Methods of implementation of a restricted State Control on HSC milling machines

Valérie Briquez[1] - Jean-Paul Sandoz[2] - Leonello Zaquini[1]

[1] HE-ARC - University of Applied Sciences - Laboratoire de Machines-Outils (LMO) – Av. De l'Hotel de Ville 7 – CH 2400 Le Locle (Switzerland) – leonello.zaquini@he-arc.ch

[2] HE-ARC - University of Applied Sciences - Signal Processing Laboratory – Av. De l'Hotel de Ville 7 – CH 2400 Le Locle (Switzerland) – jean-paul.sandoz@he-arc.ch

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Abstract.
Machines for HSC commonly adopt low friction guideway and, in particular, ball or roller linear guideways.
These elements have the advantage of very low friction factors (≤0,002) and in consequence also the fluctuation of the friction is low. This way these elements offer very high quality of the machined surfaces. At the opposite these elements have low dumping factors that reduce the margin of stability and oblige a reduction of the gain of the position loop (commonly called Kv).
The theory and the experimental tests show that additional state variables could permit higher performances of the control of an axis. In particular the feed back of the acceleration of the slider, relative to the fixed structure of the machine, permits to dump the axis, this way overcoming the inconvenient of the reduced dumping of the guideways.

The paper lists the methods adopted, during several years of experiments in realising this control structure. In particular the paper explains the following 3 methods:

a) The relative acceleration is considered as not observable; therefore it will be estimated through an observer.
b) The relative acceleration will be measured by means of additional sensors (accelerometers).
c) The relative acceleration will be measured through the double derivation of the position.

The paper shows the results of practical cases of implementation and the possibility to allow a significant increase of the proportional gain of the position loop.

Introduction

Ball or roller linear guideways and recirculating ball-screws are widely adopted in cinematic chains of High Speed Cutting (HSC) machines.
The structure of these cinematic chains allows very low friction and (in consequence) also a low noise of the friction (sum of the periodic friction variation and of the stochastic variation).

Just to offer an idea of these values, for a single carriage suitable for a medium machine we find friction forces of 14 N, due to the pre-charge, and a noise < 6N [1].
The friction factors are very low: <0,002.

In consequence these structures allow a good continuity of the motion.
At the opposite, the reduced friction factor and the dumping coefficient, reduces the stability. This way the efforts in increasing the rigidity and the own frequencies of the cinematic chain are reduced or vanished. Low values of the gain (and in particular the proportional gain of the position loop) are compulsory.

The possibility to overcome the inconvenient and increase the proportional gain of the position loop (Kv) through a more affective control technique is matter of studies and researches, since years. In particular since the HSC cutting machines entered the market.

Basic principles of operation.
Commonly, the regulation system is based on three overlapping loops manage, which controls the motor to ensure a displacement with desired speed and selected acceleration.

The aim is to reduce the path deviation and to obtain an accurate and fast positioning of the axis. The controlled state variable is the relative position of the axis. Additional state variables (relative velocity and acceleration) could permit higher performances of the control. In particular, the feedback of the acceleration of the moving part, relative to the fixed structure of the machine, permits to dump the axis, this way overcoming the inconvenient of the reduced dumping of the guideways.

Assuming that the behaviour of the axis (when the position loop is open) is properly modelled as a second order system, the model would be represented by the continuous black lines of the figure 1 representing the transfer function of equation:

$$T(s) = \frac{\omega_n^2}{s^2 + 2\zeta\omega_n s + \omega_n^2} \tag{1}$$

If we feedback the additional state values, the velocity and the acceleration, the structure of the control system will be as plotted in Fig.1, with the additional loops represented in dot red lines.

