Low Frequency Chatter Suppression using an Inertial Actuator

Aitor Bilbao-Guillerna, Ion Azpeitia, Stanislas Luyckx, Nicolas Loix, Jokin Munoa

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A. Bilbao[1], I. Azpeitia[2], S. Luyckx[3], N. Loix[3] and J. Muñoa[1]
[1]Ideko-IK4, Elgoibar, Gipuzkoa, Basque Country, Spain
[2]Danobat Railway Systems, Gipuzkoa, Basque Country, Spain
[3]Micromega Dynamics, Fernelmont, Belgium

Abstract
It is well known that inertial actuators can remove regenerative chatter in many milling processes. They are useful when active control techniques are needed and the use of passive dampers is not enough. Moreover, complex control laws can easily be implemented leading to large improvements. However, the main limitation of inertial actuators is that they can only be used in a particular range of frequencies. This limitation is due to a low frequency suspension mode, which only allows the actuator to work at larger frequencies than this suspension mode. This is an important drawback in many big milling machines where chatter appears close to this low frequency. This paper deals with this problem and shows a possible solution to avoid this limitation from a control point of view. Some real milling tests will be presented showing the usefulness of the technique.

Keywords:
Active damping, Chatter, Inertial actuator

1 INTRODUCTION

Nowadays, self-excited vibrations or regenerative chatter is one of the main limitations of milling processes. Machine tool chatter vibrations avoid obtaining the required accuracy in workpiece, reducing the lifetime of the tool and the mechanical components of the machine.

Probably one of the most used solutions to increase the stability margin is by adding a tuned passive vibration absorber to the structure [1,2]. This technique is not new and it has been widely used since it was proposed in [3]. However, a passive absorber is not feasible in many machining processes where the dynamics of the system change during the process and then an active damper is needed [4-7]. Moreover, it is common to observe a non linear behaviour in passive absorbers making more difficult an accurate tuning of it.

Another possibility is to include an inertial actuator attached to structure to be controlled. An inertial actuator can easily be seen as a device producing a reaction force on the host structure according to a designed control law [8,9]. For this reason, inertial actuators have been used to reduce vibrations in machining operations. The active damping is achieved by implementing an external control law. The vibration is measured on the structure and then the designed control law generates a control signal to the actuator. The main drawback of inertial actuators is that they usually possess a suspension mode at low frequencies. This means that actuators only behave as a linear force generator beyond its natural frequency. This fact could be a problem in many big machines where main modes appear at the same low frequency range. Depending on the materials used in their construction, a non-linear behaviour could be seen in many actuators as well.

This paper deals with this problem and shows a possible solution to this limitation. We are proposing an easy solution from a control point of view. Since the control law is implemented by software there is no need to change to a more appropriate actuator with a larger linear range. Many other solutions could be found proposed by other authors. In [10] an internal damping is proposed to increase the stability of the controller. Similar results can be see if force feedback is implemented [11]. The main idea of this type of controllers consists of implementing a second control law dealing with the dynamics of the actuator. However, this is not always possible since some of the signals needed for the internal feedback could not be available for measuring. The use of a Kalman filter or a similar state-space observer could make an estimation of this signal available, but this is not always an easy solution. Another possibility is to implement more complex control laws. A Virtual Passive Absorber (VPA) controller is a good option since it does not excite the dynamics of the actuator [5,12,13], but it is not very useful if the vibration chatter is close to the low frequency dynamics of the actuator. Another possible controller consists of implementing a displacement feedback in parallel with the main control law. A displacement feedback is able to affect the stiffness of the mode to be controlled and change its location into the linear range of the actuator. Then main control law can be applied.

In this paper we are including a compensation filter in the control-loop. This kind of filters can increase the working range of the actuator and avoid the suspension mode limitation. The objective of this kind of filters is to modify the dynamics of the actuator seen by the controller such that it does not interact with the structure dynamics anymore. This approach leads to good results. However, it relies on a good understanding of the actuator dynamics and, obviously, there are always non-modelled dynamics in the model of the actuator. The more accurate the model is, a better closed-loop performance is obtained.

