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Delayed feedback control for chatter suppression in turning machines

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ABSTRACT

The employment of active control devices has been demonstrated as one of the most effective methods to suppress chatter vibrations in machine tools. However, the commonly employed control laws are generalist and focused on the reduction of any type of vibration. The present work proposes a control law focused on the main cause of chatter vibrations. The strategy is based on a delayed feedback that distorts the regenerative effect and virtually reduces the overlap factor between successive waves. Experimental results show that the proposed law enables the suppression of chatter vibrations.

1. Introduction

In the current manufacturing industry, chatter vibrations play an important role, since they are one of the main limitations in cutting capability of machine tools. Such self-excited vibrations are produced by the tool/workpiece interaction, and induce a poor surface finish and an excessive tool wear.

The fundamental theory of chatter was developed by Tlusty and Polacek [1] and Tobias [2], who identified the regenerative effect as the major chatter mechanism. Later on, Merrit [3] presented a systematic theory for the stability analysis using a feedback loop control theory. The use of stability diagrams is one of the most popular methods to characterize the process stability [4]. These diagrams, known as stability lobes, separate the stable and unstable regions depending on the spindle speed and the depth of cut. A rich literature is addressing the chatter problem employing various techniques to increase the stability limit [5].

Some authors proposed to use stability lobe diagrams in order to choose the optimal cutting conditions [6,7]. However, currently one of the most employed techniques is to add external damping to the machines, both passively and actively. Passive actuators are characterized by their simplicity and their relative low cost [8], although they are not suitable for processes where the dynamics of the machine change according to the working position during the machining process. Therefore, the integration of active dampers has been widely analyzed for many years as one of the most effective chatter suppression methods due to their adaptability to variable conditions.

An electromagnetic inertial drive was first introduced in [9] to actively damp a machine tool structure. Since then, the active control systems have been widely employed in order to suppress chatter vibrations. Different actuator technologies have been used, such as piezoelectric [10,11], electrorheological [12], electrohydraulic [13] or electromagnetic [14] systems.

Various control laws have been tested such as direct position feedback, direct velocity feedback and direct acceleration feedback [15]. The direct velocity feedback is aimed at adding damping and is the most common control law due to its capability to raise the chatter limit in all areas of the stability lobes [16]. Moreover, compared to the acceleration and position feedbacks, the direct velocity feedback is less prone to destabilize the high frequency modes [17] and less affected by the delays in the feedback loop [18]. The feedback delay introduced by the controller, the power electronics and the actuator has generally an important impact on the performance of the active damping strategy and thus the phase shift should be compensated for the targeted frequencies [19].

Up to now, the different control laws developed are all dedicated basically to the improvement of the dynamic stiffness of the machine tool structure. However, this is not the only way to increase chatter stability, indeed, the disturbance of the regenerative effect leads to many well-known chatter suppression methods. On the one hand, it can be performed by the design of special tools like variable pitch cutters [20,21], serrated cutting edges [22] or alternating helix angles [21]. On the other hand, some authors proposed the distortion of the regenerative effect by a continuous spindle speed variation [23]. These techniques prove that chatter suppression can be achieved by focusing on the regenerative term.

Following the same principle, a novel control law for inertial actuators, called Delayed Position Feedback (DelPF), was introduced [16,24].

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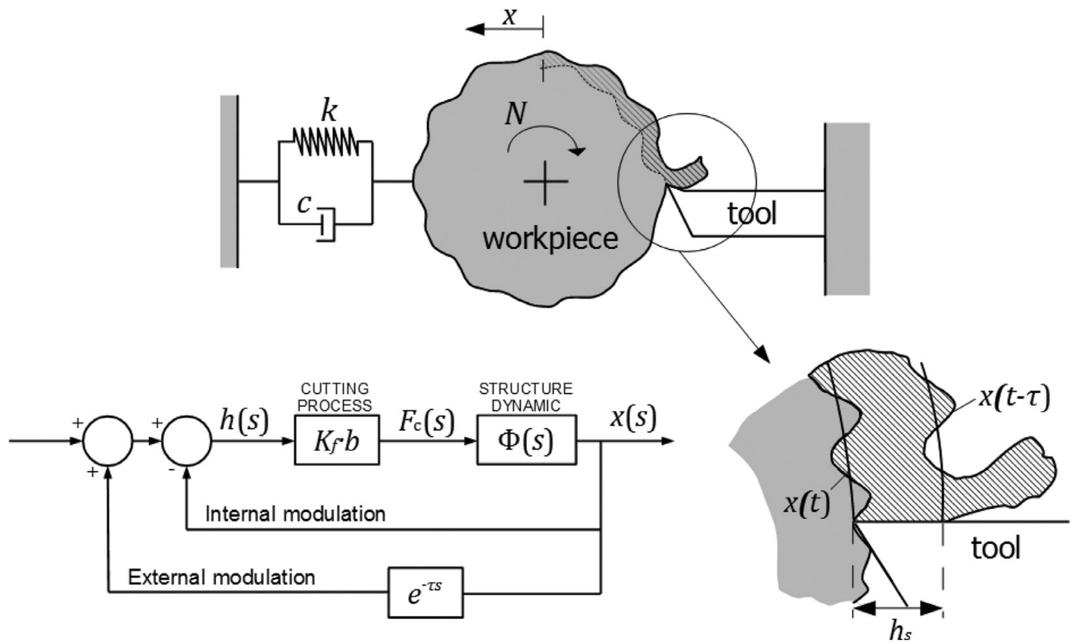

Fig. 1. Regenerative chatter vibrations in orthogonal cutting process [25].

This control strategy was compared to traditional feedback algorithms in a milling process [24]. The improvement carried out by the proposed control law was comparable to the well-known Direct Velocity Feedback (DVF), although the need for a cutting process model was demonstrated for milling cases. In poorly damped cases, DVF control strategy obtained slightly better results in low stability zones and delayed feedback had the best results close to resonances. However, the latter showed the advantage of compensating the delay coming from the actuator and control system, since it applies the force based on the vibration measured in the previous period.

