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# Control laws for chatter suppression in milling using an inertial actuator

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## Abstract

In this paper an inertial actuator is used to design an active damper for chatter suppression in a milling process. Researchers usually use the Direct Velocity Feedback (DVF) technique to design the control law for the actuator. This DVF technique is widely used for its simplicity and leads to very good results. However, it can easily excite the structural modes of the actuator and make unstable the closed-loop system. A Virtual Passive Absorber (VPA) control law is proposed to avoid this drawback. This kind of controllers are less likely to excite the actuator since the control effort is focused on the problematic mode of the structure. The parameters of the VPA controller are chosen to reduce the real part of the frequency response function (FRF) of the structure, which is well known to be important to determine the stability region. Several practical experiments are performed to show the usefulness of the control laws comparing the performance of both of them. The VPA control is tested in a pair of practical milling processes as well.

**keywords:** active damping, chatter suppression, inertial actuator, milling.

## 1 Introduction

Self-excited or chatter vibrations are well known among milling machine manufacturers and users. Nowadays, chatter is one of the main limitations of milling processes. Machine tool chatter vibrations cause an unacceptable surface finish on the workpiece reducing the life of the tool and the mechanical components of the machine.

Chatter stability margin can be increased by adding a tuned vibration absorber to the structure [14,15]. However, a passive absorber is not feasible in many machining processes where the dynamics of the system change according to the operation point and an active damper is needed [2,3,6]. The use of an inertial actuator is a possible solution. An inertial actuator produces a reaction force on the host structure according to a designed control law.

Inertial actuators have been widely used to reduce vibrations in machining operations. Researchers usually use the well-known Direct Velocity Feedback (DVF) strategy in order to design the control law [5,8,13]. This technique has been mainly used due to its simplicity and leads to very good results, avoiding the appearance of chatter in many practical cases. However, some authors have proposed alternative control laws, which may lead to better results. One of these possible control laws is the Virtual Passive Absorber (VPA) control [8,9,12]. This control law is designed to make the inertial actuator behave as a typical passive damper. The main disadvantage of passive absorbers is that they are limited to reduce a unique

mode of the system. However, a VPA control can be retuned to avoid a different mode and it can be designed to have larger mass ratios. In this paper we propose a VPA controller to increase the chatter stability margin in machining operations using an inertial actuator.

Despite its simplicity the DVF control requires some study in order to find the optimum value of the feedback gain. This is usually calculated from a root locus analysis if a model of the system is available. In most cases this value is found experimentally increasing its value until the closed-loop system is unstable. This study is necessary because a large value of the feedback gain excites the structural modes of the mass damper while a small value of it could lead to a not good enough reduction of the vibrations in the structure.

On the other hand, VPA control is more straightforward and it only requires a modal analysis of the system to be controlled. Many authors have proposed different ways of finding the optimum values of the parameters of the controller [4,11,13]. In this paper we have used the tuning methodology proposed by N. D. Sims in [16]. Passive absorbers are typically tuned to reduce the magnitude of the frequency response function near a problematic mode. However, Sims proposed an alternative tuning methodology focused on decreasing the magnitude of the real part of it. This fact is really useful in machining operations, because it is well known that the chatter stability margin is given by the real part of the frequency response of the system. In [16] the tuning method is presented as a new solution, which is easier to apply than other numerical and graphical approaches. The main novelty of this paper is to use these tuning equations and integrate them with the VPA to build an active damper for chatter suppression and provide an alternative control algorithm for an inertial actuator that leads to a larger stability range.

Another advantage of this control law is that an optimal tuning does not excite the structural modes of the actuator and guarantees the closed-loop stability without further analysis in the case of an structure with a unique mode. Structures with more than one mode require a more detailed study and probably a more complex control law. Some authors propose the use of a combination of different VPA controllers to manage this situation, where each of this controllers is tuned to avoid a different mode of the system.

We have tested both control laws mounting the actuator on different structures and found that the VPA control provides a larger chatter stability than the DVF control. Moreover, an experimental tuning was always required with the DVF control, while the parameters of the VPA controller were calculated directly from a modal analysis and the tuning equations described in [16], which results to be faster. In this paper a comparison between both controls laws is presented emphasizing the advantages and disadvantages of each one of them.

This paper is organized as follows. First, the inertial actuator is described and a model of it is developed. The different control laws are then explained. In the next section the experimental results are presented. In this section a couple of milling test are performed using the VPA control. Finally, conclusions end the paper.

## 2 Inertial actuator

An inertial actuator can be described as a reaction mass  $m_a$  supported on a spring  $k_a$  and a damper  $c_a$  attached to a base (Fig.1.a). According to an applied voltage  $V_{in}$  the reaction mass is excited and it induces a force  $F_a$  on the supporting base [13]. The dynamics of an inertial actuator can be modeled as the following transfer function

$$\frac{F_a(s)}{V_{in}(s)} = \frac{G \cdot m_a s^2}{m_a s^2 + c_a s + k_a} = g_a \frac{s^2}{s^2 + 2 \zeta_a \omega_a s + \omega_a^2} \quad (1)$$

where  $\omega_a$  is the natural frequency,  $\zeta_a$  the damping ratio and  $g_a$  is the gain of the actuator. The actuator behaves as an ideal lineal force generator beyond a determined frequency  $\omega_p$  (Fig.1.b). Another upper limitation in the bandwidth is included by the electromagnetic circuit. This means that the actuator can only be used as a force generator in this linear range. The task is to design an appropriate control law to the actuator in order to control structure vibrations.