Then a second drawback should be taken into account. Some inertial actuators possess a non-linear behaviour. This means that it is not possible to find a unique linear model describing the behaviour of the actuator for all the possible operation points. A set of different operation points is considered. Then a different transfer function is calculated for each operation point. Each of this transfer functions is calculated via linearisation of the behaviour of the actuator around the operation point. Then a compensation filter is associated to each transfer function. Finally, once the dynamics of the actuator are moved to lower frequencies, then a controller can be implemented to remove the undesired vibration. In this paper a direct velocity feedback has been used. This kind of control law is widely used on inertial actuators due to its simplicity and leads to good results. There is plenty of information about this control explaining its advantages and disadvantages [5,14].

This paper is organized as follows. First, the concept and a general definition of an inertial actuator is explained. Then the actuator used for the work is presented. The next section describes the control strategy, including the choice of the compensation filter and the direct velocity

feedback control law. Some of the performed experiments will be presented showing the usefulness of the solution proposed in the paper. These tests include some real milling tests where the chatter vibration is removed by the actuator. Finally, conclusions and future work will 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 (Figure 1). According to an applied voltage V_{in} the reaction mass is excited and it induces a force F_a on the supporting base. The dynamics of an inertial actuator can be modelled as the following transfer function

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

where ω_a is the natural frequency, ζ_a the damping ratio and g_a is the gain of the actuator. The actuator behaves as an ideal linear force generator beyond a determined frequency ω_p (Figure 2). 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.

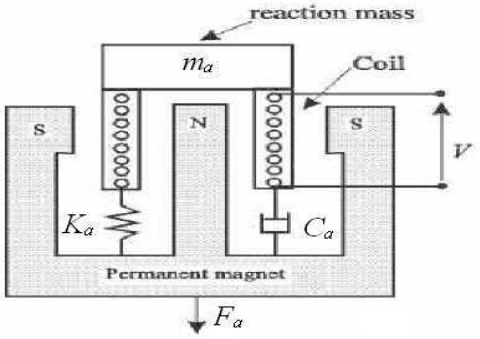

Figure 1: Inertial actuator

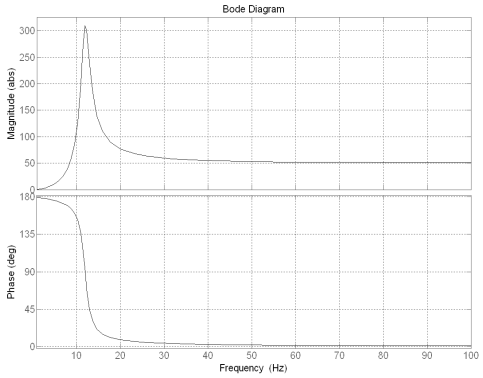

Figure 2: Bode plot of an ideal inertial actuator

In this paper the actuator used to perform the practical applications is a model of Micromega Dynamics (http://www.micromega-dynamics.com) (Figure 3). This actuator is capable of applying an up to 1000 N force on the host structure. Moreover, it is designed to work in two perpendicular axes. This is an important characteristic about this actuator, since most of them are designed to work in only one direction. In order to identify the behaviour of the actuator, it is mounted on a Kistler dynamometer plate and excited by a chirp voltage. Different amplitudes of the chirp signal are considered and the obtained frequency response functions (FRF) show a non-linear behaviour of the actuator (Figure 4). The natural frequency of the actuator depends on the amplitude of the chirp signal. Larger values of this amplitude lead to lower frequencies of the suspension mode. Most of the times this is not important if the vibration to be damped is in the linear range of the actuator. However, this non-linearity could affect the performance of the controller if the vibration to be remove is close to this suspension mode.