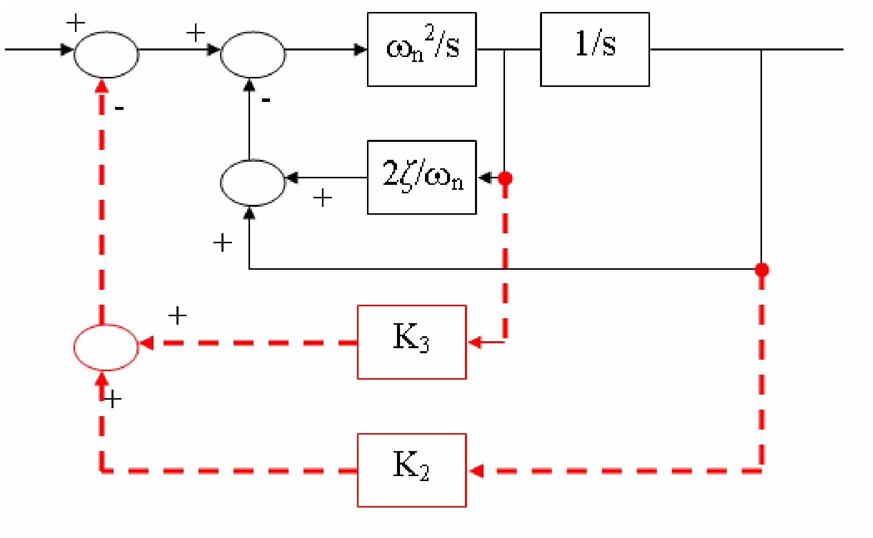


Fig.1 – Second order system (in black continuous lines) and the additional control loops of velocity, through K2, and acceleration through K3 (red dot lines):

The new transfer function (T(s)), will be the following:

$$T(s)=\frac{\omega_n^2}{s^2+(2\zeta\omega_n+K_3\omega_n^2)s+\omega_n^2(1+K_2)}=\frac{1}{\left(1+K_2\right)\left[\frac{s^2}{\left(1+K_2\right)\omega_n^2}+\frac{2\zeta\omega_n+K_3\omega_n^2}{(1+K_2)\omega_n^2}s+1\right]} \quad (2)$$

The new system will act just as a second order system, but having new characteristics:

Attenuation	$\frac{1}{(1+K_2)}$
Natural angular frequency	$\omega_{nG}=\omega_n\sqrt{1+K_2};$
Dumping factor	$\zeta_G=\frac{1}{\sqrt{1+K_2}}(\frac{K_3\omega_n}{2}+\zeta)$

Table 1 – Attenuation, resonant frequency and dumping of the new system, in relation with the characteristics of the original system and the gains of the state variables.

Therefore it will be possible to locate natural angular frequency and damping by varying the gains of the feed backs of the state variables: feed and acceleration..
In particular if the only state variable will be the acceleration, we will not have a displacement of the natural angular frequency, no new attenuation but a new damping factor:

In this case ($K_2=0$, $K_3\neq 0$) attenuation and natural frequency are unchanged and the new system reacts as a new second order system with a new dumping factor, ζ_G , as per the relation:

$$\zeta_G=(\frac{K_3\omega_n}{2}+\zeta) \quad (3)$$

Just as matter of example, we consider a second order system having; ω_n = 251 [rad/s] and ζ = 0,22. (which are the characteristics of the first oscillator present on a machine we tested. We add to the system the feed back of the acceleration (K_2 = 0 and K_3 = 0.0022). The step responses are these shown in Fig. 2.

