Comparison of model free control strategies for chatter suppression by an inertial actuator

Iker Mancisidor* and
Alexander Pena-Sevillano

Dynamics and Control,
IK4-Ideko,
Arriaga Industrialdea 2, 20870, Elgoibar, Gipuzkoa, Spain
Email: imancisidor@ideko.es
Email: apenasevillano@ideko.es
*Corresponding author

Rafael Barcena

Department of Electronic Technology,
University of the Basque Country,
UPV-EHU, 3 Rafael Moreno, 48013, Bilbao, Bizkaia, Spain
Email: rafa.barcena@ehu.eus

Oier Franco

Dynamics and Control,
IK4-Ideko,
Arriaga Industrialdea 2, 20870, Elgoibar, Gipuzkoa, Spain
Email: ofranco@ideko.es
and
Precision Controls Laboratory,
University of Waterloo,
200 University Avenue West,
Waterloo, ON, N2L3G1, Canada

Jokin Munoa

Dynamics and Control,
IK4-Ideko,
Arriaga Industrialdea 2, 20870, Elgoibar, Gipuzkoa, Spain
Email: jmunoa@ideko.es
and
Department of Mechanical Engineering,
Faculty of Engineering of Bilbao,
Aeronautics Advanced Manufacturing Center (CFAA),
Alameda de Urquijo s/n, 48013, Bilbao, Bizkaia, Spain

Luis Norberto López de Lacalle

Department of Mechanical Engineering,
Faculty of Engineering of Bilbao,
Aeronautics Advanced Manufacturing Center (CFAA),
Alameda de Urquijo s/n, 48013, Bilbao, Bizkaia, Spain
Email: norberto.lzlacalle@ehu.eus

Abstract: The employment of inertial active damping devices for chatter suppression in machining processes has been widely studied. The main drawback is that machine tools generally offer reduced spaces for the actuator location close to the cutting point and, hence, a reduced volume design is required for this external device. Consequently, the ratio between the force capability and the occupied volume is of major importance. Therefore, the performance of the actuator must be exploited to the fullest, which depends principally on the selected control strategy to determine the force to be exerted. The work presents the comparison of the most used feedback control algorithms and a novel strategy based on the regenerative effect disturbance. Their results are assessed for orthogonal cutting cases, with the aim of optimising the behaviour of the active damping system.

Keywords: active damping; inertial actuator; chatter.

Reference to this paper should be made as follows: Mancisidor, I., Pena-Sevillano, A., Barcena, R., Franco, O., Munoa, J. and de Lacalle, L.N.L. (2019) 'Comparison of model free control strategies for chatter suppression by an inertial actuator', *Int. J. Mechatronics and Manufacturing Systems*, Vol. 12, Nos. 3/4, pp.164–179.

Biographical notes: Iker Mancisidor is an Engineer in the Dynamics and Control Research Group at IK4-Ideko Technology Center, since 2009. He obtained the PhD degree in Mechanical Engineering from the University of the Basque Country (UPV-EHU) in 2014 and he spent 6 months as a postdoctoral fellow at the Precision Controls Laboratory of the University of Waterloo in 2016. His research topics include machine-tool chatter, active damping, electromagnetic actuators, mechatronic systems, machine-tool control and automation, and modal analysis.

Alexander Pena-Sevillano is an Engineer in the Dynamics and Control Research Group at the IK4-Ideko Technology Center. He earned the degree in Industrial Electronics and Automation Engineering and the MSc degree in Advanced Electronic Systems from the University of Basque Country (UPV/EHU) in 2015 and 2017, respectively. His research topics include control theory, active damping, electromagnetic actuators, mechatronics and industrial computers.

Rafael Bárcena received his MSc and PhD degrees in Physics from the University of the Basque Country (UPV/EHU) in 1994 and 2001, respectively. Since 1998, he has been working at the Department of Electronic Technology of the UPV/EHU, as a Researching Professor. His research topics include control theory, hybrid control and sampled systems.

Oier Franco is an Engineer in the Dynamics and Control Research Group at IK4-Ideko Technology Center, since 2016. He received his degree in Industrial Engineering from the University of Deusto in 2014 and MSc degree in Industrial Engineering specialising in Structural Mechanics from the University of Mondragon in 2016. Currently, he is a PhD candidate at the University of Waterloo focusing on active chatter suppression techniques. His research interest includes machine-tool chatter, active damping, machine-tool feed drive systems and control.

Jokin Munoa obtained his PhD degree in Mechanical Engineering from the Mondragon University in 2007. He has 20 years of experience in machine tool technology development. He is managing the Dynamics and Control Research Group of IK4-Ideko Technology Center since 2009 and is currently a Visiting Researcher at the Aeronautics Advanced Manufacturing Center (CFAA). Furthermore, he is associated member at the International Academy of Production Engineering (CIRP). His current research efforts include interaction of manufacturing process and machine tool, new techniques for the measurement of machine tool dynamics, active damping devices and methods, mechatronic simulation and stability models.

Luis Norberto López de Lacalle obtained his PhD degree in Mechanical Engineering in 1993 and currently is chairman in Mechanical Engineering at the University of the Basque Country (UPV-EHU). His career is distinguished by the knowledge generation and dissemination in mechanical design and manufacturing area, with more than 130 papers in JCR journals, two reference books and a strong collaboration with industrial companies. He is the leader on the university excellence group on advanced manufacturing, recognised by the Basque Government since 2010, and the manager of the Aeronautics Advanced Manufacturing Center (CFAA) since 2014.