The present work studies the delayed feedback control for chatter suppression in turning operations, where the model of the cutting process is not required. The rest of the paper is organized as follows. First, the delayed feedback is theoretically analyzed and simulated. Then, both acceleration and position delayed feedbacks are compared, and finally the proposed chatter suppression control law is validated in a hardware in the loop simulator and a vertical turning center.

2. Theoretical study of the delayed feedback

2.1. Time domain description of delayed position feedback

The effect of the delayed feedback can be explained considering a groove turning case, where an orthogonal cutting with a single degree of freedom occurs (see Fig. 1). The cutting force $F_c(t)$ is obtained as the product of the cutting coefficient K_f, the depth of cut b and the chip thickness $h(t)$:

$$h(t) = h_s - x(t) + x(t-\tau) \tag{1}$$

where the static chip thickness is denoted by h_s, the dynamic chip thickness is generated by the difference between the actual $x(t)$ and past position $x(t-\tau)$ of the tool, the revolution period is $\tau = 60/N$ (see Fig. 1), and N the spindle speed in rpm.

Therefore, the following differential equation describes the behavior of the system, where m, c and k are the modal mass, damping and stiffness of the system, respectively:

$$m\ddot{x}(t) + c\dot{x}(t) + kx(t) = F_c(t) \equiv K_f b\big(h_s - x(t) + x(t-\tau)\big). \tag{2}$$

When an active inertial device is introduced on the structure, the general cutting equation can be written as

$$m\ddot{x}(t) + c\dot{x}(t) + kx(t) = K_f b\big(h_s - x(t) + x(t-\tau)\big) + F_{act}(t), \tag{3}$$

where F_{act} is the force introduced by the actuator which is dependent on the selected control law.

Instead of using the active force to change the dynamics of the structure as it is usually done, the DelPF acts on the cutting forces to reduce the regenerative term. Indeed, this strategy is feeding back the vibration produced during the previous revolution $F_{act}(t) = -G_p x(t-\tau)$. The measured position is delayed by the revolution period τ which is a known constant for a given process. The gain of the delayed position feedback G_p is a parameter that must be tuned. Theoretically, the feedback delay τ should only depend on the spindle speed but this control law advantageously offers the possibility to compensate the control and actuator delays as it will be detailed further. With this control strategy, the regenerative term can be reduced as shown in the following governing equation:

$$m\ddot{x}(t) + c\dot{x}(t) + kx(t) = K_f b\left(h_s - x(t) + \left(1 - \frac{G_p}{K_f \cdot b}\right)x(t-\tau)\right). \tag{4}$$

The proposed control strategy can be analogously explained as a virtual reduction of the overlap factor between the current and the previous waves [3], cleaning the workpiece from former undulations. For instance, the equivalence of this overlap factor can be illustrated in different turning operations (see Fig. 2). When a grooving operation is performed, a complete overlapping exists between successive revolutions. In the extreme situation, if the tool is threading in a single pass no chatter regeneration can happen. However, in longitudinal turning, depending on the feed, tool geometry, depth of cut and the spindle speed, the tool cuts again a portion μ of the surface left by the tool in the previous revolution [3,26]. The equivalent of this portion for this control law is

$$\mu = 1 - \frac{G_p}{K_f b}, \tag{5}$$

with

$$m\ddot{x}(t) + c\dot{x}(t) + kx(t) = K_f b\big(h_s - x(t) + \mu x(t-\tau)\big). \tag{6}$$

The perfect behavior of the control law could avoid regenerative chatter for any cutting process when the regenerative part is completely

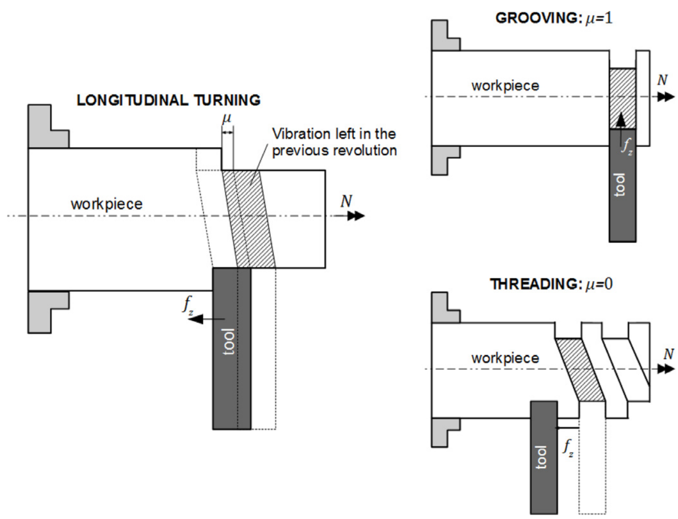

Fig. 2. Scheme of different turning operations.

removed ($\mu = 0$). Therefore, an optimal control gain exists, which depends on the cutting coefficient K_f and the depth of cut b:

$$G_{\mathrm{Popt}} = K_f b \tag{7}$$

2.2. Stability analysis in frequency domain

As explained in the introduction, the stability of a process is usually studied by means of stability lobe diagrams. When non linearities such as force saturation or limited bandwidth of the actuator are not considered, the analysis of the equations in the frequency domain offers the possibility to obtain these diagrams directly by scanning the frequencies of interest.