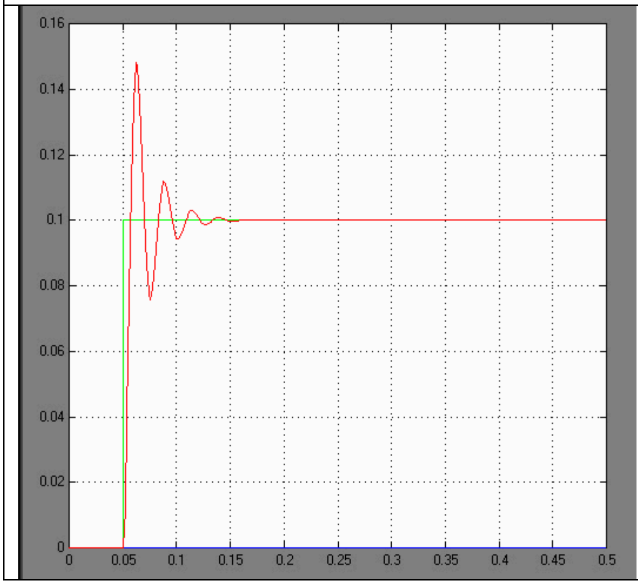
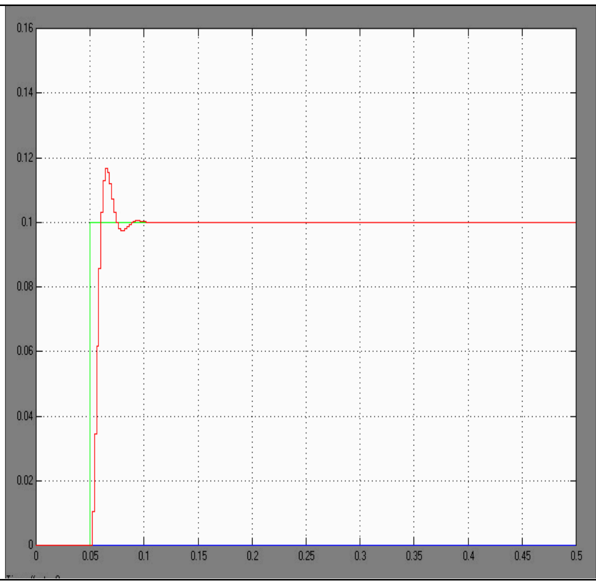
A - Without feed back of acceleration	B – With state space feed back restricted to the acceleration.
	

Fig. 2 – Step response of the speed of the original second order oscillator. Case A, response of the original second order system. Case B, a feed back of the acceleration through a and of the same oscillator with a space control restricted to the acceleration.

For the mechanical transfer chain, we theoretically have to feedback two state space information's (i.e. velocity and acceleration) for each oscillating mechanical element we want to take into consideration. In our practical experiments, we limited the controller to the feedback of the acceleration and to a single oscillator. We see that this “restricted state control” is capable to actively damp a (k_V-limiting) mechanical oscillation inside the mechanical transmission chain.
With these limitations, the general structure of the control of an axis of a machine tool is presented in Fig. 3.

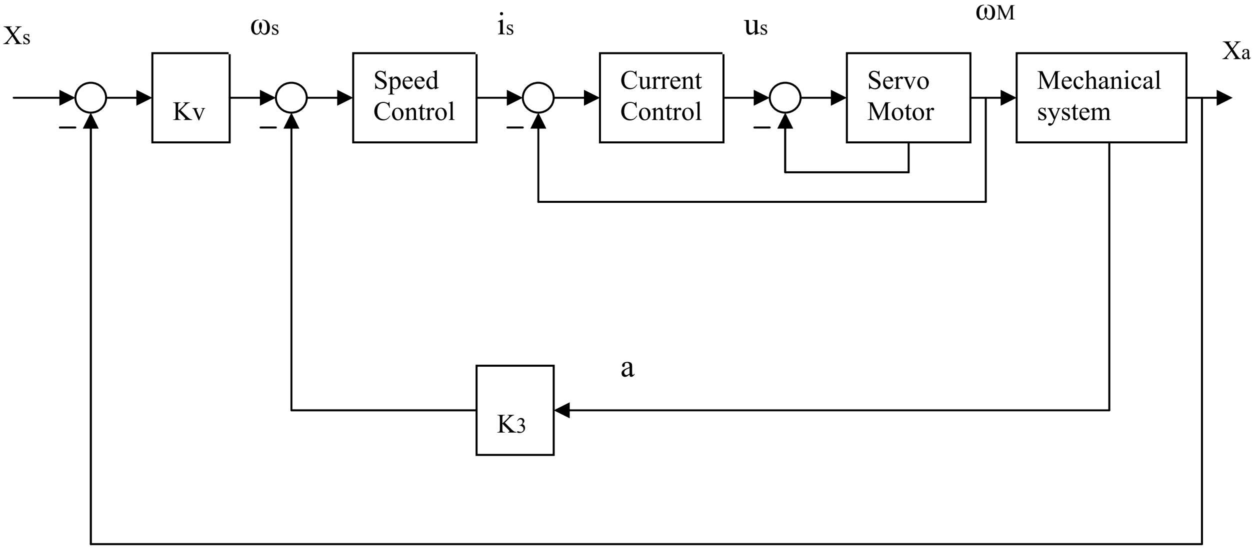


Fig. 3 - General structure of the control of the axis of a machine tool with the state control restricted to the relative acceleration (a = $\ddot{X}_a$).