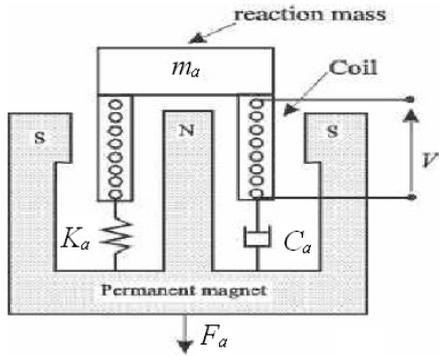


Fig.1.a: Inertial actuator

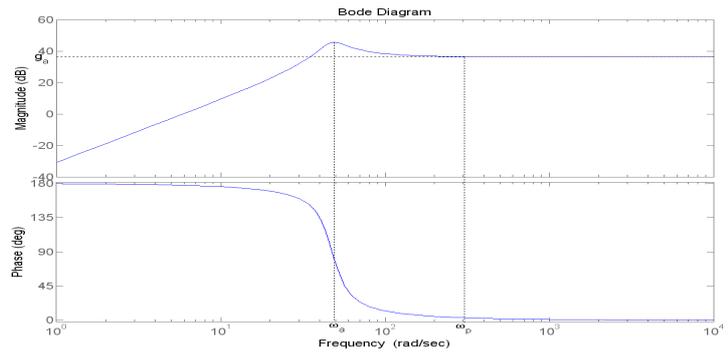


Fig.1.b: Bode plot of an ideal inertial actuator

In this paper the actuator used to perform the practical applications is a model of Micromega Dynamics company [18]. This actuator is capable of applying an up to 900 N force on the host structure. The actuator is mounted on a Kistler dynamometer plate and excited by a chirp voltage signal with the aim of calculating its dynamics. Fig. 2 shows the FRF (Frequency Response Function) between the measured force and the applied Voltage.

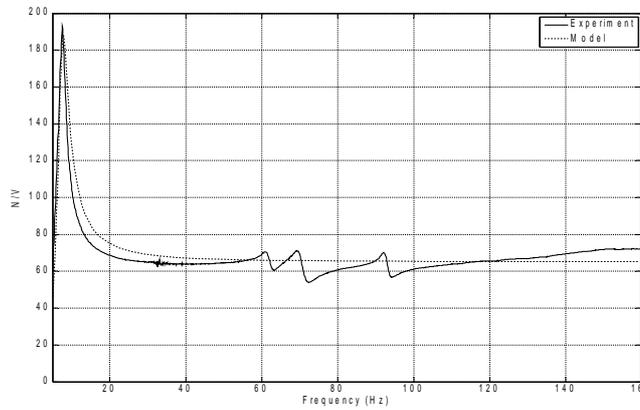


Fig. 2: Experimental and modeled FRF of the inertial actuator

In Fig.2 we can see that the actuator possesses an oscillation mode at low frequencies (around 8 Hz). This fact limits not only the bandwidth but also should be taken into account for stability consideration. This will be explained in detail in next section. The upper limitation of the linear range is found to be around 200 Hz. The actuator is modeled as in equation (1) by the following transfer function

$$\frac{F_a(s)}{V_{in}(s)} = \frac{65s^2}{s^2 + 16.76s + 2280} \frac{N}{V} \quad (2)$$

The identification is performed using a parameter estimator based on a least squares algorithm [6].

### 3 Control laws

In this section the two different control laws considered in this paper will be explained in detail. First, the typical Direct Velocity Feedback control law is explained. Nowadays, most of the inertial actuators used for vibration control include this control law. This is due to its simplicity and good results. However, this type of controllers has some drawbacks which should be considered to guarantee the closed stability and sometimes they could limit the performance of the controller. The use of a Virtual Passive Absorber controller is proposed to avoid some of them. VPA control law is explained in the second part of this section. Finally, the advantages and disadvantages of both control laws are compared.

### 3.1 Direct Velocity Feedback

The main idea of this control law consists in a negative feedback of the velocity of the structure. The transfer function of the controller can be written as

$$V_c(s) = -g_v \dot{X}(s) = -g_v \cdot s \cdot X(s) \quad (3)$$

where  $V_c(s)$  is the applied control input to the inertial actuator,  $sX(s)$  is the velocity on the structure and  $g_v \in R^+$  is the control gain. Considering that the actuator is working in the linear range, then the transfer function between the applied force on the structure and the control voltage input can be written as

$$F_a(s) = g_a \cdot V_c(s) = -g_a \cdot g_v \cdot s \cdot X(s) \quad (4)$$

If the structure is modeled as a linear time invariant SISO system, the characteristic equation can be calculated as follows

$$1 + g_v \cdot g_a \cdot s \cdot G(s) = 0 \quad (5)$$

where  $G(s)$  is the transfer function describing the structure to be controlled. Now if we consider a structure with a unique mode, equation (5) can be written as

$$1 + g_v \cdot g_a \cdot s \cdot \frac{g}{s^2 + 2\omega_n \zeta_n s + \omega_n^2} = 0 \rightarrow s^2 + s(g_v \cdot g_a \cdot g + 2\omega_n \zeta_n) + \omega_n^2 = 0 \quad (6)$$