The governing Eq. (4) has equilibrium position at $x_0 = K_f b\ h_s/k$ around which a perturbation $x(t) = x_0 + \tilde{x}(t)$ can be introduced to remove the effect of the static chip thickness h_s [27]. In the Laplace domain, the perturbed system will have the form:

$$\left(s^2 m + sc + k\right) \cdot \tilde{x}(s) = K_f b\left(\left(1 - \frac{G_{\mathrm{p}}}{K_f \cdot b}\right) e^{-s\tau} - 1\right)\tilde{x}(s). \tag{8}$$

Considering the critically stable case ($s = \mathrm{i}\omega_c$) where vibration oscillates at the chatter frequency ω_c and $b_{\lim}$ is the limit depth of cut for chatter-free machining, the stability of the system can be analyzed:

$$-\omega_c^2 m + \mathrm{i}\omega_c c + k = K_f b_{\lim}\left(\left(1 - \frac{G_{\mathrm{p}}}{K_f \cdot b_{\lim}}\right) e^{-\mathrm{i}\omega_c\tau} - 1\right). \tag{9}$$

The equation can be partitioned into real and imaginary parts, considering the Euler formulation. First, the next relation can be obtained from the imaginary part:

$$-\omega_c c = \left(K_f b_{\lim} - G_{\mathrm{p}}\right)\sin\omega_c\tau \tag{10}$$

$$K_f b_{\lim} = G_{\mathrm{p}} - \frac{\omega_c c}{\sin\omega_c\tau}. \tag{11}$$

Considering the real part of Eq. (9) the following relations can be obtained:

$$K_f b_{\lim} - \omega_c^2 m + k = \left(K_f b_{\lim} - G_{\mathrm{p}}\right)\cos\omega_c\tau. \tag{12}$$

Considering Eqs. (11), (12) can be rewritten as:

$$\frac{G_{\mathrm{p}} - \omega_c^2 m + k}{\omega_c c} = \frac{1 - \cos\omega_c\tau}{\sin\omega_c\tau} = \tan\frac{\omega_c\tau}{2} \tag{13}$$

Then, the equation for the period as a function of ω_c can be obtained:

$$\tau_n = \frac{2}{\omega_c}\left(\arctan\left(\frac{G_{\mathrm{p}} - \omega_c^2 m + k}{\omega_c c}\right) + n\pi\right), \quad \text{where } n = 0, 1, 2, \ldots \tag{14}$$

If the trigonometric identities are applied over Eqs. (10) and (12), the analytical formulation for the limit depth of cut as a function of ω_c can be obtained, which can be employed to construct the stability lobe diagrams.

$$\begin{aligned}&\left(\left(K_f b_{\lim} - G_{\mathrm{p}}\right)\sin\omega_c\tau\right)^2 + \left(\left(K_f b_{\lim} - G_{\mathrm{p}}\right)\cos\omega_c\tau\right)^2\\ &\quad = \omega_c^2 c^2 + \left(K_f b_{\lim} - \omega_c^2 m + k\right)^2\end{aligned} \tag{15}$$

$$b_{\lim} = \frac{{G_{\mathrm{p}}}^2 - \left(\omega_c^2 m - k\right)^2 - \omega_c^2 c^2}{2K_f\left(G_{\mathrm{p}} + k - \omega_c^2 m\right)}. \tag{16}$$

Therefore, the limit boundary is defined by the critical boundaries $(\tau_n(\omega_c),\ b_{\lim}(\omega_c))$. The stability diagrams are built scanning the chatter frequency ω_c and obtaining values of limit depth of cut $b_{\lim}(\omega_c)$ and spindle speed through the delay $\tau_n(\omega_c)$ for each chatter frequency ω_c. The lobe number or order (n) is an integer value with an important role in the stability analysis. First of all, for each n a lobe or a branch of the stability diagram $\tau_n(\omega_c)$ can be obtained. The n value defines the number of entire waves that are created in a rotation period τ by the chatter vibration, and the entire ratio between the chatter frequency and rotation frequency.

Furthermore, it is especially interesting to relate the limit depth of cut ($b_{\lim}$) to the minimum value of the stability ($b_{\min}$) for the case without any active control ($G_{\mathrm{p}} = 0$). In this simple case, the minimum stability is associated with the minimum of the real part of the receptance or dynamic flexibility of the machine [3,4,28] that occurs when $\omega_{c,\min} = \omega_n\sqrt{1 + 2\xi}$:

$$b_{\min} = \frac{2k\xi(1 + \xi)}{K_f}, \tag{17}$$

where $\omega_n = \sqrt{k/m}$ and $\xi = c/(2\ m\ \omega_n)$ are the natural frequency of the dominant mode and its relative damping ratio, respectively.

According to Tobias [2], a dimensionless formulation can be developed for single degree of freedom (DOF) systems. This formulation permits the analysis of the stability diagrams in function of a minimum number of parameters, and it allows to obtain more general conclusions. Indeed, in the present work dimensionless parametric functions, normalized rotation frequency ($\beta_n(\lambda)$) and normalized limit depth of cut ($\delta_{\lim}(\lambda)$), are defined as in [29] using relative damping ratio ξ and normalized chatter frequency $\lambda = \omega_c/\omega_n$. Note that the floor of the inverse of the dimensionless rotation frequency is equal to the lobe number ($n = floor(1/\beta_n)$). The critically stable boundary in dimensionless parameters is defined as:

$$\beta_n = \frac{2\pi}{\tau_n\omega_n} = \frac{\lambda\pi}{\arctan\left(\frac{g_{\mathrm{p}} - \lambda^2 + 1}{2\xi\lambda}\right) + n\pi}, \quad \text{where } n = 0, 1, 2, \ldots \tag{18}$$

$$\delta_{\lim} = \frac{b_{\lim}}{b_{\min}} = b_{\lim}\frac{K_f}{2k\xi(1 + \xi)} = \frac{-\left(\lambda^2 - 1\right)^2 - (2\xi\lambda)^2}{4\xi(1 + \xi)\left(g_{\mathrm{p}} + 1 - \lambda^2\right)}, \tag{19}$$

where $g_{\mathrm{p}} = \frac{G_{\mathrm{p}}}{k}$ is the normalized gain for DelPF.

In this way, given the normalized gain g_{p} and the relative damping ratio ξ, a scan of different values of λ permits the calculation of the normalized limit depth of cut $\delta_{\lim}(\lambda)$ by Eq. (19) and the normalized rotation frequencies $\beta_n(\lambda)$ by Eq. (18), where each n value draws a different stability lobe. Therefore, the shape of the stability lobes only depends on the relative damping ratio ξ and the normalized gain g_{p}.