Practical constraints.
In the practical implementation of the method on a machine tool, the main problem arises from the fact that the acceleration to be measured it that of the sliding part of the machine axis, relative to the fixed structure of the machine. Thus, we need the relative acceleration and not the inertial one. The technique used to measure or estimate the relative acceleration makes the main difference between the different methods of application of the restricted state control to the machine tools.

Here we list the techniques adopted, during several years of experiments, and in particular the following three:

a) The relative acceleration is estimated through an observer.
b) The relative acceleration is measured by means of additional sensors (accelerometers).
c) The relative acceleration is computed through the double derivation of the position.

An additional constraint, which is always present no matter what technique is adopted, is due to the sampling time.
Of course the damping capability of the state space controller is strictly limited by the sampling time T of the digital controller, where the algorithms are implemented. For a good active damping we should sample at least 10 points of a single period of a particular mechanical oscillation with resonant f_{0Me}, which means:

$$f_{0Me} < \frac{1}{10T} \tag{4}$$

Additionally, a mechanical oscillation with a resonant frequency f_{0Me} ,can be effectively damped only if the bandwidth of the following PI-speed control loop (f_{3dBSC}, 3dB frequency of the speed control loop) is high enough. This means:

$$f_{0Me} < f_{3dBSC} \tag{5}$$

Therefore it only makes sense to feed back the state information from mechanical oscillators, when conditions (4) and (5) are fulfilled.
Now, the sampling time T is varying as per the location, where the state space controller is implemented. For digital drives we have the possibility to implement the state space control algorithms either together with the position controller (sampling time T_{PC}) or with the speed controller (sampling time T_{SC}). In case that both numbers T_{PC} and T_{SC} are equal, it doesn't matter where the state space control algorithms are implemented. Nevertheless T_{SC} is in the range of 50..500 µs, while T_{PC} reaches 1..4 ms. Obviously, in this case it is better to choose the speed controller level because of the higher sampling rate.
In any case the sampling time is not the only limitation, even in case of very little sampling time (i.e. < 100 µs) we will be limited by the bandwidth of the speed control (relation (5)).

In case of common drives on the market, the access to the speed loop is, in general, not open. For this reason our experiments operate at the level of the position controller (T_{PC}). Nevertheless, the resonant frequency of the oscillators we had to dump, was below the limit of 50 Hz (which is the limit when T_{PC} = 2 ms).

Method a), the relative acceleration is estimated through an observer.
In this case the relative acceleration is supposed to be not measurable. Therefore it will be estimated trough an observer.
An observer is basically a model of a physical system or "Model of the plant" (in our case the cinematic chain). The model is submitted to the same input signal as the physical system (in our case, the feed reference signal ω_s) and it is also supplied with measurable values of the physical system (in our case: the velocity), which is processed in the part of the observer named "Model correction" (see Fig. 4).
The model produces the not-measurable values of the state variables (i.e. the relative acceleration).

In our case we implemented an observer of second order (one for each axis). Its general structure is shown on Fig. 4, with the following configuration of coefficients:

$$\underline{A}_{Ob} = \begin{pmatrix} a_{11} & a_{12} \\ a_{21} & a_{22} \end{pmatrix};\ \underline{b}_{Ob} = \begin{pmatrix} b_1 \\ b_2 \end{pmatrix};\ \underline{H} = \begin{pmatrix} h_1 \\ h_2 \end{pmatrix};\ \underline{M} = \begin{pmatrix} 1 & 0 \\ 1 & 0 \end{pmatrix};\ \underline{x}_{Ob} = \begin{pmatrix} v_{Ob} \\ a_{Ob} \end{pmatrix};\ \underline{m}_{me} = \left(v_a\right) \qquad (6)$$

$d_{Ob} = d_1 = 0$; $\underline{c}_{Ob}{}^T = (c_1; c_2) = (1; 0)$

The coefficients are tuned in order to build a good model of the axis of the machine, which is approximated to a second order system.