The damping can be increased in equation (6) increasing the value of the feedback gain  $g_v$ . The effect of the inertial actuator can be studied including its transfer function in previous characteristic equation

$$1 + g_v \cdot s \cdot \frac{g_a s^2}{s^2 + 2\zeta_a \omega_a s + \omega_a^2} G(s) = 0 \quad (7)$$

In this case the low frequencies dynamics of the actuator should be considered to guarantee closed-loop stability. In Fig.3 a root locus analysis is performed for a unique mode structure including the actuator dynamics. It shows that the poles of the structure are damped increasing the feedback gain. However, the poles of the actuator become unstable for large values of  $g_v$ . This means that for sufficiently large values of  $g_v$  the close loop is unstable. The largest value of this gain that keeps the system stable could be calculated if we had a linear model of the structure. However, this is not always possible and the optimal value of the feedback gain  $g_v$  is usually calculated experimentally. Initially it is chosen to be small and it is increased until the closed-loop system becomes unstable.

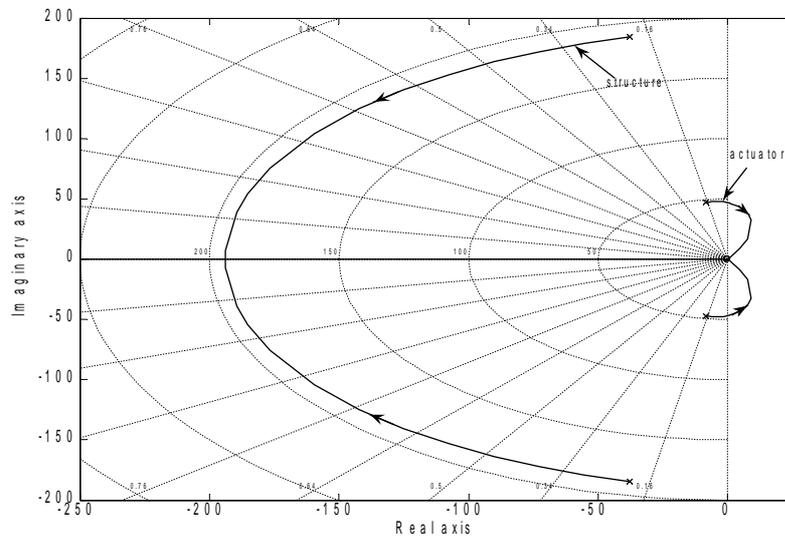


Fig.3: Root locus analysis including the dynamics of the actuator

### 3.2 Virtual Passive Absorber

The second control law is designed to make the actuator behave as a typical passive vibration absorber. A passive absorber is usually used to attenuate any of the modes of a structure and it is tuned to work in that range. The state space equations of the controller can be written in the following way [8]

$$\begin{cases} \begin{bmatrix} \dot{x}_c \\ \ddot{x}_c \end{bmatrix} = \begin{bmatrix} 0 & 1 \\ -\omega_c^2 & -2\zeta_c\omega_c \end{bmatrix} \cdot \begin{bmatrix} x_c \\ \dot{x}_c \end{bmatrix} + \begin{bmatrix} 0 \\ 1 \end{bmatrix} \cdot y \\ u = y_c = M\mu_c g_a^{-1} \begin{bmatrix} -\omega_c^2 & -\zeta_c\omega_c \end{bmatrix} \cdot \begin{bmatrix} x_c \\ \dot{x}_c \end{bmatrix} \end{cases} \quad (8)$$

where  $y = \ddot{x}$  is the acceleration on the structure,  $\mu = \frac{m_c}{M}$  is the mass ratio between the controller mass  $m_c$  and the system mass  $M$ .  $\omega_c$  and  $\zeta_c$  are the natural frequency and the damping ratio of the controller, respectively. In this case the measured acceleration is used as an input to the feedback controller. Then the controller output feeds the actuator. From (8) the transfer function of the VPA controller can be calculated as

$$G_c(s) = \frac{M\mu_c(2\zeta_c\omega_c s + \omega_c^2)}{g_a(s^2 + 2\zeta_c\omega_c s + \omega_c^2)} \quad (9)$$

The characteristic equation of the closed-loop system for a linear time-invariant SISO system including the dynamics of the actuator can be calculated as

$$1 + \frac{g_a s^2}{(s^2 + 2\zeta_a\omega_a s + \omega_a^2)} \cdot \frac{M\mu_c s^2(2\zeta_c\omega_c s + \omega_c^2)}{g_a(s^2 + 2\zeta_c\omega_c s + \omega_c^2)} \cdot G(s) = 0 \quad (10)$$

The optimal values of  $\omega_c$  y  $\zeta_c$  can be directly calculated from passive absorbers optimization methods. These methods just require a model analysis of the host structure. If these parameters are not appropriately tuned a resulting passive absorber may still be stable, but the closed loop behavior may not be good enough. On the other hand, an inertial actuator with this kind of control law can be unstable if the parameters of the controller are not appropriately tuned due to the low frequency dynamics of the actuator. The most used techniques for absorber tuning can be found in [3,10,12]. In most of them the parameters of the absorber are chosen such that the magnitude of the frequency response is reduced around the problematic mode. The aim of this paper is to design a control law to reduce the chatter in a milling process. It is well known that the stability range is given by the real part of the FRF instead of the magnitude. Using a typical turning model for the chatter process, the stability limit can be calculated using Nyquist Criterion [1]