Figure 3: Inertial actuator from Micromega Dynamics

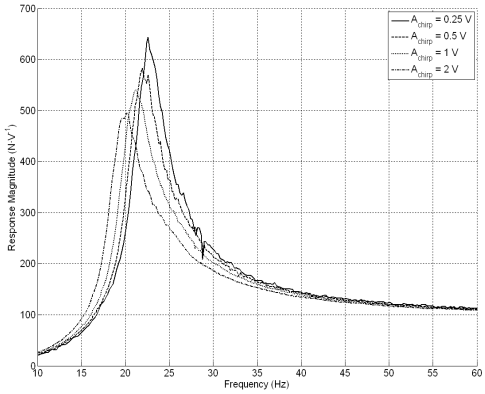

Figure 4: FRFs for different amplitudes of the excitation input

3 CONTROL LAW

3.1 Direct Velocity Feedback

In this paper a direct velocity feedback is implemented to introduce some damping in the process. The transfer function of the controller can be written as

$$V_c(s)=-g_v\dot{X}(s)=-g_v\cdot s\cdot X(s) \qquad (2)$$

where $V_c(s)$ is the applied control input to the inertial actuator, $s\cdot X(s)$ 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

$$F_a(s)=g_a\cdot V_c(s)=-g_a\cdot g_v\cdot s\cdot X(s) \qquad (3)$$

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

$$1+g_v\cdot g_a\cdot s\cdot G(s)=0 \qquad (4)$$

where $G(s)$ is the transfer function describing the structure to be controlled. Now considering a structure with a unique mode, equation (4) can be written as

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

$$s^2+s(g_v\cdot g_a\cdot g+2\omega_n\zeta_n)+\omega_n^2=0 \qquad (5)$$

From (5) the damping of the closed loop poles can be augmented increasing the value of the feedback gain g_v. However, the effect of the inertial actuator should taken into account. The dynamics of the inertial actuator can be studied including its transfer function in previous characteristic equation

$$1+g_v \cdot s \cdot \frac{g_a s^2}{s^2+2\zeta_a \omega_a s+\omega_a^2} G(s)=0 \qquad (6)$$

In this case the low frequency dynamics of the actuator should be considered to guarantee closed-loop stability. In Figure 5 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 a linear model of the structure was available. 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.

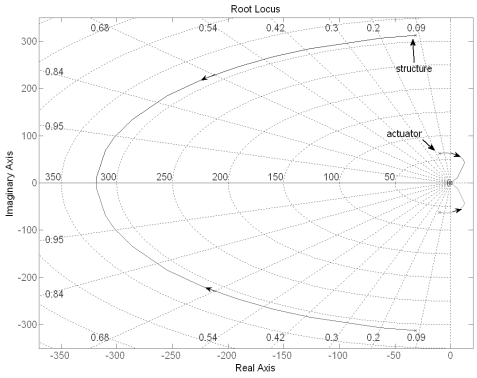

Figure 5: Root locus analysis including the dynamics of the actuator

3.2 Compensation Filter

When the vibration to be removed is close to the suspension mode of the actuator, then a controller based on just a direct velocity feedback could not be enough to achieve a good result. The optimum control is achieved when the applied force by the actuator possesses the opposite phase than the velocity of the structure. The phase of the voltage applied to the actuator is always close to the correct one since it is generated by the controller. If the vibration to be damped is in the linear range of the actuator, then the applied force has similar phase to the voltage input and the desired effect is achieved. However, this does not happen if the vibration is close to the suspension mode. Then there is not certainty that the applied force has the correct phase.

Different solutions can be applied to avoid this limitation. One could deal with the structure and try to move the main mode to higher frequencies, where the actuator behaves as a linear force generator. A very simple strategy is to include a position feedback controller. The effect of this controller is to move the frequency of the mode by changing its stiffness. Then, once both modes are sufficiently separated, one can apply the velocity feedback and expect some damping. The disadvantage of this strategy is that some amount of the force that the actuator can apply is wasted moving the mode to another frequency and full capacity can not be used in the damping process.

Another solution is to increment the linear range of the actuator moving its suspension frequency to lower frequencies. This can easily be done including a compensation filter in the control-loop. A compensation filter modifies the dynamics of the actuator seen by the controller such that it does not interact with the structure dynamics anymore. Of course it relies on a good understanding of the actuator dynamics and in practice there is always a difference between the real behaviour and the linear model used to define the filter. This means that a perfect actuator dynamics is never achieved. However, most of the times this technique leads to good results and there is always some improvement.