1 Introduction

The recent developments in machine tools and material technology are leading to an increased demand for a higher productivity and precision in machining processes, while consuming at the same time the smallest possible amount of material and energy. Manufactured parts have also become lighter and less stiff to minimise costs or fuel consumption in transport. This all results in a greater likelihood to have self-excited vibrations problems, which prevent the machining process from obtaining the required surface finishes and decrease the life of tools and machine's mechanical components. These vibrations, indeed, commonly known as chatter, are considered one of the major limitations in current machine tools (Altintas, 2012).

Consequently, the modelling and suppression of chatter vibrations has been one of the major concerns for researchers (Altintas and Weck, 2004). After the first studies, the regenerative effect was reported as the principal reason for chatter vibrations by Tobias and Fischwick (1958) and Tlusty and Polacek (1963). Different mathematical models were then presented (Altintas and Budak, 1995; Zatarain et al., 2004) in order to explain the phenomenon and predict the process stability diagrams, from a simple orthogonal cutting case (Merrit, 1965) to more complicated interrupted milling processes (Davies et al., 2002; Munoa et al., 2013; Iglesias et al., 2016).

Different solutions to suppress chatter have been presented in the literature (Munoa et al., 2016). Some of them are based on changing the cutting parameters such as the spindle speed (Smith and Tlusty, 1992; Bediaga, 2009) or the tool geometry (Vanherck, 1967; Iglesias et al., 2019). The regenerative effect distortion is another method to stabilise the cutting process by a continuous spindle speed variation (SSV) (Hoshi et al., 1977; Sexton and Stone, 1978). However, all these solutions require a modification of the cutting conditions that is not always possible.

One of the most effective methods which is valid for all cutting conditions is to introduce additional damping to the cutting process (Uriarte et al., 2014). Passive dampers can be appropriate in many cases (Sims, 2007) but they present limitations when dynamic parameters can vary considerably. Active systems can be adapted to changing conditions since their actuation force is based on a vibration measurement and a control strategy (Cowley and Boyle, 1970). In reference to the manner to introduce active dampers on a machine, Ehmann and Nordmann (2002) concluded that inertial actuators located close to the vibration source were the most efficient option. In this regard, several applications of inertial active dampers have been proposed for different machining operations (Loix and Verschueren, 2004; Baur and Zaeh, 2012; Barrenetxea et al., 2018).

When these active systems are employed, one of their most important parameters is the ratio between the force capability and the occupied volume. The inertial actuators are efficient in high vibration displacement locations and, since modes with high flexibility on the cutting point are causing chatter vibrations, the actuator proximity to the tool is required for an efficient performance. The problem is that generally machine tools offer reduced spaces for the actuator location near the tool and the force capacity of the actuators is limited by their size. Therefore, the performance of the actuators must be exploited to the fullest, optimising their control strategy.

Various methods of active vibration control have been proposed in the literature, which can be classified as feedback or feedforward algorithms (Preumont, 2002). Generally, feedback controls are used in machine tools, since feedforward controllers require a reference disturbance signal, which is very complicated to construct in the case of chatter vibrations.

On the one hand, the strategies can be based on a system behaviour model, such as the Virtual Passive Absorber (Huyanan and Sims, 2007) or the $H\infty$ robust control (Zaeh et al., 2017). Although these algorithms can focus the actuation force on the frequencies of interest, in some cases the elaboration of a reliable mathematical model can be a complicated task, and at best, it is just a low dimensional approach of the actual system. Moreover, the model must be adapted for any change on the dynamics or the cutting parameters. On the other hand, model free control strategies can adapt their actuation to any changing condition without the need of a model of the system. Indeed, they can control a wide frequency range, which is usually limited by a high-pass and a low-pass filter.

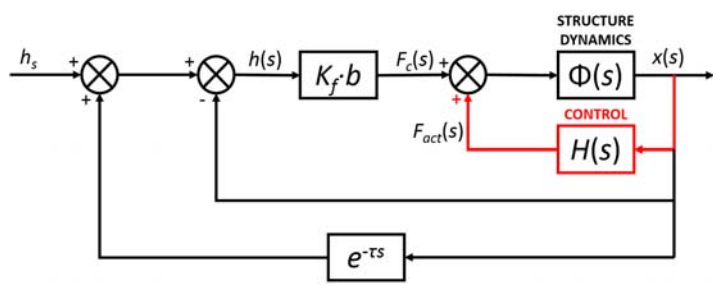
As Fuller et al. (1996) stated, the target of the common model free feedback control laws employed to reduce the vibrations are based on a structural response modification of the system. That was demonstrated by Brecher and Schulz (2004), where the main dynamic parameters were modified by using position, velocity and acceleration feedback. Apart from these three common feedback strategies, the paper proposes the comparison of another novel feedback algorithm based on disturbing the regenerative effect (Mancisidor et al., 2015b; Mancisidor et al., 2019). In this way, the aim of the paper is to define the optimum control strategy for inertial actuators depending on the cutting conditions, which is demonstrated by stability lobe diagram comparisons.