$$K_s b_{lim} u G(j\omega) (1 - e^{-j\omega T}) = -1 \quad (11)$$

where  $G(s)$  is the tool transfer function between the cutting force and the displacement on the surface. Any value of the depth cut  $b > b_{lim}$  makes the system unstable. The stability condition can be written as

$$b_{lim} = -1 / (2K_s \text{Re}[uG(j\omega_c)]) \quad (12)$$

where  $\omega_c$  is the chatter frequency. According to equation (11) the stability is given by the real part of  $G(\omega)$ . Since  $b_{lim} \in R^+$  only the frequency range that makes  $\text{Re}(uG(j\omega))$  positive should be taken into account.  $u$  is used to denote the orientation coefficient and it is positive in an upmilling cutting process and negative in a downmilling process. This means that for a downmilling process the negative part of the FRF is considered to calculate the stability condition while for an upmilling process the positive real part of it is considered. Figure 4 shows a graphical representation of the relation between the real part and the resulting stability lobe for a downmilling process.

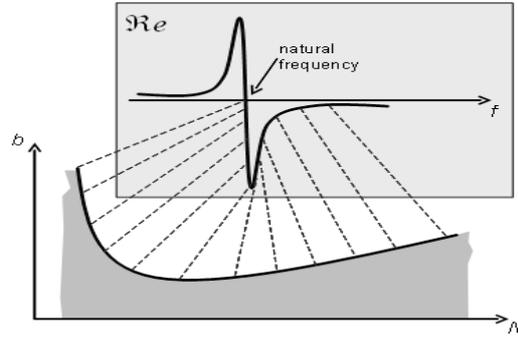


Fig.4: Relation between the real part and the stability lobe in a downmilling milling process.

From equation (12) it is clear that reducing the positive real part in an upmilling process or the negative part in a downmilling the chatter stability is increased. In this point one could think that the effort of the controller could be focused on this fact instead of reducing the magnitude. This kind of tuning for passive absorbers has been considered by many authors before, but in [16] Sims presents a new analytical solution for this problem. In this paper we use those results to tune the parameters of the VPA controller. The effect of the controller is that two equal peaks or troughs appear on the real part of the FRF around the tuned mode. The equations for the optimal parameters of the controller defined in (8) are obtained from the following equations

$$f_n^2 = \left( \frac{\omega_c}{\omega_n} \right)^2 = \frac{\mu + 2 + \sqrt{2\mu + \mu^2}}{2(1 + \mu)^2} \quad (13)$$

$$f_p^2 = \left( \frac{\omega_c}{\omega_n} \right)^2 = \frac{\mu + 2 - \sqrt{2\mu + \mu^2}}{2(1 + \mu)^2} \quad (14)$$

$$\zeta = \sqrt{\frac{3\mu}{8(1 + \mu)}} \quad (15)$$

where the subscript  $n$  denotes a downmilling milling process, while the subscript  $p$  is used for an upmilling one and  $\omega_n$  is the structural frequency to be damped. In case there is more than one problematic mode, a possible solution is to use a set of different VPA controllers in parallel. Each of these controllers is tuned to attenuate a different mode of the structure. The resulting controller can be calculated as

$$G_c(s) = \sum_{i=1}^N \frac{M^{(i)} \mu_c^{(i)} (2\zeta_c^{(i)} \omega_c^{(i)} s + \omega_c^{(i)2})}{g_a (s^2 + 2\zeta_c^{(i)} \omega_c^{(i)} s + \omega_c^{(i)2})} \quad (16)$$

where  $N$  is the number of mode to attenuate. In this point we should remark that the tuning process is not as easy as in a unique mode case. A VPA controller excites the structural modes at lower frequencies. For this reason the tuning requires an experimental tuning, beginning with the mode of larger frequency which is not going to be affected for any other controller and continuing until the lower one is tuned. The experimental tuning is started from the theoretical values in equations (13)-(15) and then they are slightly changed until two peaks or trough are observed in the real part of the FRF.

### 3.3 A comparison between both control laws

In this section the control laws are compared remarking the main advantages and disadvantages of both of them.

The main advantage of DVF control is its simplicity, since only a parameter has to be tuned. Despite this it leads to very good damping results. This fact makes this control law the most used one among researchers. Moreover, if there is not any kind of information about the system to be damped the experimental tuning is easier than in the VPA control, where a modal analysis is always required. On the other hand, with DVF finding theoretically the optimal value of the feedback gain is more difficult if a linear model of the

system is not available. The VPA control just requires a modal analysis of the structure, which is easier to perform than finding a linear model. When a DVF is used a high-pass filter should be added in the control loop in order to minimize the effects of low frequency dynamics of the actuator. In addition, the closed-loop can easily become unstable for sufficiently large values of the feedback gain. This can limit the performance of the controller in some cases.

The VPA control is designed to work in a particular frequency range. This makes the control less likely to affect the dynamics of the actuator and the high-pass filter is not needed in the control loop. In the particular case of chatter suppression the VPA control is focused on reducing the real part of the FRF, while the DVF consumes significant energy trying to reduce the whole frequency range including regions with no contribution to chatter. However, the VPA control has some disadvantages that should be taken into account. The parameters should be appropriately chosen in order to achieve the desired performance. Moreover, if the dynamics of the system change during the process, then the controller could no longer be the appropriate one and an adaptive law should be implemented [17].