Nonetheless, the application of the gain g_{p} on the equations creates an unusual effect. In the traditional stability lobes only negative values of the real part of the dynamics give positive values for the limit depths of cut [1–3]. However, the application of g_{p} causes the appearance of positive depth of cut limits due to the positive values of the real part. Therefore, two branches are shown up on the mathematical solution.

2.3. Theoretical effect of delayed position feedback on the stability

The mathematical equations demonstrate that when the optimal gain (Eq. (7)) is applied with an ideal actuator, without force saturation and bandwidth limit, any process can be stabilized thanks to the proposed control strategy. However, if the gain has a fixed value and is not adapted to the depth of cut, only a range of cases can be improved. Moreover, the main problem is that exceeding the optimal gain causes a regenerative effect in the opposite direction, and in the case that this gain exceeds a limit value, the process can become unstable again. This means that for each depth of cut there is an optimum gain and if a lower depth of cut is machined with the same gain, an originally stable process could be destabilized.

For instance, the optimal value for the normalized gain (g_{pmin}) for the point associated with the minimum stability is obtained combining Eqs. (7) and (17):

$$g_{\mathrm{p_{min}}} = \frac{G_{\mathrm{p_{min}}}}{k} = \frac{K_f b_{\mathrm{min}}}{k} = 2\xi(1+\xi) \cong 2\xi. \tag{20}$$

Therefore, the optimal value of the normalized gain (g_{pmin}) to suppress chatter in the point related to minimum stability can be approximated by two times the relative damping of the vibration mode. In a similar way, the value of the absolute optimal gain for the minimum (G_{pmin}) can be set as the value of the amplitude of the vibration mode ($G_{\mathrm{pmin}} \cong 2\xi k$).

Fig. 3 shows the theoretical effect of the Delayed Position Feedback (DelPF) control law over the dimensionless stability lobes of a single DOF system ($\xi = 1.4\%$) when an ideal active force,without any bandwidth or force limitation from actuator side, is used applying a common normalized gain (g_{p}) for all the cutting conditions of each dimensionless stability diagram. The uncolored range is referred to the chatter free zone once the active control is applied. Fig. 3a shows the stability lobes when a constant optimal gain for the minimum limit depth of cut (Eq. (20)) is applied in the whole set of cutting conditions. It can be observed that the stability margin is increased and, since it is not a huge gain, the lower stability limit branch does not appear in the positive depth of cut region.

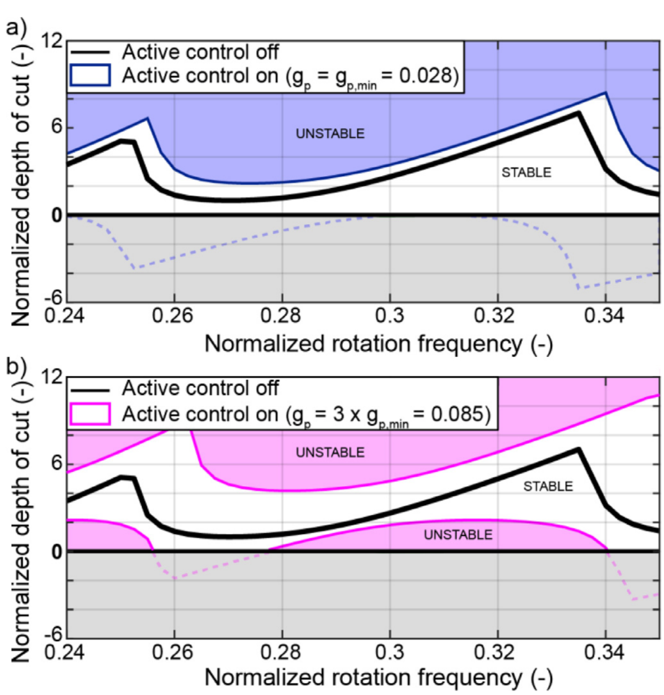

Fig. 3. Effect of the control law over dimensionless stability lobes ($\xi = 1.4\%$; $n = 4$) with an ideal actuator and constant optimum gain. (a) $g_{\mathrm{p}} = g_{\mathrm{p_{min}}} = 0.28$; (b) $g_{\mathrm{p}} = 3g_{\mathrm{p_{min}}} = 0.085$.

When a higher gain is set ($3 \times g_{\mathrm{pmin}}$) as shown in Fig. 3b, originally stable lower depths of cut can be destabilized due to the opposite regenerative effect. The figures show that the positive values of the real part create downside stability lobes while the negative values create the typical upside lobes, similarly to effect of positive and negative directional factors [4,29]. When the gain g_{p} is increased, downside lobes come up and can define positive depths of cut. Fig. 3 shows negative depths of cut for understanding this effect, but they do not have any physical sense. Therefore, it can be stated that a certain gain can improve a range of depths of cut and any depth of cut can be stabilized with the proper gain if the actuator has no force limit.

However, the required force to remove completely the regenerative effect is high and, hence, it is hardly achieved. Indeed, the force saturation of the actuator is generally the most critical parameter preventing the complete elimination of chatter problems with active control systems. In practice, the gain can also be limited by the destabilization of the structure. Therefore, the objective of the control law is to reduce as much as possible the regenerative term.