For that purpose, theoretical simulations and an experimental validation have been carried out. Experimental cutting tests can generate some problems to compare control strategies, due to the lack of repeatability caused by non-controllable parameters such us tool wear or material properties. These uncertainties can be controlled by a mathematical model when a hardware-in-the-loop (HIL) simulator is employed. By means of the HIL, the cutting vibration is induced on a mechanical structure while the active damping system is introduced in a realistic manner. Thus, real problems such as mechatronic delays or electrical noises on the sensors appear on the test bench and their influence on the results can be assessed. Some authors used a cantilever beam as the structure to introduce the cutting vibration and the active damping (Ganguli et al., 2005; Huyanan and Sims, 2007; Mancisidor et al., 2013), but the delay existing on the mechatronic systems was neglected to calculate the cutting vibrations. In the present work, the HIL proposed by Mancisidor et al., 2015a is tested. In this test bench, the delay is considered and, hence, the real stability lobes can be reproduced in a linear flexure.

2 Control strategies

The feedback strategies rely on the output measurement information to adapt the response to a desired value. Such principle can be easily applied to the chatter vibrations scheme proposed by Merrit (1965) by adding a new closed loop where a control strategy $H(s)$ calculates the actuation force $F_{\text{act}}(s)$ (Figure 1).

Figure 1 Addition of control closed loop to the block diagram of machining processes (see online version for colours)



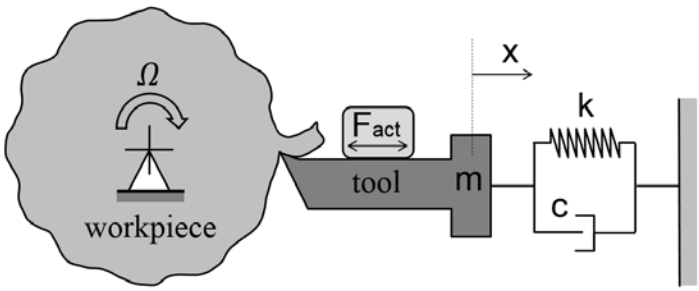
$$x(s) = \Phi(s)\left(F_c(s) + F_{\text{act}}(s)\right) \tag{1}$$

$$x(s) = \Phi(s)\left(K_f b\left(h_s + x(s)\left(e^{-\tau s} - 1\right)\right) + F_{\text{act}}(s)\right) \tag{2}$$

being h_s the static chip thickness related to the feed per revolution, $h(s)$ the total chip thickness, K_f the specific cutting coefficient, b the axial depth of cut, $F_c(s)$ the cutting force and τ the revolution period.

For a clear comparison of different control strategies, the present paper considers an orthogonal single degree of freedom (DOF) case (Figure 2). In this case, the structure dynamics are described as a simple transfer function (eq.3), where m, c and k are the structure mass, damping and stiffness respectively. Thus, the effect of each control algorithm over different dynamic parameter terms can be mathematically explained through the general cutting equation with the actuator force [equation (4)]. Nevertheless, the same statements could be then applied in more complex processes, such as interrupted milling.

Figure 2 Orthogonal single degree of freedom cutting system



$$\Phi(s) = \frac{1}{ms^2 + cs + k} \tag{3}$$

$$\left(ms^2 + cs + k\right)x(s) = K_f b\left(h_s + x(s)\left(e^{-\tau s} - 1\right)\right) + F_{\text{act}}(s) \tag{4}$$

Direct velocity feedback (DVF) is the most common control law in machine tools with chatter problems, when an external actuator is employed. It is based on the measurement of vibration velocity and its negative feedback is multiplied by a gain ($H(s) = -G_v \cdot s$). Thus, the control forces perform as viscous damping [equation (6)] and increase the overall damping of the process.

$$F_{\text{act}}(s)_{\text{DVF}} = -G_v \cdot s \cdot x(s) \tag{5}$$

$$\left(ms^2 + (c + G_v)s + k\right)x(s) = K_f b\left(h_s + x(s)\left(e^{-\tau s} - 1\right)\right) \tag{6}$$

Direct acceleration feedback (DAF) is a similar algorithm to DVF, but an acceleration feedback is considered ($H(s) = -G_a \cdot s^2$) instead. Therefore, the modal mass is affected, and the natural frequency is increased or reduced, depending on the sign of the gain [equation (8)].

$$F_{\text{act}}(s)_{\text{DAF}} = -G_a \cdot s^2 \cdot x(s) \tag{7}$$

$$\left((m + G_a)s^2 + cs + k\right)x(s) = K_f b\left(h_s + x(s)\left(e^{-\tau s} - 1\right)\right) \tag{8}$$

A similar effect is caused by direct position feedback algorithm (DPF) ($H(s) = -G_p$), due to modification of stiffness [equation (10)].

$$F_{\text{act}}(s)_{\text{DPF}} = -G_p \cdot x(s) \tag{9}$$

$$\left(ms^2 + cs + \left(k + G_p\right)\right)x(s) = K_f b\left(h_s + x(s)\left(e^{-\tau s} - 1\right)\right) \tag{10}$$

Delayed position feedback (DelPF) is focused on the remaining term, which is the regenerative effect ($H(s) = -G_\tau \cdot e^{-\tau s}$). The control strategy is based on reducing virtually the engagement between the current and previous waves and, hence, the stability can be successfully increased. Moreover, the delays existing on the control system can be compensated due to its delayed actuation.

$$F_{\text{act}}(s)_{\text{DelPF}} = -G_\tau \cdot e^{-\tau s} \cdot x(s) \tag{11}$$

$$\left(ms^2 + cs + k\right)x(s) = K_f b\left(h_s + x(s)e^{-\tau s}\left(1 - \frac{G_\tau}{K_f b}\right) - x(s)\right) \tag{12}$$

If the actuator did not reach saturation, the control delay could totally stabilise all chatter vibrations, by means of an optimal gain that cancels the delayed term ($G_\tau = K_f b$). However, the required force is usually high and therefore only a reduction of the regenerative term is possible (Mancisidor et al., 2019).