## 4 Experimental results

In this section some experimental results are presented. Two different structures are considered and both control laws are tested. The first structure is a unique mode workpiece fixed to the ground. In the second part of our tests the actuator is mounted on the ram of a milling machine. Both control laws are again tested for different ram outputs. Finally some milling processes are performed with the VPA control to show the usefulness of this control law.

### 4.1 Workpiece

First the DVF control law is tested on the workpiece. The structure is excited with an impact hammer and an accelerometer is attached on the surface of the structure to measure the acceleration. Fig 5 shows the magnitude of the FRF for different values of the feedback gain. The real part is shown in Fig. 6. As expected the vibration is reduced increasing  $g_v$ . However, values larger than 400 lead to unstable results.

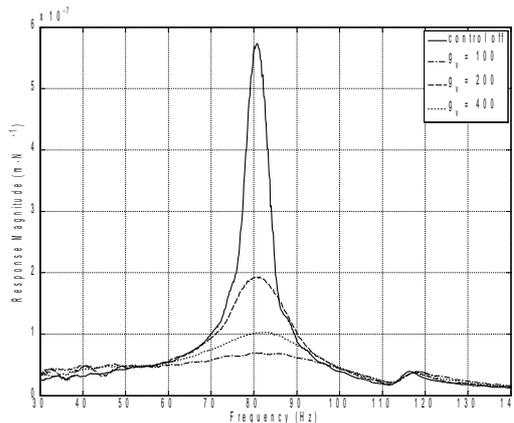


Fig. 5: Magnitude of the FRF of the workpiece using the DVF

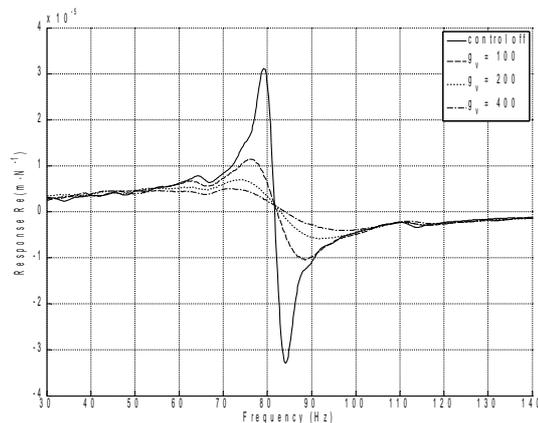


Fig. 6: Real part of the FRF of the workpiece using the DVF

Subsequently the VPA control law is tested on the workpiece. In this case it is not very important to consider the upmilling or the downmilling case since we are not dealing with a milling process. However, the downmilling tuning is chosen for comparison. Fig. 7 displays the magnitude of the obtained FRF for different values of gain  $\mu$ . The effect of the controller can be clearly seen in Fig. 8 where the real part is plotted. In this Figure we can see the appearance of two troughs around the negative real part of the FRF. The optimal tuning is the one that makes these two troughs equal. Both Figures show that a larger damping is obtained increasing the value of  $\mu$ . One could think to use very large values of this gain to obtain a better result. However, there are a few limitations for this. First, increasing this value the low frequency gain of the controller is increased as well. On the hand, the voltage input to the actuator can not take larger values

than a determined value. This means that for sufficiently large values of  $\mu$  the control signal can be saturated and then the control law could not be efficient enough. Moreover, increasing this gain the frequency range with negative real part is increased as well, while the positive peak is moved to a lower frequency (see Fig. 8). This is not very important while the minimum in the negative real part is augmented, but for large values of  $\mu$  the positive peak could reach the low frequencies where there are the structural modes of the actuator and make the whole process unstable. In practical situations, we should consider the magnitude of the oscillations on the structure to guarantee that the control signal remains inside the admissible voltage amplitude range for the actuator. We should check if the structural mode of the structure is near the modes of the actuator as well. If that happens then we should use small values of  $\mu$  not exciting the dynamics of the actuator. Finally, both controllers are compared for their optimal tuning. Fig. 9 shows that using a DVF controller a larger reduction of the magnitude of the FRF in the whole range of frequencies is obtained, while the VPA results to be better increasing the minimum of the negative part of the FRF near the problematic mode of the structure. For this reason, the VPA seems to be more efficient for chatter considerations, where the real part of the FRF is used to determine the chatter stability.