One could think that it is not always easy to find a linear model describing the behaviour of the actuator. In previous section, it was shown that the actuator presents a non-linear behaviour. Then it is not possible to find a linear model describing its behaviour for any condition. In this paper, we are proposing to linearise the behaviour around the operation point. This technique requires an a priori knowledge of the process. However, this is a feasible condition for many milling processes, where it is common to know the value of the different signals of the process. Then a set of compensation filters are considered. Each one calculated from a different operation point in such a way that the dynamics of the actuator seen by the controller is good enough for the control objectives.

4 EXPERIMENTAL RESULTS

4.1 Impact Tests

In this section the experimental results are presented. The different tests are performed using a Soraluce milling machine The actuator should be mounted as much as closer to the tool, where the cutting process takes place. The actuator can not be implemented in the spindle head due to its size, so it is attached under the ram of the machine (Figure 6). Moreover, it was designed to compensate the effect of gravity on the moving mass when it is mounted in that position. Then the part of the ram closer to the spindle head, where the displacement is larger, is the best option.

Figure 6: The actuator mounted under the ram

First a dynamical analysis is performed hitting the structure with an impact hammer. Then the response on the structure is measured with an accelerometer. The direct velocity control law is used. Different outputs for the ram are considered and both horizontal and vertical axes are tested. Figure 7 shows the frequency response functions (FRF) for different values of the feedback gain. The output of the ram is 1100 mm and the horizontal axis is first excited. The main mode of the structure can

be damped by increasing the feedback gain. On the other hand, both actuator mode (22Hz) and machine suspension (15Hz) are easily excited. This effect limits the effect of the controller and prevents achieving a good performance. Note that g_v is a constant that multiplies the velocity measured on the structure and the result is a voltage signal directly applied to the actuator, so its units are $V \cdot m^{-1} \cdot s$.

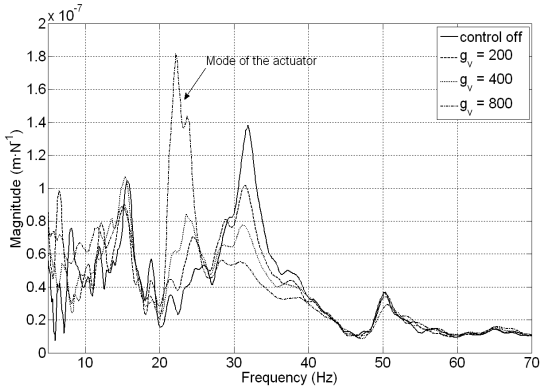

Figure 7: FRF for different values of the feedback gain

In order to improve this result a compensation filter (C_1) is included in the control loop. The filter is designed to locate the suspension mode of the actuator at 10 Hz. In this case, since an impact is used, the linearisation of the behaviour of the actuator for low levels of the excitation has been used. Then different values of the feedback gain are considered and the FRFs are displayed in Figure 8. The mode of the actuator is not excited, but it appears an oscillation at 15 Hz. This is due to the fact, while the new actuator dynamics is now able to damp the main machine mode, it is not yet good enough to tackle the other mode of the machine, which is close to the new dynamics of the actuator.

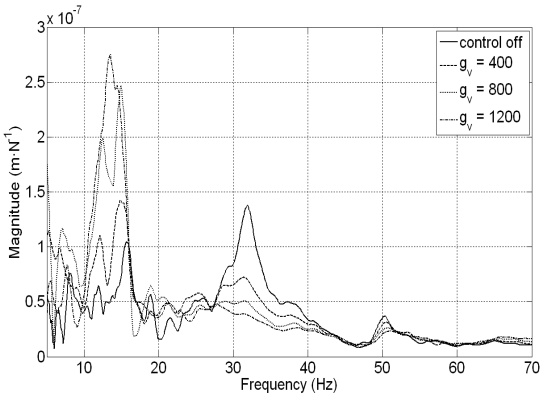

Figure 8: FRFs for different feedback gains using the compensator filter C_1

It is of course possible to improve the compensation filter in order to be able to address also the machine mode at 15 Hz. The compensation filter is chosen to locate now the suspension mode of the actuator at 5 Hz. Figure 9 shows the FRF for the new compensation filter (C_2). Now the oscillation appears at lower frequencies, but with this compensator larger values of the feedback gain can be used before the instability appears. Moreover, the mode at 15 Hz is not excited.