Furthermore, the acceleration signal can be employed in the same way as the position signal, since in critically stable cases these both signals are in counterphase. It is called delayed acceleration feedback (DelAF) and in this case, a neutral type of delay differential equation (NDDE) is obtained. Indeed, the feedback must be positive and the gain should be much smaller.

$$F_{act}(s)_{DelAF} = G_{\tau\text{a}} \cdot e^{-\tau s} \cdot s^2 \cdot x(s) \tag{13}$$

$$\left(ms^2 + cs + k\right)x(s) = K_f b\left(h_s + x(s)e^{-\tau s}\left(1 + \frac{G_{\tau\text{a}} s^2}{K_f b}\right) - x(s)\right) \tag{14}$$

Therefore, if all the control strategies are considered, the next equation is obtained:

$$\left((m + G_{\text{a}})s^2 + (c + G_{\text{v}})s + k + G_{\text{p}}\right) = K_f b\left(h_s + x(s)e^{-\tau s}\left(1 - \frac{G_\tau}{K_f b}\right) - x(s)\right) \tag{15}$$

Considering only the dynamic part and the critically stable case ($s = \text{i}\omega_c$) where vibration oscillates at the chatter frequency ω_{c} and b_{lim} is the limit depth of cut for chatter-free machining

$$-\omega_{\text{c}}^2(m + G_{\text{a}}) + \text{i}\omega_{\text{c}}(c + G_{\text{v}}) + k + G_{\text{p}} = K_f b_{\text{lim}}\left(\left(1 - \frac{G_\tau}{K_f b_{\text{lim}}}\right)e^{-i\omega_{\text{c}}\tau} - 1\right) \tag{16}$$

As the imaginary part of equation (16) must be zero, the next relation is obtained:

$$K_f b_{\text{lim}} = \frac{-\omega_{\text{c}}(c + G_{\text{v}})}{\sin \omega_{\text{c}}\tau} + G_\tau \tag{17}$$

which can be brought in the real part to obtain the relation for the period [equation (20)]:

$$-\omega_{\text{c}}^2(m + G_{\text{a}}) + k + G_{\text{p}} + K_f b_{\text{lim}} = \left(K_f b_{\text{lim}} - G_\tau\right)\cos \omega_{\text{c}}\tau \tag{18}$$

$$-\omega_{\text{c}}^2(m + G_{\text{a}}) + k + G_{\text{p}} + G_\tau = \frac{\omega_{\text{c}}(c + G_v)}{\sin \omega_{\text{c}}\tau}(1 - \cos \omega_{\text{c}}\tau) \tag{19}$$

$$\tau(\omega_c) = \frac{2}{\omega_c}\left(\arctan\left(\frac{-\omega_c^2(m+G_a)+k+G_p+G_\tau}{\omega_c(c+G_v)}\right)+j\pi\right), \quad \text{where } j = 1, 2, \ldots \tag{20}$$

If the trigonometric identity approach (($A\cos\alpha)^2 + (A\sin\alpha)^2 = A^2$) is applied over the real and imaginary parts of equation (16), the analytical formulation for the limit depth of cut can be obtained, which can be used to build the stability lobe diagrams:

$$\left(-\omega_c^2(m+G_a)+k+G_p+K_f b_{\lim}\right)^2+\left(\omega_c(c+G_v)\right)^2=\left(K_f b_{\lim}-G_\tau\right)^2 \tag{21}$$

$$b_{\lim}(\omega_c)=\frac{\left(-\omega_c^2(m+G_a)+k+G_p\right)^2+\omega_c^2(m+G_a)^2-G_\tau^2}{2K_f\left(\omega_c^2(m+G_a)-k-G_p-G_\tau\right)} \tag{22}$$

3 Theoretical simulations

In the present section, the improvement achieved by the described active control strategies is studied on the stability lobes diagrams. According to Tobias (1965), a dimensionless formulation can be developed for single DOF systems. The dimensionless stability diagrams allow the comparisons among different machining conditions, where the shape of the lobes only depends on the relative damping ratio. Therefore, the stability study turns out to be more general and independent from machining circumstances.