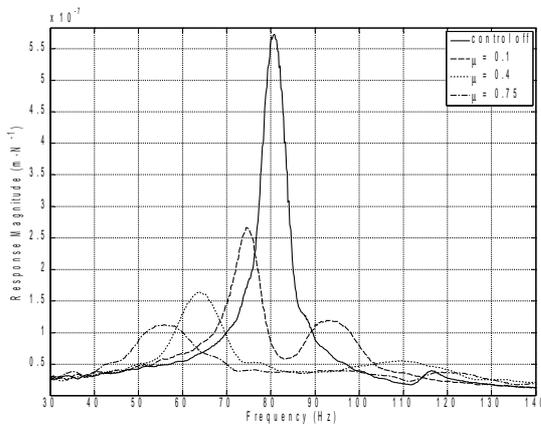


Fig. 7: Magnitude of the FRF of the workpiece using the VPA control

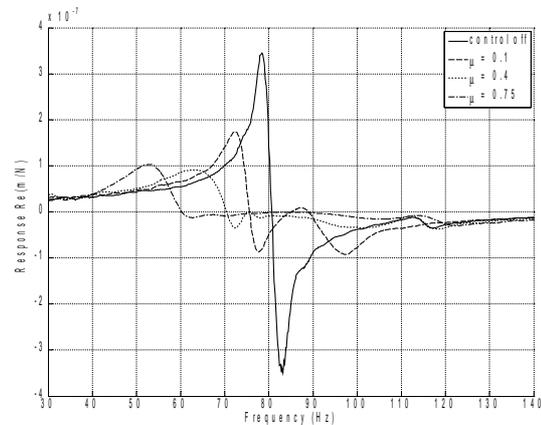


Fig. 8: Real part of the FRF of the workpiece using the VPA control

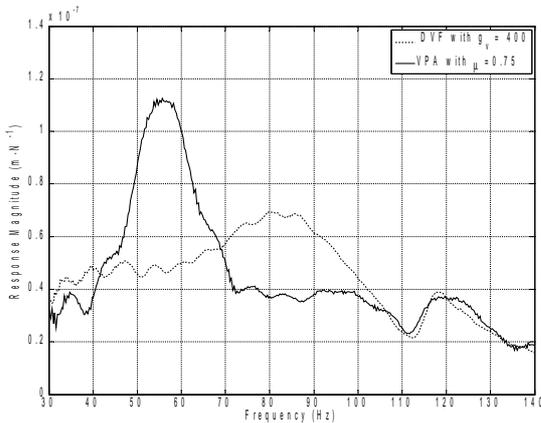


Fig.9: Magnitude comparison between DVF and VPA controls

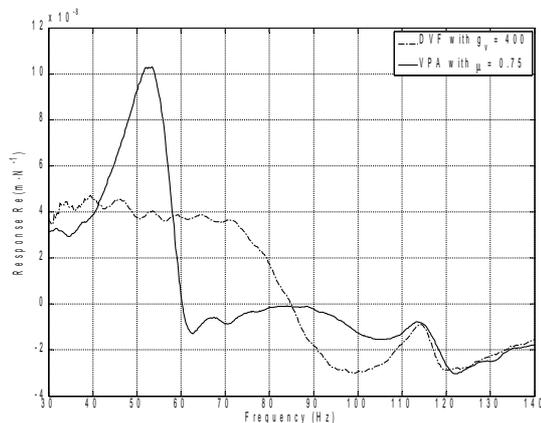


Fig.10: Real part comparison between DVF and VPA controls

## 4.2 Ram of a milling machine

Now the actuator is mounted on the ram of a milling machine. The dynamics of the ram change according to the operation point. For that reason the parameters of the VPA controller should be calculated for each operating point. The experiments are performed with the machine switched off to avoid further vibrations.

Fig. 11 shows the magnitude of the FRF for both control laws when the ram output is 1000mm. The DVF control has been used with the optimal feedback gain obtained from an experimental study. The VPA control has been tuned to avoid the main structural mode at 42 Hz. The real part of the FRF is plotted in

Fig. 12. In both Figures it is clear that the VPA achieves a better performance near the tuned mode. Looking in detail those two figures we can observe some of the advantages and disadvantages of both control laws commented in section 3. The DVF control law attenuates the whole frequency range, but it excites the structural mode of the actuator. We can see that there is a small peak at low frequencies. Remember that in the identification performed to the actuator in section 2 the structural mode was found around 8 Hz. On the other hand, the VPA control has no effect on it. The drawback of the VPA control is that the response of secondary structural modes at lower frequencies are excited. We should look if the influence of these modes is relevant for chatter stability. Since we are tuning the controller for a downmilling process we just look at the negative real part and in this case stability is not lost due to these secondary modes. Another important conclusion is that the parameters of the VPA control have been calculated directly from the modal analysis using equations (14)-(15) without further experimental tuning while the feedback gain of the DVF has to be calculated from an experimental tuning. Since a modal analysis is faster to perform than a manual tuning of a feedback gain, where several tests should be performed until the optimal value is found, we can conclude that VPA control results to be more straightforward and easier to implement in this case.

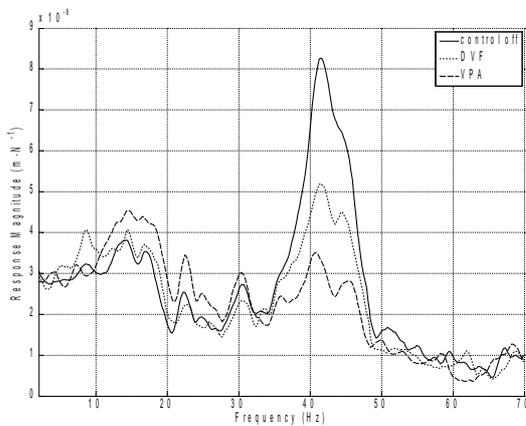


Fig. 11: Magnitude of the FRF for a 1000mm ram output

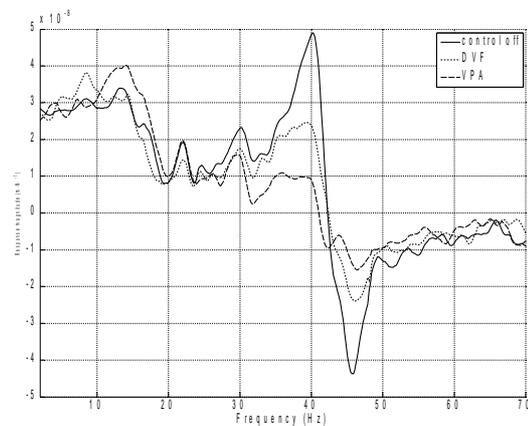


Fig. 12: Real part of the FRF for a 1000mm ram output

Other two tests are performed for different ram outputs. Fig. 13-16 show the FRFs for both cases. In these cases we can observe similar results. The DVF control excites the mode of the actuator, while the VPA control excites secondary modes. In both cases the VPA control results to be the best option to increase the chatter stability.