It is also possible to consider that there is no need to damp the machine mode at 15 Hz since it has little influence on the chatter vibration. Then the original compensation filter can be used if a “Notch filter” is added at 15Hz to prevent the control to address this machine mode.

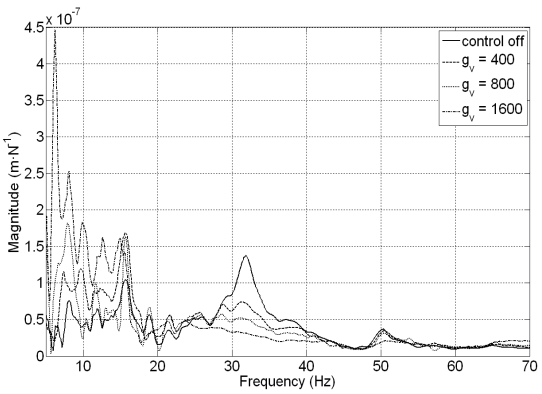

Figure 9: FRFs for different feedback gains using the compensator filter C_2

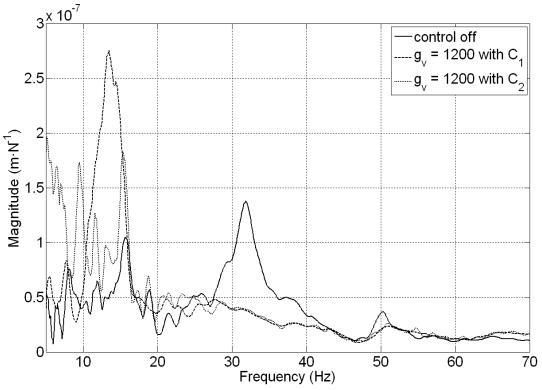

Figure 10: Comparing both compensation filters C_1 and C_2 with $g_v = 1200$

4.2 Milling tests

Finally, some milling test are performed in the same milling machine (Figure 11). The cutting conditions are displayed in Table 1, while the cutting tool is described in Table 2. Note that the experiments will be performed for two different values of F_z. The compensation filter is chosen considering that the actuator needs to apply a large force to cancel the chatter vibrations on the ram. The results without the compensation filter are not shown since the actuator was easily excited and large values of the feedback gain could not be used.

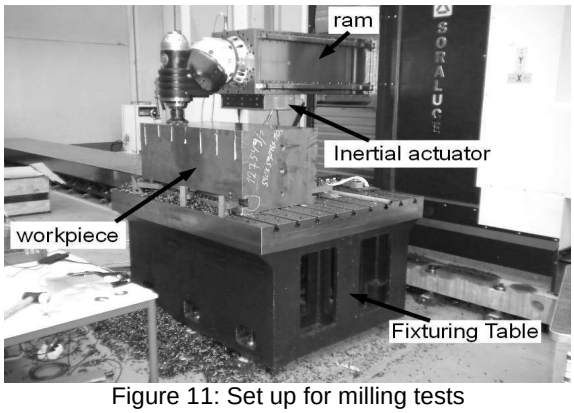

Figure 11: Set up for milling tests

Cutting conditions	
spindle speed	570 rpm
F_z (feed per tooth)	0.1-0.2 mm/z
a_e (radial immersion)	100 mm (downmilling)
workpiece material	Steel F1140

Table 1: Cutting Conditions

Cutting tool	
Reference	SANDVIK R245-125Q40-12M
D (diameter)	125 mm
Z (inserts)	8
k (lead angle)	45 degrees