The present study defines four dimensionless parameters (Zatarain et al., 2010): relative damping ratio ξ, normalised chatter frequency λ, normalised depth of cut μ, and normalised rotation frequency β. Such parameters can be defined by the next equations [equations (23), (24), (25) and (26)], where ω_n is the natural frequency of the structure.

$$\xi = \frac{c}{2m\omega_n} \tag{23}$$

$$\lambda = \frac{\omega_c}{\omega_n} \tag{24}$$

$$\mu(\lambda,\xi) = b_{\lim}(\lambda)\frac{K_f}{2k\xi(1+\xi)} \tag{25}$$

$$\beta(\lambda,\xi) = \frac{\Omega}{\omega_n} = \frac{2\pi}{\tau(\lambda)\omega_n} \tag{26}$$

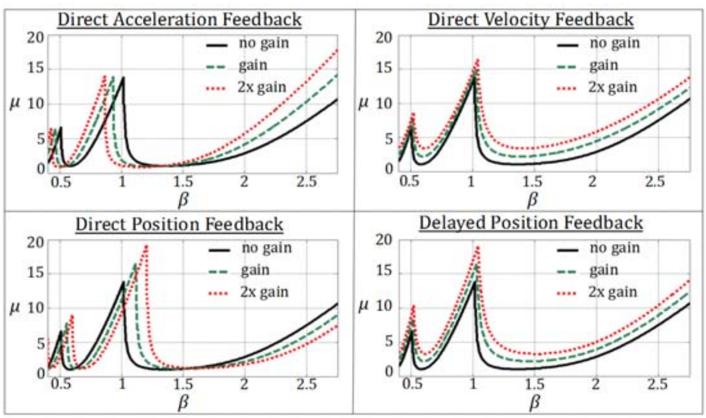
3.1 Frequency domain simulation

The first simulation is run in the frequency domain, where nonlinearities are discarded. The obtained stability diagrams allow studying the effect obtained by each control strategy in a theoretical way (Figure 3). The magnitude of the improvement is not comparable since the introduced gain is unlimited and therefore, only the tendency is analysed.

It can be observed that both DPF and DAF shift the maximum stability point, also known as 'sweet spots', because their modification of mass or stiffness results in a natural frequency change. Consequently, these two control strategies can be interesting options when cutting conditions close to the sweet spot need to be stabilised. Conversely, the improvement obtained by DVF around the sweet spots is negligible, but it increases considerably the limit depth of cut. As system instabilities are usually happening near the limit depth of cut, the DVF is the typically used control strategy for inertial actuators. On the other hand, the novel control strategy (DelPF) shows a similar increase of the chatter-free zone along every β values. Therefore, it is can be considered as a promising control law.

However, as previously mentioned, the principal limitation on active drives is the available volume and, hence, the applicable actuation force. Therefore, in order to compare these control laws, a force saturation should be also taken into account.

Figure 3 Effect of different control strategies over stability diagrams (see online version for colours)



3.2 Time domain simulation

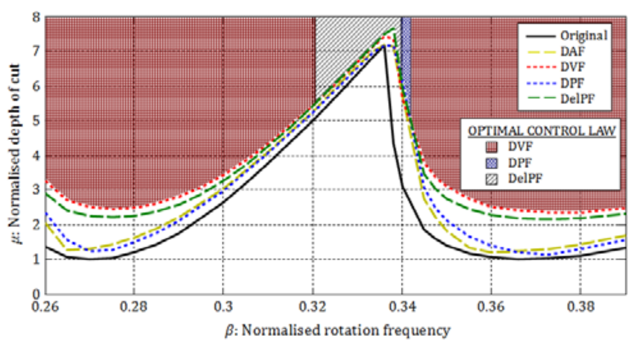
When a nonlinear factor such as the inertial actuator force saturation is added to the process, the combined system equations become nonlinear. This makes the frequency domain simulation not suitable to simulate the behaviour of the inertial device. Therefore, in the present work, the same model proposed by Mancisidor et al. (2015b) is adapted to orthogonal cutting process simulation in time domain, considering the cutting process and the behaviour of the actuator at the same time.

First, the proposed feedback strategies have been theoretically simulated for the dynamic parameters of the HIL structure tested on the experimental validation (Table 1). The simulated actuator characteristics are described on Table 2. The optimal gain has been applied for each strategy and the optimal stability limit for an orthogonal single DOF case has been obtained (Figure 4).

The figure shows that DVF is the best strategy at almost all spindle speeds, which explains why this strategy is the most employed for chatter vibrations. The novel delayed position feedback (DelPF) is not far from the effectiveness of DVF and can even overcome it around the sweet spot zones.

However, the magnitude of improvement is not the same for all zones when DelPF is used, as frequency domain tests predicted. The reason is that the higher depth of cut the lower proportion of regenerative part is reduced [see equation (12)], and then the actuator force saturation can affect significantly. For that reason, the improvement over the DVF on the sweet spot zones is not as much as expected. After the sweet spot, there also exists a narrow zone where DPF is the best control law, but with no measurable difference over the other control strategies. It is demonstrated that DPF and DAF strategies, which modify the natural frequency, require a huge quantity of force to improve the stability.

Figure 4 Effect of different control strategies over stability diagrams with 1.4% damping (see online version for colours)



It is known that the effect of DVF is higher in structures with a low damping ratio and, as explained before, the shape of the dimensionless stability lobes only depends on the relative damping ratio. Therefore, different relative damping values are simulated in Figure 5 (ξ = 4%) and Figure 6 (ξ = 12%) while the force saturation is maintained, in order to compare the control strategies in different conditions.