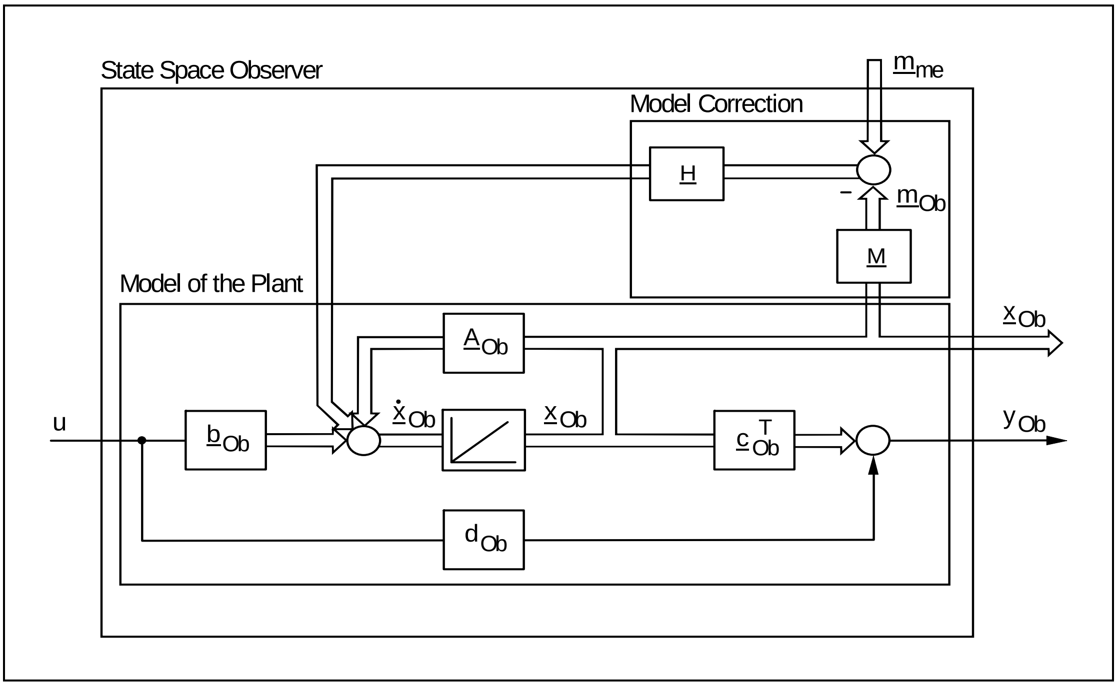


Fig. 4 – The observer, general structure.

The observer is placed into the control system as indicated on Fig. 5:

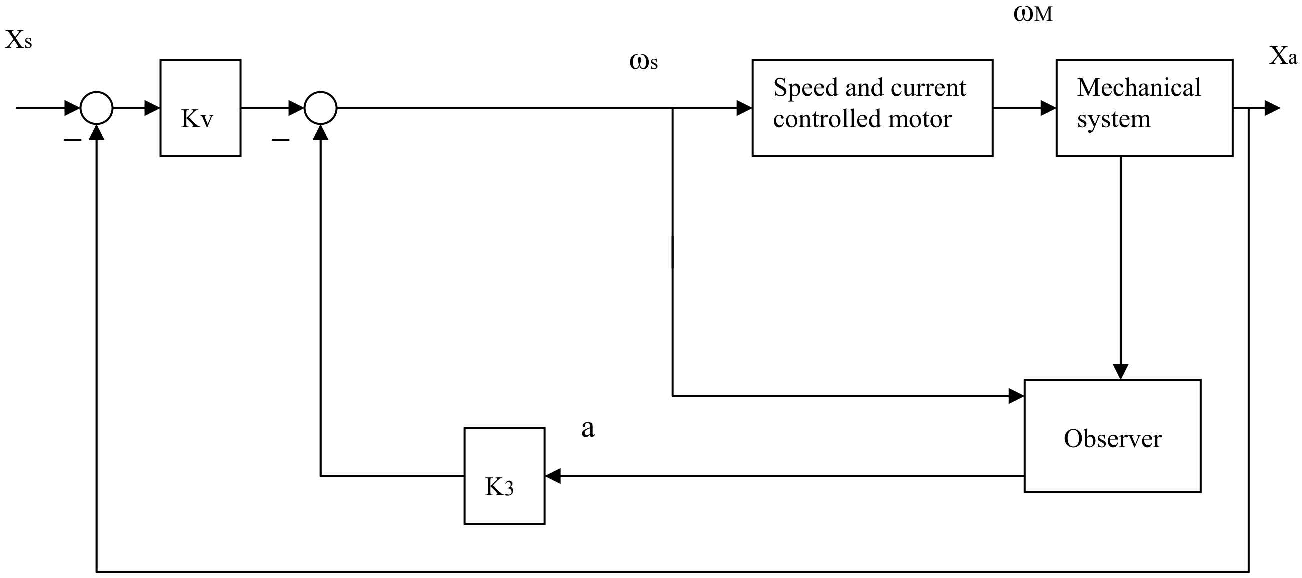


Fig. 5 - General structure of the control system with the observer.

During the year, this method has been tested on several machines. As a matter of example, we will just refer to one of the first implementations, on a German milling machine (Bokö, VH4/12). The X-axis with $f_{0FS} = 21$Hz, $D_0 = 0.1$, limited the ***Kv*** of the machine to 32 1/s.
The adoption of the state control restricted to the relative acceleration estimated through an observer based on a model of second order, allowed to set a new value of the ***Kv*** equal to 66 1/s. That means an improvement of factor 2.

Nevertheless the adoption of an observer encounters the following inconveniences:
1) The dynamic behaviour of the structure could change as per the mass of the piece. Depending on the morphology of the machine, the piece could be moved by an axis and the parameters of the observer need to be adapted.
2) The stiffness of the cinematic chain changes as per the position of the axis. It is more rigid when the ball screw is not inserted (on the side of the motor) and less rigid to the opposite side.
3) The model has to run in real time and involves an additional computation charge.

In our practical experiments, realised during the years on different kind of machines, we observed that the case 1) can require a new tuning of the observer; the case 2) appears to be less important and the module “model correction” (present in the structure of the observer) is able to adapt the model to the variations of the structure.
But it is evident that the sensibility of the method to theses weaknesses is strongly dependent on the specific machine.

Method b), the relative acceleration is measured by means of additional accelerometers.
A way to overcome the inconvenient of the observer is a direct measure of the relative acceleration.

Installing two accelerometers, one on the moving part and the second on the fixed body of the machine and subtracting the measured accelerations, we should be able to measure the relative acceleration.
Aiming to test this, we adopted capacitive accelerometers (in order to avoid the offset which is present on the piezo accelerometers) of the company BDC and we installed them on the x axis of an Huron KX10, milling machine.
The accelerometers introduced a noise that was unacceptable.

Our practical experiments were not successful. Attempts have been made to reduce noise by filtering the signal or measuring with two different accelerometers and improving the signal – noise ratio, but measures where not good enough

Probably, the adoption of more suitable accelerometers would improve the results of this method.
At this point, we have to point out that we did not carried out new experiments (not yet) using other accelerometers or adopting methods for the reduction of the noise.
But we are aware of experiments made at ISW (Germany) [2] where they adopted a Ferraris accelerometer (based on the Ferraris's principle) which is able to measure directly the relative acceleration. The accelerometer works on the basis of the Eddy currents excited in a conducing material by exiting magnets.
The Eddy currents are proportional to the relative velocity between the magnet and the conducing material (usually a simple metal plate). Therefore, the variation of the relative velocity produces variations of Eddy currents which are sensed by detecting coils, thus measuring the relative acceleration.
The only disadvantage of the method is the use of additional sensors.

Method c), the relative acceleration is computed through the double derivation of the position.
The acceleration can be measured through double derivation of the position. Furthermore, we know that each derivation will dramatically increase the noise power. Therefore the quantization noise represents the main problem of this method. However, the shape of the power spectral density of the position noise, which is flat between 0 and 250 Hz, is significantly modified by the double derivation. It has been determined that more than 95% of the acceleration noise power is concentrated above 100 Hz. Thus, it will be easily removed by low-pass filters which are inevitable present in a real-system. Fig. 6 shows the acceleration noise power spectral density (0 → 250 Hz) whereas Fig. 7 presents a plot of a 0.33 Hz, 0.4 m/s^2 acceleration signal (simulating an axis movement) with the noise due to the position resolution (the amplitude of the noise corresponds to a path deviation of 0.2µm, as it is explained below).

