency range

According to Figure 6, there are two modes in the frequency range of the grinding wheel. The first one is a bending mode in *z* direction at 23.6Hz, whereas the second one is a torsion mode in the *xy* plane at 27.8Hz.

The excitation has been introduced through an unbalance motor located on the table where the gantry type structure is clamped, simulating the unbalance coming from a wheel in a real machine. A 56000gmm unbalance has been applied in order to achieve a high sensitivity in the performed tests, which turned out in a centrifugal force range of roughly 200-1000N, depending on the rotation speed. The vibration response has been measured at the location of the measuring probe, which is the point whose vibration directly affects the dimensional measurement.

The modal parameters of the original mode and the added dynamic vibration absorber are summarized in the table below:

| Parameter | Mode x | Mode z |
|---|---|---|
| Natural frequency | 27.8Hz | 23.6Hz |
| Modal mass | 194kg | 233kg |
| Damping ratio | 0.0058 | 0.0076 |
| Mass ratio | 0.19 | 0.16 |

Table 4: Modal parameters of the measuring system

## 5.2 Results

Figure 7 shows the response achieved when the dynamic vibration absorber is tuned at the frequency of these main modes, compared with an optimal fixed tuning for that mode:

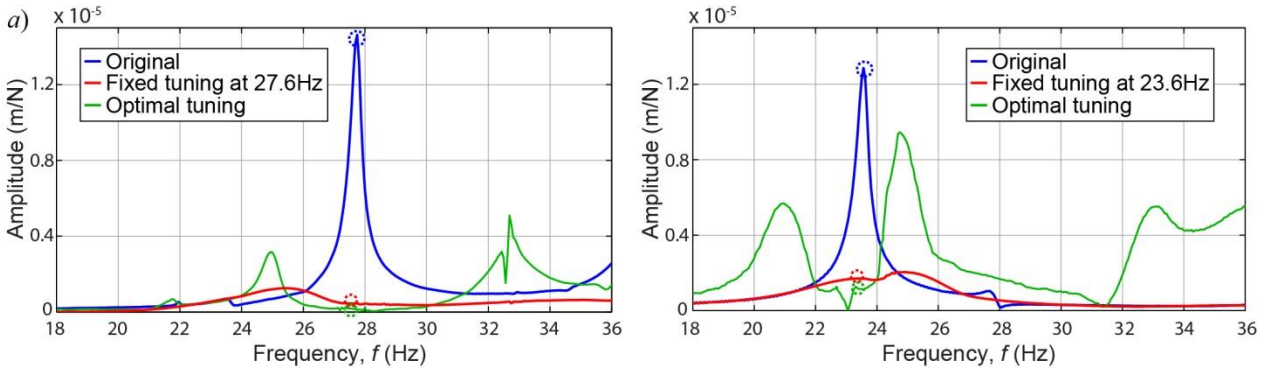



Figure 7: *a*) shows the comparison between the detuned and tuned (27.8Hz) frequency response functions in *x* direction; *b*) shows the comparison between the detuned and tuned (23.6Hz) frequency response functions in *z* direction

According to the figure, the improvement at the resonant frequency is remarkable with the fixed tuning with optimal damping and especially with the optimal tuning with no damping, achieving reductions of dynamic response of up to 97% and more than 99% respectively.

The scale measuring unit has been validated under different conditions:

a) DVA detuned (constantly tuned at 46.5Hz)
b) DVA tuned for the x mode with optimal damping at a fixed frequency of 23Hz
c) DVA tuned for the y mode with optimal damping at a fixed frequency of 27Hz
d) Self-tunable DVA tuned as *f=g* with minimum damping.

The purpose is to compare the strategy of tuning the DVA at a fixed frequency according to a single mode, which can be a sufficiently good approach when a single mode is affecting the operation range but can be highly unsuitable for multiple mode structures.

Figure 8 shows the vibration result at different wheel rotation speeds (different frequencies) for the four previously defined DVA states.

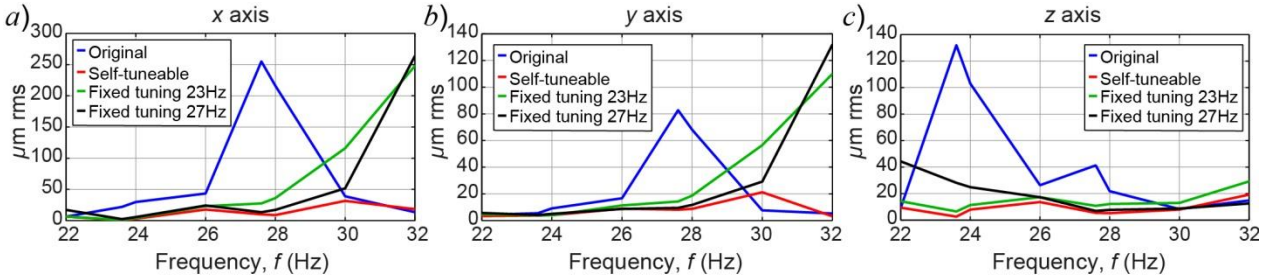



Figure 8: Vibration amplitude in the measuring probe location of the scale measuring system, *a*) in *x* axis, *b*) in *y* axis and *c*) in *z* axis

When the DVA is not tuned, the vibration is magnified at the natural frequency regions (23Hz in *z* axis and 27Hz in *x* and *y* axes). The fixed tuning at 23Hz gives rise to amplification at higher frequencies, which can even result in higher amplitude than the original vibration value. The fixed tuning at 27Hz also leads to high vibration values below and over that particular frequency. The lowest vibration values are achieved with the self-tunable strategy. Therefore, the self-tunable system has proven its improved behavior with respect to traditional fixed damping systems, especially for multiple mode measuring structures. The vibration at the tip of the measuring probe is reduced more than 90% in every coordinate axis.

# 6 Conclusions

A tunable dynamic vibration absorber has been developed to minimize vibration in gantry type in-process measuring systems for external cylindrical grinding machines. The best tuning strategy has been defined, which consists of equating the wheel rotating frequency in order to minimize the amplification of the wheel unbalance vibration. This tuning is only suitable for pure single frequency sinusoidal excitation cases. The system is driven by a tuning motor which adds a self-tuning capability to minimize the amplification of the transmitted vibration throughout the whole wheel rotating speed range. The advantages of the system with respect to a conventional fixed tuning system have been demonstrated through theoretical simulations and experimental tests.

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