Figure 5 Effect of different control strategies over stability diagrams with 4% damping (see online version for colours)

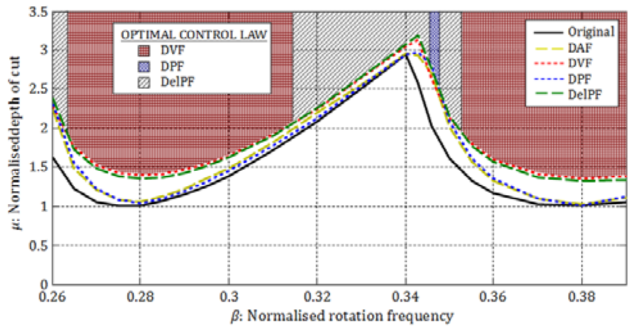
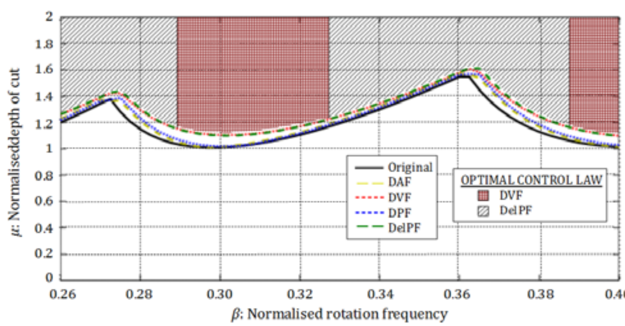


Figure 6 Effect of different control strategies over stability diagrams with 12% damping (see online version for colours)



The figures show that the difference between DVF and DelPF are considerably reduced when the damping is increased. The zone where DelPF bests the other control strategies becomes wider at the expense of DVF control strategy. Nevertheless, the difference is negligible.

In conclusion, the results show that when orthogonal cutting is performed, DVF control law is the best strategy, although its difference is reduced as the original damping of the system is higher. Moreover, the excellent theoretical performance of the novel control law presented in this study (DelPF) with respect to the other strategies is highlighted.

4 Experimental validation in HIL

The theoretical results are simulated in an ideal environment, but the introduction of some noise or delay can have a big impact on the obtained conclusions.

Apart from the real inertial actuator saturation, another important effect which limits the performance of a feedback controller in mechanical systems is the unmodelled system delay (Fuller et al., 1996). Such phase shift may arise because of the dynamic response of the sensors and actuators being used or due to time delays in the controller. If either position or acceleration feedback are implemented in a control loop with a system delay, the effect of the delay changes considerably the damping of the system. This fact significantly modifies the behaviour of the control and, in some cases, the system can become unstable. In contrast, when velocity feedback is employed, the delay has only a small effect on the effective mass and stiffness, which makes it a more robust control strategy. On the other hand, when the novel strategy (delayed position/acceleration feedback) is applied, the force is introduced with the revolution period delay, which is usually higher than the existing delays due to the actuator and controller. Therefore, these undesired delays can be compensated, reducing the revolution period delay of the feedback control.

In order to analyse these problems, the present work uses the hardware-in-the-loop (HIL) proposed by Mancisidor et al. (2015a) to compare the stability improvements achieved by each control law. This system reproduces experimentally on a simple mechanical structure any equivalent orthogonal cutting process where the regenerative chatter can appear. The use of this kind of test-benches avoids several repeatability problems which can arise in real cutting tests due to the involvement of a large number of uncertain parameters such as tool wear or material properties.

Table 1 Dynamic parameters of the structure

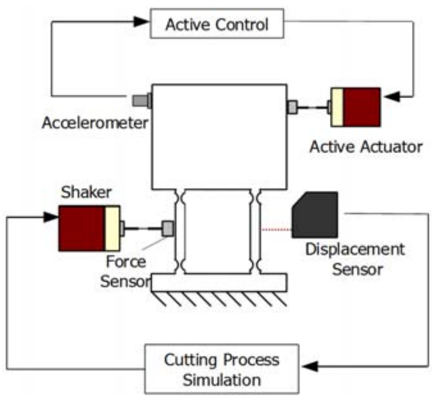
Natural frequency ω_n *(Hz)*	*Damping* ξ *(%)*	*Rigidity* *k (N/m)*
177.8	1.4	43.24 · 107

Table 2 Dynamic parameters Data Physics V2 actuator

Suspension frequency $\omega_{n_{act}}$	13 Hz
Damping ratio ξ_{act}	0.9%
Stroke	±1.25 mm
Actuator delay τ_{act}	650 µs
Bandwidth	20–300 Hz
Force capability $F_{act_{max}}$	5 N

The HIL system employed in this work is based on a linear flexure structure (see Table 1) and the damping is provided by means of eddy currents, which introduce a pure linear viscous damping without changing other dynamic properties. In this way, the possibility of reproducing different damping values (up to 4%) is enabled. The active control system is mounted as shown in Figure 7, where Data Physics V2 shaker (Table 2), with a 5N force capability is hung and a collocated accelerometer is used for feedback.

Figure 7 Hardware-in-the-loop system with the active control loop (see online version for colours)



Source: Mancisidor et al. (2015a)

The feedback control is performed by a dSPACE DS1005 with a 500 µs sampling period. The low and high frequency noise is filtered by a band-pass between 100 and 1,000 Hz and the commanded voltage has been limited in order not to exceed the maximum force capability of the active actuator.

In order to verify the theoretical simulations showed previously, the damping ratios of 1.4% and 4% have been tested. Figure 8 shows the results for the HIL simulator with original damping (1.4%), which are not far from the theoretical simulation. DVF control law is still the best in almost all zones, although the other strategies, principally DelPF, can overcome DVF around the sweet spot. When the damping of the system is increased to 4% (Figure 9), the delayed feedback strategy is very close to DVF and its improvement more sizeable on the sweet spot. This demonstrates that DelPF is a control strategy to bear in mind in future inertial actuator applications.