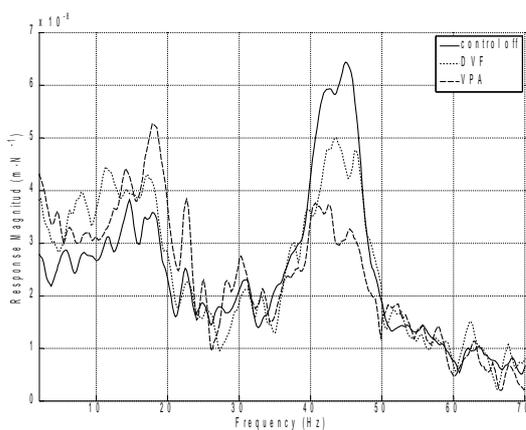


Fig. 13: Magnitude of the FRF for 800 mm ram output

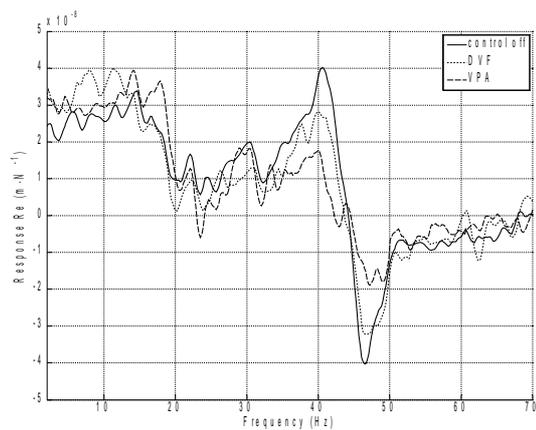


Fig. 14: Real part of the FRF for 800 mm ram output

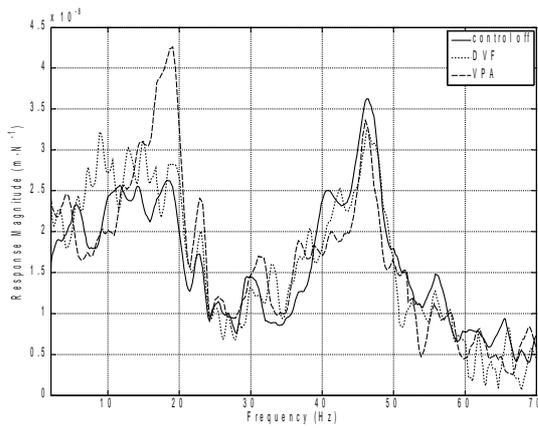


Fig. 15: Magnitude of the FRF for a 600 mm ram output

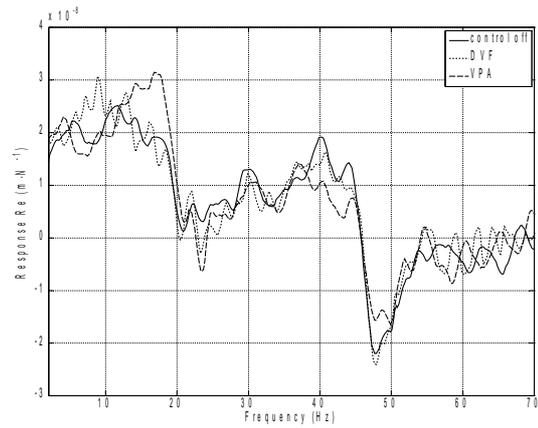


Fig. 16: Real part of the FRF for a 600 ram output

### 4.3 Milling Tests

Finally some milling tests are performed for VPA control. Tests have been performed for a 120 mm tool of 8 teeth with a radial immersion of 80 mm, a depth cut of 4.2 mm, 500 rpm and downmilling conditions. The material of the workpiece is steel F1140. In the first experiment the ram is chosen to operate in a region where the chatter is not very severe. Figure 17 shows the measured acceleration on the ram through time. The control signal to the actuator is plotted in Figure 18. In Figure 17 the improvement is not apparent, so a frequency analysis is performed and displayed in Figure 19. In this case the chatter frequency is 44.39 Hz and we can see that it is completely removed by the actuator when the control signal is applied. The tooth pass frequency is the one located at 66.7 Hz. Finally, another test is performed moving the ram to a another position where stiffness is reduced and stronger chatter is obtained. In this case the time domain signal is enough just to see the improvement obtained with the VPA control law (Fig.20). The frequency analysis of this signal shows that the chatter peak does not appear again (Fig.22) when the control is applied (Fig. 21).