3. Theoretical comparison of delayed position and acceleration feedback

The previous section has probed theoretically the effect of the delayed displacement signal over the stability of a certain process. Nevertheless, it is well known that in critically stable cases ($s = \mathrm{i}\omega_{\mathrm{c}}$), displacement and acceleration are counterphased signals ($\ddot{x}(\mathrm{i}\omega_{\mathrm{c}}) = -\omega_{\mathrm{c}}^2 x(\mathrm{i}\omega_{\mathrm{c}})$), thus, the acceleration signal could be also directly used to disturb the regenerative effect. Accelerometers are widely used in vibration measurement due to their robustness and low cost. Moreover, as the acceleration feedback has been already employed in the literature for chatter suppression [15,30,31], it is worth considering the Delayed Acceleration Feedback (DelAF) method. In this case the governing equation will have the following form

$$m\ddot{x}(t) + c\dot{x}(t) + kx(t) = K_f b\big(h_{\mathrm{s}} - x(t) + x(t-\tau)\big) + G_{\mathrm{a}}\ddot{x}(t-\tau). \tag{21}$$

This equation is a neutral type of delay differential equation (NDDE), which can behave as a discrete map due to the delayed highest order (acceleration) term in the right-hand-side. It is well-known in this case that the necessary condition for stability is to have non-growing discrete mapping between the acceleration and delayed acceleration [27], that is

$$\left|\frac{G_{\mathrm{a}}}{m}\right| < 1. \tag{22}$$

Considering again the critically stable case of the dynamic part, Eq. (23) is obtained,

$$-\omega_{\mathrm{c}}^2 m + \mathrm{i}\omega_{\mathrm{c}} c + k = K_f b_{\mathrm{lim}}\left(\left(1 - \frac{\omega_{\mathrm{c}}^2 G_{\mathrm{a}}}{K_f b_{\mathrm{lim}}}\right)e^{-\mathrm{i}\omega_{\mathrm{c}}\tau} - 1\right). \tag{23}$$

It can be shown that the optimum gain for DelAF is dependent on the chatter frequency and should be much smaller compared to the one of DelPF of Eq. (7):

$$G_{\mathrm{aopt}} = \frac{K_f b}{\omega_{\mathrm{c}}^2}. \tag{24}$$

The analytical equations for the stability lobe diagram construction can be obtained in the same way as for DelPF, obtaining similar results to (18) and (19):

$$\beta_n = \frac{\lambda\pi}{\arctan\left(\frac{g_{\mathrm{a}} - \lambda^2 + 1}{2\xi\lambda}\right) + n\pi} \tag{25}$$

and

$$\delta_{\mathrm{lim}} = \frac{g_{\mathrm{a}}^2 - \left(\lambda^2 - 1\right)^2 - (2\xi\lambda)^2}{4\xi(1+\xi)\left(g_{\mathrm{a}} + 1 - \lambda^2\right)}, \tag{26}$$

where the normalized gain is defined as $g_a = \frac{\omega_c^2 G_a}{k}$ for DelAF.

The results shown in the previous section are based on an ideal behavior of the control and they could be far from the reality. When a real control system is employed, problems such as bandwidth and force saturation of the actuator or delays may arise, and they must be considered. The implementation of the delayed feedback control has a principal advantage that consists on applying the force one revolution (τ) after the measurements, hence, some problems such as the actuation delay can be compensated.

As explained in the introduction, the feedback control always carries negative effects due to the delays of the controller, the power electronics and the actuator. On the one hand, the controller delay can be approximated as equal to its sampling period (T_s) [32]. On the other hand, the system composed by the actuator and its power electronics adds another delay τ_{act} related to its bandwidth, which can be identified beforehand [19]. In some cases, the sum of these delays is too large to be neglected and control strategies do not work properly if these delays are not compensated. However, the total delay is usually lower than the revolution period delay τ that should be introduced by the delayed feedback strategy. Hence, if the exact value of the delays is known, they can be compensated using the following expression for the delayed feedback $\tau_{feedback}$:

$$\tau_{feedback} = \tau - \tau_c - \tau_{act} \approx \tau - T_s - \tau_{act}, \tag{27}$$

being τ_c and τ_{act} the delay of the controller and the delay of the actuator and power electronics, respectively. This kind of delay compensation cannot be performed with the classical feedback strategies used up to now.

In order to analyze the behavior of the delayed control when real problems appear, a theoretical time domain simulation of an orthogonal cutting has been performed by adapting the approach proposed in [19] to turning applications. In this simulation, the bandwidth and the force saturation of the actuator are considered, as well as the delays. It is assumed that the direction of the chip thickness is constant, since the vibration can occur in only one direction. Moreover, the direction of the active control force is identical and therefore, the performed force is completely employed on the regenerative term reduction.

The dynamic parameters of the hardware in the loop presented in the next section have been simulated (Table 1). For the control application, the dynamics of the Data Physics V2 actuator (Table 2) are simulated and a sample time (T_s) of 500 µs is considered. It should be mentioned that having a high enough sampling rate with respect to the target frequencies is necessary, in order to avoid problems related to the discrete control [33]. In this work, the sampling frequency is ten times higher than the natural frequency of the system, so the effect of the discrete control can be neglected.

Fig. 4 shows the results of the chatter stability simulation, where the stability improvement provided by both Delayed Position Feedback (DelPF) and Delayed Acceleration Feedback (DelAF) are compared. In this case, the gain has been adapted and the optimum gain for the limit depth of cut without control $\delta_{lim,\,off}$ (black line in Fig. 4) has been applied for each rotation frequency. Nevertheless, due to the force saturation and bandwidth limitation, the required force is not achieved. The results demonstrate that both strategies can improve the stability of the process in all areas of the stability lobes, although the areas with higher original stability are less improved due to the higher force required for these depths of cut.

Furthermore, Fig. 4 shows that almost equal results are obtained with both strategies. In the present work, since accelerometers will be used in the experimental part for measuring the vibration due to their lower cost and higher robustness, DelAF control strategy will be employed from now on. In this way, the measured signal does not need any integration and only high and low pass filters are used to avoid feeding back the measurement noise out of the frequencies of interest.

The simulations shown in Fig. 4 have a perfect compensation of the delay, employing Eq. (27). However, the knowledge of all the required data for this calculation is not simple in the real world and therefore the compensation can have some minor errors. Fig. 5 shows the sensitivity of the DelAF control strategy with respect to the error performed on the delay $\tau_{feedback}$ calculation. It can be observed that large errors can mainly affect the low stability zone and hence, the importance of a proper calculation of the delay is demonstrated.