Figure 8 Comparison of strategies in the HIL simulator with 1.4% damping (see online version for colours)

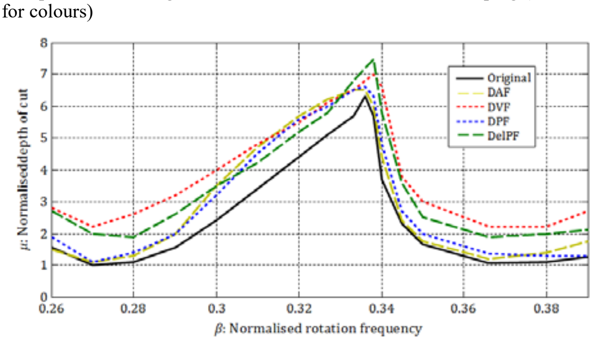
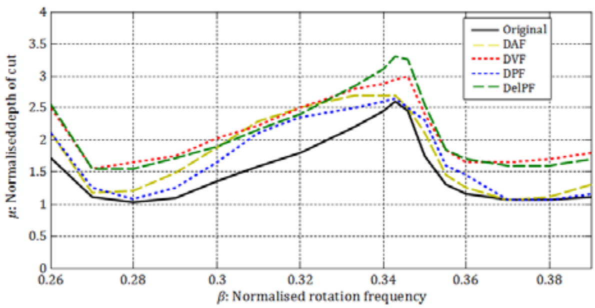


Figure 9 Comparison of strategies in the HIL simulator with 4% damping (see online version for colours)



5 Conclusions

The machine tools offer reduced spaces for integrating inertial actuators close to the cutting point with the objective of suppressing chatter vibrations, and, therefore, the actuation force must be optimised as much as possible. This work presents a comparison of model free feedback control strategies for chatter suppression in a single degree of freedom orthogonal cutting case.

First the effect of each strategy on the terms of the dynamic characteristic equation is analysed and then the improvement carried out by the control strategies is studied in the stability diagrams.

The theoretical simulations show that direct velocity feedback (DVF) obtains the highest improvement on the minimum stability zone of the lobe diagram, while the novel delayed position feedback (DelPF) can overcome the improvement obtained by DVF in maximum stability zones. Moreover, the higher the damping of the system, the lower the improvement difference of DVF with respect to the delayed feedback strategy is.

These results are confirmed by hardware-in-the-loop simulations, which are closer to the reality due to the introduction of problems such as electrical noise or control delays. Therefore, in orthogonal cutting process, the DVF law performs better, although the DelPF is close to it and even better in small areas around the resonance speeds.

Acknowledgements

This work has been partially supported by the Spanish Ministry of Economy, Industry and Competitiveness under CIMEC (RTD-2016-5097-3) project and by the Basque government under COACVIRE (KK-2018/00084).

References

Altintas, Y. (2012) *Manufacturing Automation*, 2nd ed., Cambridge University Press, Cambridge, UK.

Altintas, Y. and Budak, E. (1995) 'Analytical prediction of stability lobes in milling', *CIRP Annals – Manufacturing Technology*, Vol. 44, No. 1, pp.357–362.

Altintas, Y. and Weck, M. (2004) 'Chatter stability of metal cutting and grinding', *CIRP Annals – Manufacturing Technology*, Vol. 53, No. 2, pp.619–642.

Barrenetxea, D., Mancisidor, I., Beudaert, X. and Munoa, J. (2018) 'Increased productivity in centerless grinding using inertial active dampers', *CIRP Annals*, Vol. 67, No. 1, pp.337–340.

Baur, M. and Zaeh, M.F. (2012) 'Development and experimental validation of an active vibration control system for a portal milling machine', *Proceedings of 9th International Conference on High Speed Machining*, 7–8 March, San Sebastian, Spain.

Bediaga, I. (2009) *Supresión del chatter regenerative mediante variación en proceso de la velocidad de giro*, PhD thesis, Mondragon University, Mondragon, Spain.

Brecher, C. and Schulz, A. (2004) 'Active auxiliary mass damper', *Proceedings of ISMA 2004*, 20–22 September, Leuven, Belgium, pp. 3563-3572.

Cowley, A. and Boyle, A. (1970) 'Active dampers for machine tools', *Annals of the CIRP*, Vol. 18, No. 1, pp.213–222.

Davies, M.A., Pratt, J.R., Dutterer, B. and Burns, T.J. (2002) 'Stability prediction for low radial immersion milling', *Journal of Manufacturing Science and Engineering*, Vol. 124, No. 2, pp.217–225.

Ehmann, C. and Nordmann, R. (2002) 'Low cost actuator for active damping of large machines', *IFAC Conference on Mechatronic Systems*, Berkeley, California, USA.

Fuller, C.R., Elliot, S.J. and Nelson, P.A. (1996) *Active Control of Vibration*, Academic Press Limited, London, UK.

Ganguli, A., Deraemaeker, A., Horodinca, M. and Preumont, A. (2005) 'Active damping of chatter in machine tools – demonstration with a 'hardware in the loop' simulator', *Proceedings of the Institution of Mechanical Engineers. Part I: Journal of Systems and Control Engineering*, Vol. 219, No. 5, pp.359–379.

Hoshi, T., Sato, M., Sakisaka, N. and Moriyama, I. (1977) 'Study of practical application of fluctuating speed cutting for regenerative chatter control', *CIRP Annals*, Vol. 25, No. 1, pp.175–179.