Table 2: Cutting Tool

First F_z = 0.2 mm/z is chosen. Figure 12 shows the stability conditions for different values of the ram output and the depth of cut. The real values of the depth of cut are not given due to confidentiality concerns and all the values are compared to the minimum value of the depth of cut leading to an unstable process (100%). For lower values of a_p, the process is always stable and the actuator is not needed to avoid the appearance of chatter. However, larger values than 100-125% lead to unstable conditions. Then the controller is connected and the stability margin is doubled for almost all the ram outputs. This means that the productivity of the machine is increased as well. The MRR (material removal rate) has been doubled. For very large values of a_p the actuator is not able to remove completely the chatter vibration. However, there is always some improvement and the vibration is always reduced when the controller is connected.

The cutting conditions are changed reducing the feed per tooth to 0.1 mm/z. In this case the controller actuator is able to remove the chatter vibrations for the maximum depth of cut and all the ram outputs (Figure 13). This means that vibrations do not limit the productivity and any condition can be used. This is always the best result when active control is applied in milling processes.

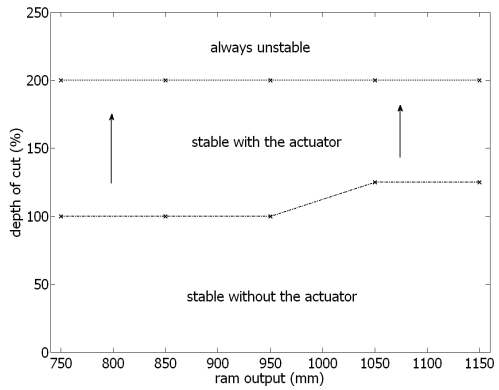

Figure 12: stability conditions for different ram outputs and depths of cut when F_z = 0.2 mm/z

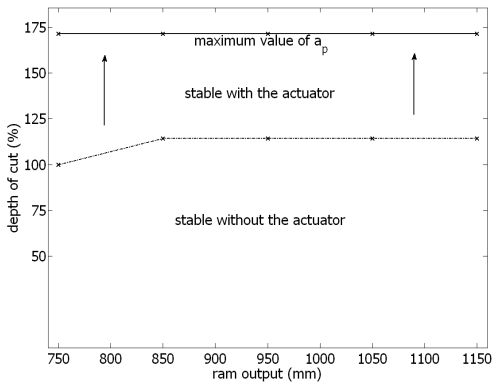

Figure 13: stability conditions for different ram outputs and depths of cut when F_z = 0.1 mm/z

One could ask why the stability margin could not be increased until the maximum value when F_z = 0.2 mm/z. The most common limitations are the dynamics of the actuator or a saturation of the force applied by the actuator. In this case, the limitation comes from the fixturing table used in the process (Figure 11). A new test is performed with a_p larger than 200%. According to Figure 12 this condition leads to an unstable process. In Figure 14 the acceleration measured during the test is displayed. First the actuator is switched off resulting to an unstable process. After a few seconds the actuator is connected with g_v = 2200 $V \cdot m^{-1} \cdot s$ and the vibration on the ram is reduced. A frequency analysis is performed in order to obtain more information about the process (Figure 16). When no controller is applied then a chatter vibration appears at 34.5 Hz. Clearly this is due to the structure of the milling machine since its main mode is close to this frequency. Then when the controller is connected, this vibration is removed, but it appears another at 46 Hz, which results to be close to the main mode of the fixturing table (48Hz). This effect is more evident if the acceleration measured on the fixture table is analysed. Figure 15 shows the acceleration of it during the same test. One could see how the signal is increased when the controller is applied. Moreover, a frequency analysis of the velocity (Figure 17) leads to the same conclusion. The main conclusion from this result is that the machine is no longer the most important limitation; the fixturing table is now the main limitation. Different solutions can be implemented to deal with this drawback. Changing the fixturing table is the most evident one, but similar control techniques can be applied as well. A passive absorber or a second actuator could be attached to the fixturing table.