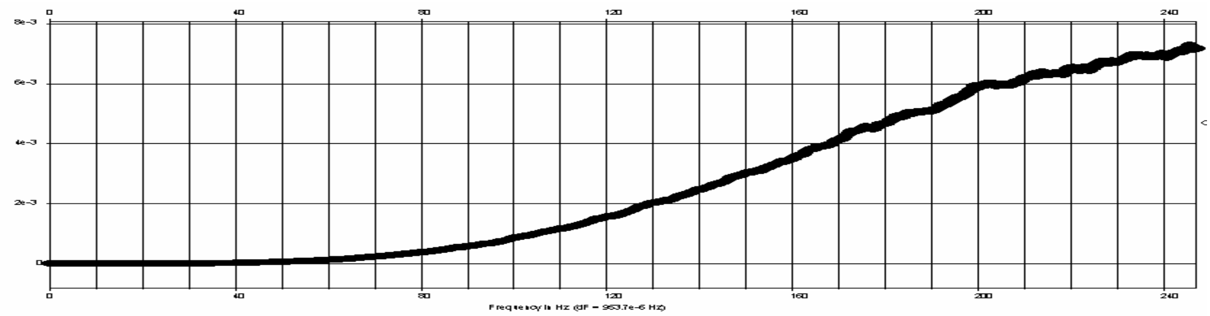


Fig. 6: Acceleration Noise Power Spectral Density

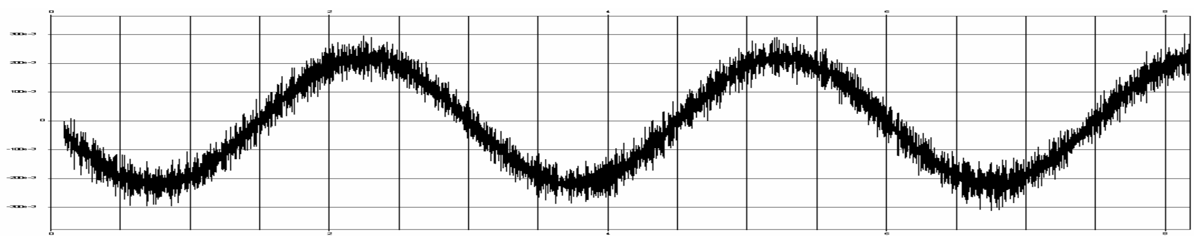


Fig. 7 Acceleration, 0.4 m/s^2 peak-tp-peak, and noise, σ_{noise} = 0.035 m/s^2

We carried our practical experiments on the X axis of the Huron KX10 milling machine, with control SIEMENS 840D, of our laboratory.

We had to dump a first oscillator, f_{0Me} = 42 Hz, ζ= 0,2.

The bandwidth of the speed control is f_{3dBSC} = 400 Hz (see the Fig. 8),

The sampling time of the position loop was T_{PC} = 0,004 s.
The condition 4), was not perfectly respected.

We decided to reduce the sampling time, fixing it at T_{PC} =0,002 s, corresponding to a sampling frequency f_{PC} = 500 Hz. Adopting this value, the limit of the frequency that may be dumped is 50Hz (as per relation (4)).
(Unfortunately, in our case the speed loop was inaccessible; therefore we did not have the possibility to adopt the sampling time of the speed control loop, more than 10 times lower).