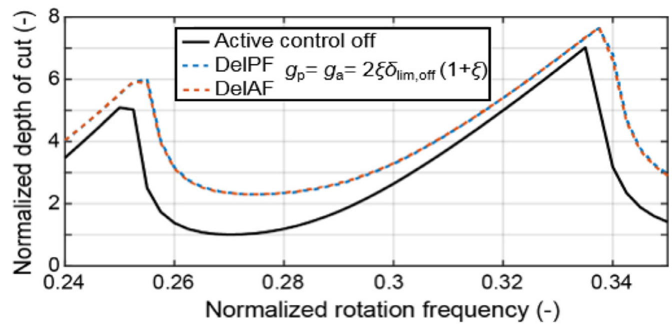

Fig. 4. Time domain simulation results of the improvement carried out with both DelPF and DelAF control strategies over fourth stability lobe ($n = 4$).

4. Hardware-in-the-loop experiments

Once the effectiveness of the proposed control law has been proved theoretically in an ideal environment, it has been applied in a hardware-in-the-loop (HIL) test bench. This system developed by the authors [28] reproduces experimentally on a simple mechanical structure any equivalent orthogonal cutting process where the regenerative chatter can appear. In this way, chatter suppression methods such as active

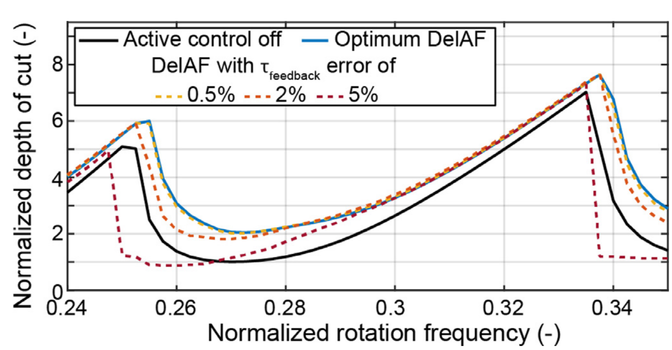

Fig. 5. Delay error sensitivity analysis with DelAF control law ($g_a = 2\xi\delta_{lim,off}(1+\xi)$, $n = 4$).

Table 1
Dynamic parameters of the structure.

Natural frequency ω_n (Hz)	Damping ξ (%)	Stiffness k (N/m)
177.8	1.4	43.24×10^6

Table 2
Dynamic parameters Data Physics V2 actuator.

Force capability $F_{act\,max}$	13.5 N
Suspension frequency $\omega_{n\,Act}$	13 Hz
Damping ratio ξ_{act}	0.9 %
Stroke	±1.25 mm
Actuator delay τ_{act}	650 µs
Bandwidth	20–500 Hz

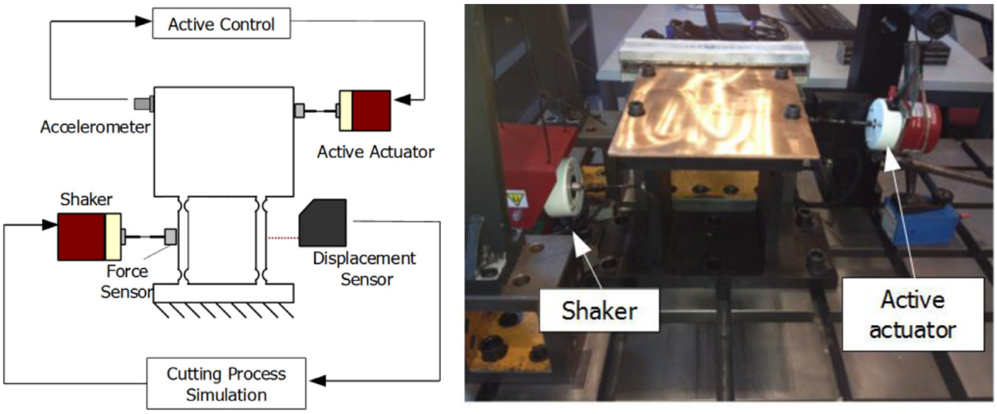

Fig. 6. Hardware-in-the-loop system with the active control loop [28].

control devices can be tested on a physical structure. Moreover, real problems such as mechatronic delays or electrical noise appear on the test bench and their influence on the results can be observed.

Hardware-in-the-loop systems have been widely employed in several industries [34–36]. However, in manufacturing, few studies have been conducted based on such simulators. Some authors proposed the construction of HIL systems for reproducing machining processes based on a cantilever beam [37–39], where active control methods were tested. The HIL system employed in this work is based on a linear flexure structure and the damping is provided by means of eddy currents, which introduces a pure linear viscous damping without changing other dynamic properties [28]. The cutting force is reproduced by a shaker (Table 2) hung on a small portal structure by elastomers, in which the time interval between the command sent to the shaker and the force generated is compensated [28]. The dynamic properties of the Hardware-in-the-Loop simulator are summarized in Table 1.

In order to test the control law presented in this paper, an accelerometer and another inertial actuator are added to the HIL system. A Data Physics V2 voice coil shaker (Table 2) has been hung on the opposite side of the structure, introducing the force in the most flexible zone (see Fig. 6). A collocated accelerometer has been introduced to measure the vibration and the Delayed Acceleration Feedback (DelAF) has been employed. The system delay due to the actuator delay τ_{act} and to the controller delay τ_c, has been measured to be 1.15 ms, although it can vary depending on the actuation frequency, due to the bandwidth of the actuator. Such delay is compensated as explained in Eq. (27).

The dSPACE DS1005 controller has been employed to perform the active control. In order to avoid the suspension frequency of the actuator and high frequency noise, filters have been introduced in the feedback loop (see Table 3). Finally, the commanded force generating current has been limited in order not to exceed the maximum force capability of the active actuator.

Fig. 7 shows the stability diagram obtained with the Delayed Acceleration Feedback by applying the optimal gain for each rotation frequency. It can be observed that similar results as the ones achieved by theoretical simulations are obtained, where the novel control strategy can improve the stability in all stability lobe areas. Fig. 8 shows a temporal force signal measurement performed by the HIL system for an

Table 3
Signal conditioning parameters for the employment of delayed acceleration feedback.