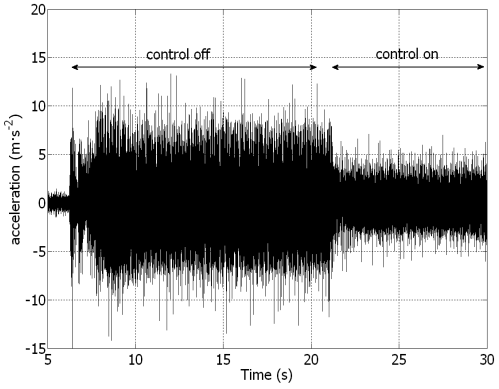

Figure 14: Acceleration measured on the ram with a_p larger than 200%

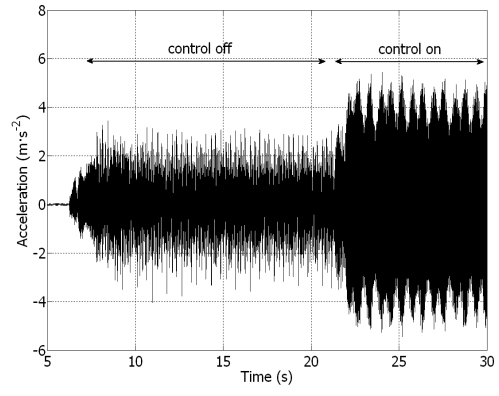

Figure 15: Acceleration measured on the fixturing table with a_p larger than 200%

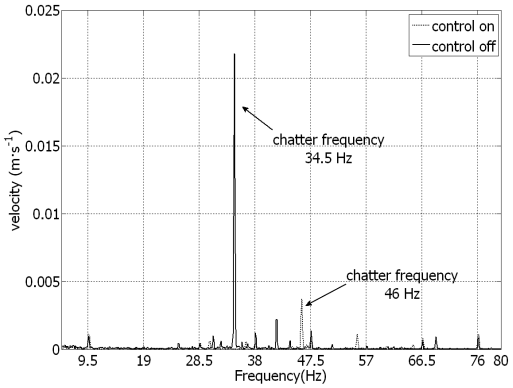

Figure 16: Frequency analysis of the velocity measured on the ram with a_p larger than 200%

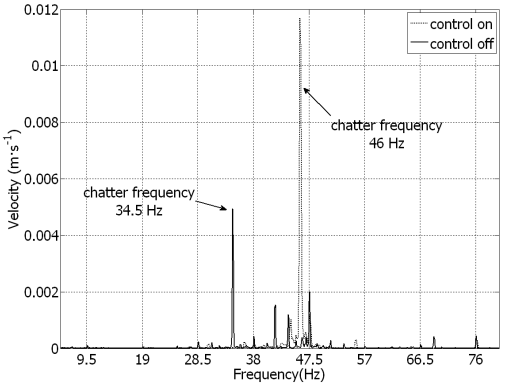

Figure 17: Frequency analysis of the velocity measured on the fixturing table with a_p larger than 200%

5 CONCLUSIONS AND FUTURE WORK

In this paper an active control using an inertial actuator for low frequency chatter suppression in milling processes has been presented. The appearance of low frequency chatter limits the performance of most inertial actuator due to the suspension mode. An easy solution from a control point of view has been used leading to a lower sensitivity to chatter instabilities. The inclusion of a compensation filter in the control loop increases the linear range of the actuator moving the suspension mode to lower frequencies. Moreover, the non-linear behaviour of the actuator has been overcome by considering a set of possible linearisation of the actuator around different operating points.

Results show the necessity of an a priori knowledge of the conditions for an appropriate choice of the compensation filter. The performed milling tests show that the productivity and the stability margin can be increased by the actuator even if the chatter frequency is close to the suspension mode of the actuator. The machine was found to be no more the main limitation for the appearance of chatter.

On the other hand, the fixturing table used for the tests introduces an important limitation. The future work will be focused on the solution to this drawback. The main idea consists of adding an additional absorber to the fixturing table. This absorber could be a passive absorber or a second inertial actuator attached to the fixturing table.

6 ACKNOWLEDGEMENTS

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