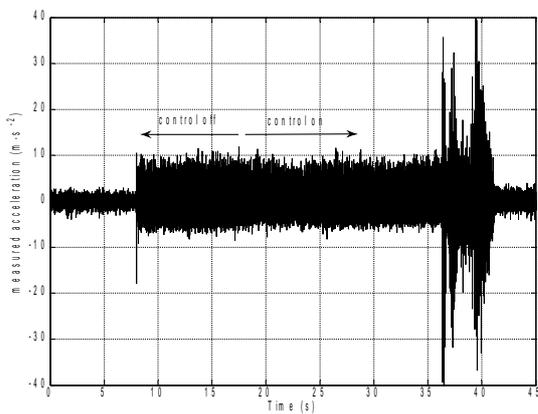


Fig.17: Acceleration measured in milling process 1

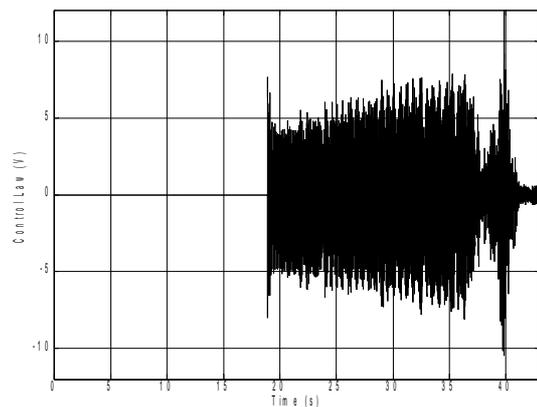


Fig.18: Control law in milling process 1

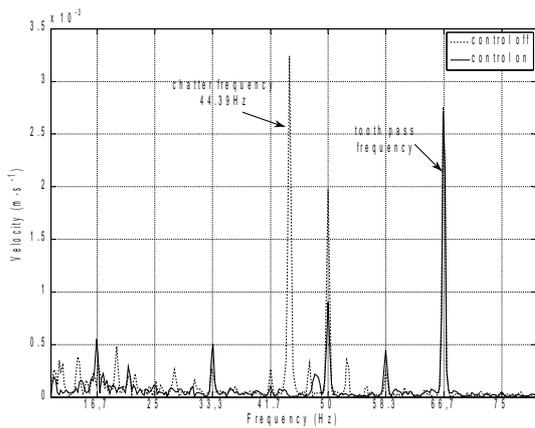


Fig.19: frequency analysis for milling process 1

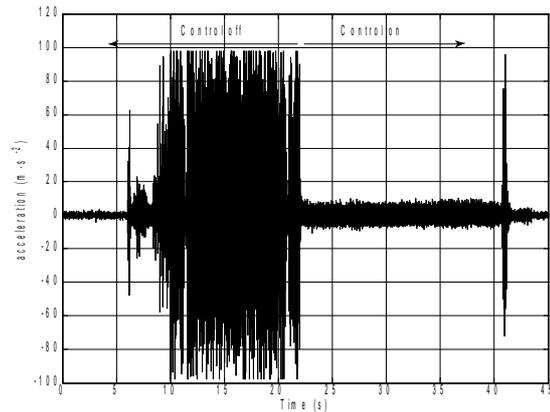


Fig.20: Acceleration measured on milling test 2

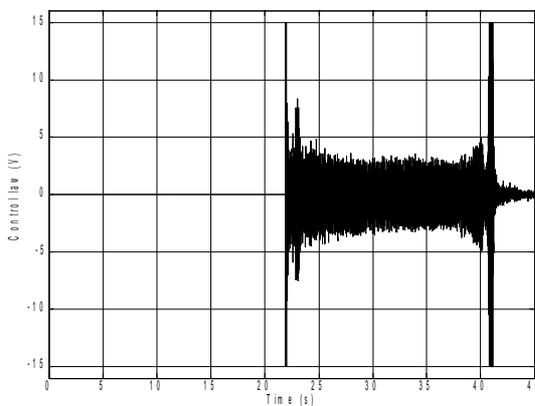


Fig.21: Control law for milling test 2

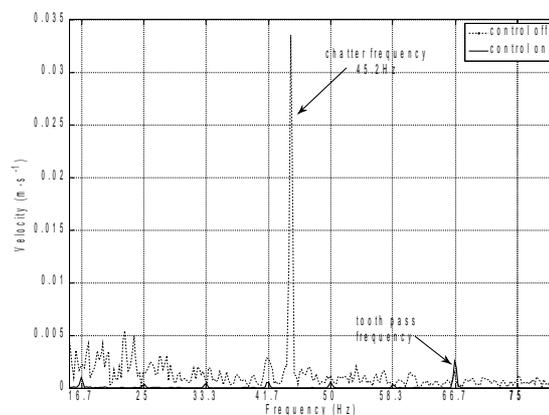


Fig.22: Frequency analysis for milling test 2

## 5 Conclusions

In this paper an active damper using an inertial actuator for chatter suppression is presented. Two different control laws are considered and compared. First the typical DVF control is explained. This control law is widely used for active damping due to its simplicity. Different authors have proposed different type of controller which could lead to better results. We propose the VPA control using a tuning method focused in the real part of the FRF of the structure to be damped. This fact is found to be considerably useful to increase the chatter stability. The VPA control law overcome some of the main drawbacks of the typical DVF control. The parameters of the VPA control can be directly calculated from a direct model analysis of the structure, while the DVF requires a manual tuning to guarantee the stability. In addition, the VPA control focuses its effect on the problematic mode and is less likely to excite the structural modes of the actuator, main drawback of the DVF control. Some practical tests have been performed showing that a theoretically tuned VPA control improves performance obtained with the DVF control. However, the VPA can not be directly implemented in some cases. When the dynamics of the structure change during the process then an unique VPA controller could not be valid and an adaptive law should be implemented. Moreover, if there are more than one structural mode to be damped then a more complex controller should be implemented. One possible solution to be studied is a combination of different VPA controllers.

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