Feedback gain	G_{DelAF}	2.4 N/(m/s^2)
High pass filter	ω_{hp}	25 Hz
Low pass filter	ω_{lp}	1000 Hz
Saturation force	F_{sat}	±4.8 N
Sampling period	T_s	500 µs

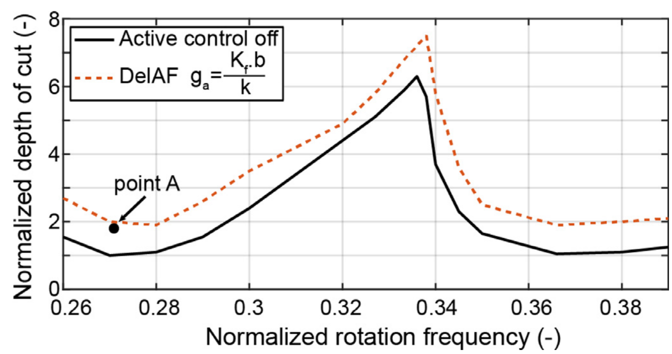

Fig. 7. Hardware-in-the-loop results with and without delayed acceleration feedback control law over fourth stability lobe ($n = 4$).

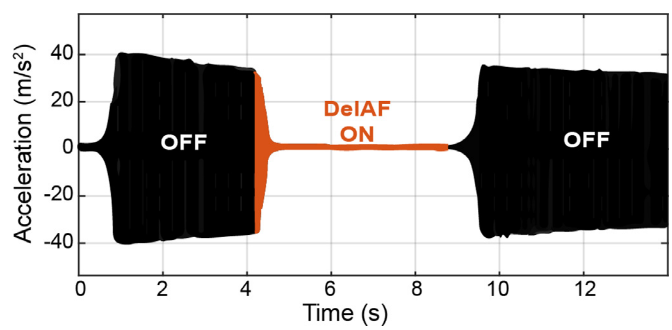

Fig. 8. Effect of the delayed acceleration feedback control on the chatter force of the HIL (Point A in Fig. 7).

originally unstable case (point A in Fig. 7). It can be clearly seen that when the actuator is switched on the chatter vibration is completely suppressed.

The delayed feedback control laws showed that it is possible to disturb the regenerative term with an active damper attached to the structure. The entire stability limit can be increased, so it is demonstrated that it is an adequate control law for orthogonal cutting cases where chatter occurs. Hence, it offers a new control option to solve chatter problems which will now be tested in real conditions.

5. Experimental turning tests

An experimental turning test has been performed in a SORALUCE vertical turning center (see Fig. 9). The machine can perform different types of machining processes, such as milling, turning or drilling. In the case of the turning operations, the machine has vibration problems when the cutting conditions defined in Table 4 are set, due to its flexibility

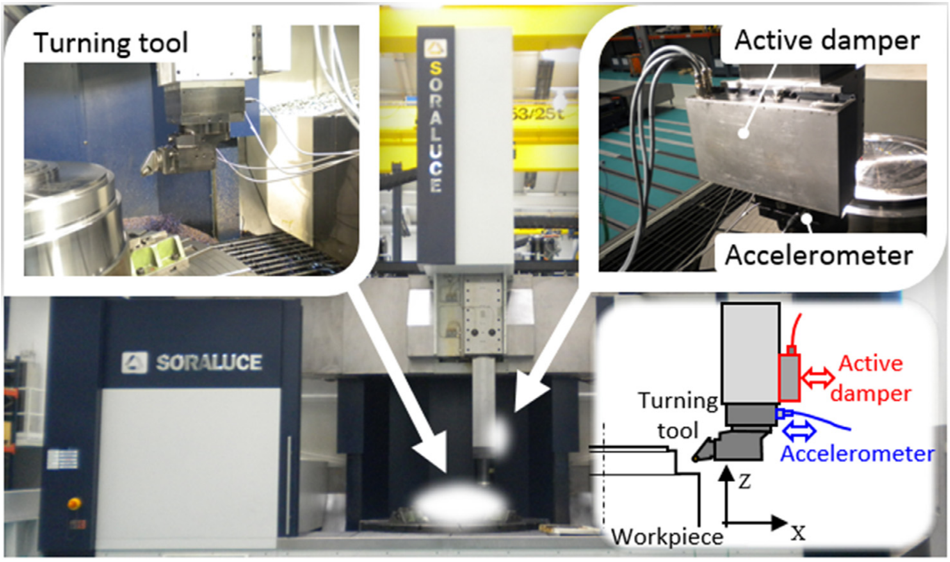

Fig. 9. SORALUCE vertical turning center.