Huyanan, S. and Sims, N.D. (2007) 'Vibration control strategies for proof-mass actuators', *Journal of Vibration and Control*, Vol. 13, No. 12, pp.1785–1806.

Iglesias, A., Dombovari, Z., Gonzalez, G., Munoa, J. and Stepan, G. (2019) 'Optimum selection of variable pitch for chatter suppression in face milling operations', *Materials*, Vol. 12, No. 1, p.112.

Iglesias, A., Munoa, J., Ciurana, J., Dombovari, Z. and Stepan, G. (2016) 'Analytical expressions for chatter analysis in milling operations with one dominant mode', *Journal of Sound and Vibration*, Vol. 375, pp.403–421.

Loix, N. and Verschueren, J.P. (2004) 'Stand-alone active damping device', *Actuator 2004 – 9th International Conference on New Actuators*, 14–16 June, Bremen, Germany.

Mancisidor, I., Barcena, R., Munoa, J. and Etxebarria, A. (2013) 'Design of a bench hardware-in-the-loop system for the study of chatter in turning', *2013 European Control Conference (ECC)*, 17–19 July, Zurich, Switzerland.

Mancisidor, I., Beudaert, X., Etxebarria, A., Barcena, R., Munoa, J. and Jugo, J. (2015a) 'Hardware-in-the-loop simulator for stability study in orthogonal cutting', *Control Engineering Practice*, Vol. 44, pp.31–44.

Mancisidor, I., Munoa, J., Barcena, R., Beudaert, X. and Zatarain, M. (2015b) 'Coupled model for simulating active inertial actuators', *International Journal of Advanced Manufacturing Technology*, Vol. 77, Nos. 1–4, pp.581–595.

Mancisidor, I., Pena-Sevillano, A., Dombovari, Z., Barcena, R. and Munoa, J. (2019) 'Delayed feedback control for chatter suppression in turning machines', *Mechatronics*, November, Vol. 63, p.102276 [online] https://doi.org/10.1016/j.mechatronics.2019.102276.

Merrit, H.E. (1965) 'Theory of self-excited machine-tool chatter. Contribution to machine-tool chatter. Research 1', *Journal of Engineering for Industry*, Vol. 87, No. 4, pp.447–453.

Munoa, J., Beudaert, X., Dombovari, Z., Altintas, Y., Budak, E., Brecher, C. and Stepan, G. (2016) 'Chatter suppression techniques in metal cutting', *CIRP Annals – Manufacturing Technology*, Vol. 65, No. 2, pp.785–808.

Munoa, J., Dombovari, Z., Mancisidor, I., Yang, Y. and Zatarain, M. (2013) 'Interaction between multiple modes in milling processes', *Machining Science and Technology*, Vol. 17, No. 2, pp.165–180.

Preumont, A. (2002) *Vibration Control of Active Structures: An Introduction*, Kluwer Academic Publishers, New York, USA.

Sexton, J. and Stone, B. (1978) 'The stability of machining with continuously varying spindle speed', *Annals of the CIRP*, Vol. 27, No. 1, pp.321–326.

Sims, N.D. (2007) 'Vibration absorbers for chatter suppression: a new analytical tuning methodology', *Journal of Sound and Vibration*, Vol. 301, Nos. 3–5, pp.592–607.

Smith, S. and Tlusty, J. (1992) 'Stabilizing chatter by automatic spindle speed regulation', *CIRP Annals*, Vol. 41, No. 1, pp.433–436.

Tlusty, J. and Polacek, M. (1963) 'The stability of machine tools against self-excited vibrations in machining', *ASME International Research in Production Engineering*, pp.465–474, 9–12 September, Pittsburgh, USA.

Tobias, S.A. (1965) *Machine Tool Vibrations*, Blackie & Sons Limited, Glasgow.

Tobias, S.A. and Fischwick, W. (1958) 'Theory of regenerative machine tool chatter', *The Engineer*, Vol. 205, pp.199–239.

Uriarte, L., Etxaniz, I., Iturbe, J., Zatarain, M. and Munoa, J. (2014) 'Chatter suppression in a high speed magnetic spindle by adding damping', *Journal of Machine Engineering*, Vol. 14, No. 4, pp.65–74.

Vanherck, P. (1967) 'Increasing milling machine productivity by use of cutter with non-constant edge pitch', *Proceedings of the 8th MTDR Conference*, Manchester, UK.

Zaeh, M., Kleinwort, R., Fagerer, P. and Altintas, Y. (2017) 'Automatic tuning of active vibration control systems using inertial actuators', *CIRP Annals – Manufacturing Technology*, Vol. 66, No. 1, pp.365–368.

Zatarain, M., Bediaga, I., Munoa, J. and Insperger, T. (2010) 'Analysis of directional factors in milling: importance of multi-frequency calculation and the inclusion of the effect of the helix angle', *International Journal of Advanced Manufacturing Technology*, Vol. 47, Nos. 5–8, pp.535–542.

Zatarain, M., Munoa, J., Villasante, C. and Sedano, A. (2004) 'Estudio comparativo de los modelos matemáticos de chatter en fresado: monofrecuencia, multifrecuencia y simulación en el tiempo', *XV Congreso de Máquinas-Herramienta y Tecnologías de Fabricación*, San Sebastian, Spain.