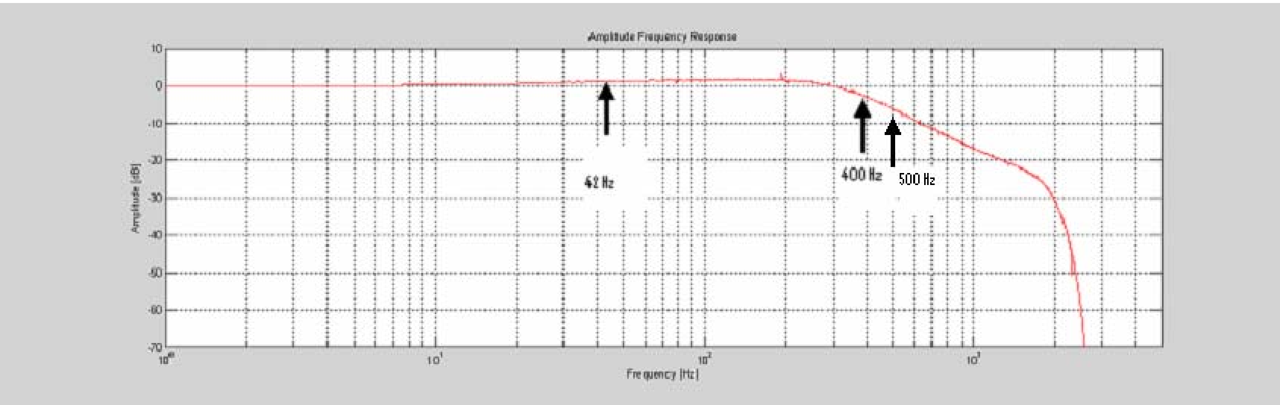


Fig. 8 – f_{0Me} = 42Hz, f_{3dBSC} = 400 Hz, f_{PC} = 400 Hz, placed on the response of the model of the speed loop of the machine.

The functionality of the NC, named "Synchron actions", allows the addition of algorithms that will be executed in the real-time part of the NC.
The measuring system of the machine has a resolution of 0,2 µm. At each sampling time, the measuring system and the electronics, send to the NC the displacement of the axis executed during the previous sampling period. This way, at each sampling time, the NC acquires the value of the average feed of the axis during the sampling period. The resolution of this value is: 0.0001 m/s. We added a simple Euler's digital derivation of these values that produces the acceleration relative acceleration suitable for the state control.

The Kv of the machine was, originally, 36.6 1/s.
We increased it at the value of: 95.8 1/s (an improvement of a factor 2.8).
The best value of K_3, as per the tests, was: 0,00584. The noise of the acceleration (due to the sampling) represents a noise added to the position reference signal of 0,21 µm, before the filters.

The result was very impressive. It is represented on Fig. 9 that shows the inertial acceleration measured by a physical accelerometer which is placed on the sliding axis.

In this condition, without feed back of relative acceleration (i.e. without reduced state control), the machine is definitely unstable (see at the left side of the Fig. 7, "Without State Control").

In the same conditions, when closing the additional loop (with reduced state control) the axis is stable (see at the right side of the Fig. 9, "With State Control").

With this method, on this machine, we obtained the possibility to increase the Kv gain of a factor 1,8 (from the original value of 36.3 1/s to the new value of 66.6 1/s).

<table>
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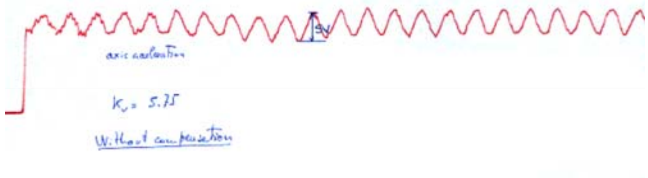
<tr><td>
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Fig.9 – The plots show the inertial acceleration measured on an axis of a machine tool when the gain of the position loop (Kv) has been increased of a factor: 2.8.

Conclusions.

The reduced friction factor of the ball or roller linear guideways, reduces the dumping factor of the axis of a machine tool and limits the stability of the machine. The gain of the position loop (kv) is negatively affected, and its value need to be limited.

A state control, restricted to the relative acceleration may dump the axis, this way allowing a significant increase of the Kv factor.
We have presented the basic principles of operation of a reduced state control applied on machine tools. We have described the conditions and the limits of the application.

We have presented the results obtained with three different methods, based on three different methods of acquisition of the relative acceleration.

The practical experiments executed during several years on different machines and adopting different methods, show that the three methods offer the possibility to dump the axis of the machine, thus offering the possibility to increase the Kv factor of the position loop in a rate that is near the factor 2.

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