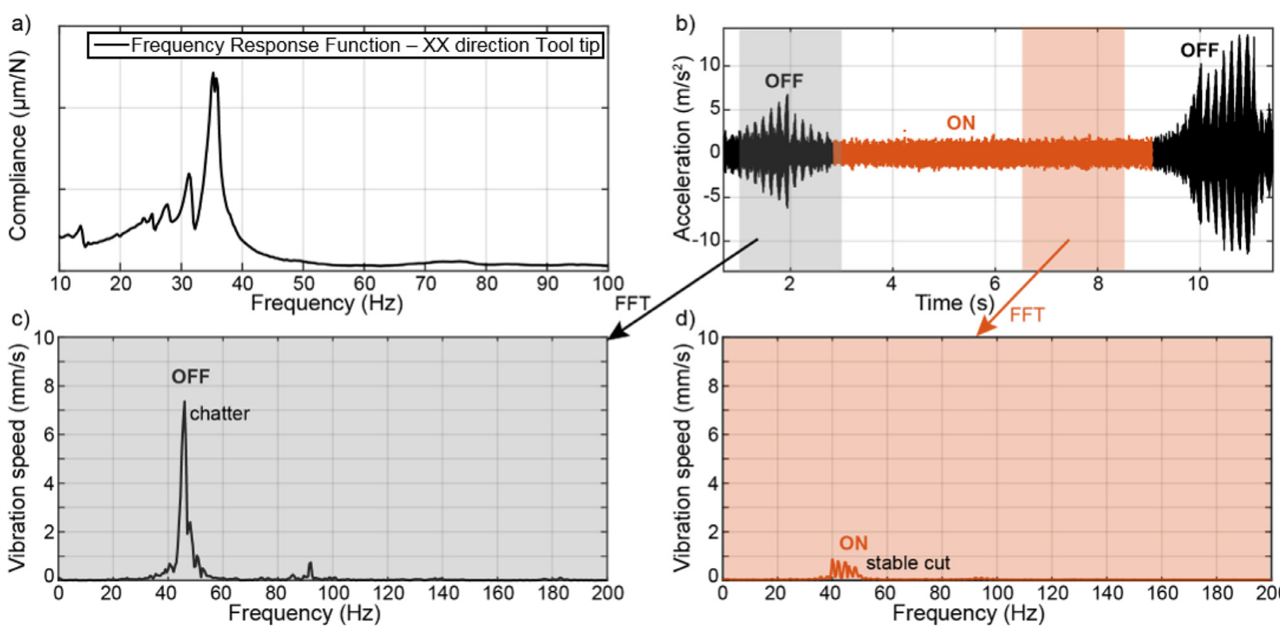

Fig. 10. (a) Dynamical response of the machine structure, (b) effect of the delayed acceleration feedback on the time domain acceleration signal on the tool, (c) and (d) vibration spectrum for the cutting conditions described by Table 4 without and with active control.

Table 4
Description of the cutting parameters.

Cutting tool	
Reference	C8-PRS CL-55080-32
Inserts reference	RCMX 320900 4225
Cutting conditions	
Spindle speed	135 rpm
Feed per rev	0.4 mm/rev
Workpiece diameter	950 mm
Workpiece material	F1140 Steel (C45)

Table 5
Description of the actuator and accelerometer.

Micromega ADD-2D-1kN	
Force capability F_{actMax}	1000 N
Suspension frequency ω_{nAct}	22.5 Hz
Damping ratio ξ_{act}	11%
Stroke	±5 mm
Actuator delay τ_{act}	700 ms
Bandwidth	30–200 Hz
IMI accelerometer (ICP® type)	
Sensitivity	100 mV/g
Measurement range	50 g
Frequency range	0.5–8000 Hz

in the cutting direction, mainly around 36 Hz (Fig. 10). Comparing to the lobes studied in the HIL, in this case the process conditions will be located at a higher lobe number ($n = 16$). Therefore, it is possible to have the influence of the process damping effect [40] and the lobes obtained in HIL are not totally analogous.

To test the proposed control law, a Micromega ADD-2D-1 kN electromagnetic actuator (Table 5) and a collocated low-cost IMI industrial

Table 6
Description of the control parameters.

Control gain	G	$150\,\mathrm{N/(m/s^2)}$
High pass filter	ω_{hp}	10 Hz
Low pass filter	ω_{lp}	100 Hz
Sampling period	T_s	500 µs

accelerometer (Table 5) have been located on the ram of the machine, close to the tool (see Fig. 9). Although the inertial actuator is a biaxial damper, in these tests, the force has only been introduced in the cutting direction (X direction), since it is the most flexible direction related to the critical mode at 36 Hz (see Fig. 10a), and coincides with the chip thickness direction. The control has been implemented on the dSPACE DS1005 controller with a sampling period (T_s) of 500 µs, which is high enough for the targeted frequencies (40 Hz). Delayed Acceleration Feedback control law has been applied with the control parameters defined in Table 6.

Fig. 10b shows the stability improvement carried out by this control system for a certain depth of cut that is not detailed in order to honor confidentiality agreements. The figure shows the considerable reduction of the vibration level and the removal of the chatter peak is clearly observed in the spectrum of each portion of the signal (see Fig. 10c and d). This cutting test proves that the Delayed Acceleration Feedback (DelAF) can be implemented in industrial conditions to suppress chatter vibrations. The combination of the different scenarios in the tests carried out in the HIL and the turning machine shows that the delayed feedback can be applied in any condition and relative location in the stability diagram.

6. Conclusions

This paper has presented a control law for active control devices to suppress chatter vibrations. The novelty of this feedback is that, as opposed to the classical feedback law, the aim is not to change the dynamical behavior of the structure but to reduce the regenerative effect. The reduction of the regenerative effect is achieved by the introduction of the revolution period delay over the vibration measurement feedback. In this way, a force negatively proportional to the previous vibration is fed back, and the reduction of the overlap factor between successive waves is virtually obtained. Since position and acceleration are counter phase signals, both acceleration and position can be employed for the delayed feedback.

The revolution period is usually higher than the delays existing on the controller and actuator; therefore, the proposed control law offers the possibility of compensating these delays, which is an improvement compared to other classical control laws.

The proposed control law is validated for orthogonal cutting processes. Theoretical simulations on ideal environments prove that any case can be stabilized if the optimal gain is employed and if neither force saturation nor bandwidth limitation exist in the force application. The control strategy has been analyzed in frequency domain, and when a fixed gain is applied only a range of depths of cut can be stabilized. In cases where the gain exceeds the optimal gain, even originally stable cases could be destabilized. However, the force required by the optimal gain to stabilize all cutting conditions, which is high in machining processes, is not achievable in real processes. Therefore, under most conditions, only a reduction of the regenerative effect is reached.

This effect has been observed on a hardware-in-the-loop, where a real actuator is used. Nevertheless, the improvement of the stability is clearly observed all over the frequency range. Finally, an experimental cutting test has been performed in a vertical turning center, where the improvement carried out by the control law is demonstrated. Further work should address the performance of this control law for the milling process.

Declaration of Competing Interest

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

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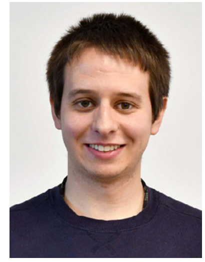

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