Limiting factors for the active suppression of structural chatter vibrations using machine’s drives
Xavier Beudaert, Asier Barrios, Kaan Erkorkmaz, Jokin Munoa

To cite this version:
Xavier Beudaert, Asier Barrios, Kaan Erkorkmaz, Jokin Munoa. Limiting factors for the active suppression of structural chatter vibrations using machine’s drives. XIIth International Conference on High Speed Machining (HSM2015), Oct 2015, Nanjing, China. hal-01342267

HAL Id: hal-01342267
https://hal.archives-ouvertes.fr/hal-01342267
Submitted on 5 Jul 2016

HAL is a multi-disciplinary open access archive for the deposit and dissemination of scientific research documents, whether they are published or not. The documents may come from teaching and research institutions in France or abroad, or from public or private research centers.

L’archive ouverte pluridisciplinaire HAL, est destinée au dépôt et à la diffusion de documents scientifiques de niveau recherche, publiés ou non, émanant des établissements d’enseignement et de recherche français ou étrangers, des laboratoires publics ou privés.
Limiting factors for the active suppression of structural chatter vibrations using machine’s drives

X. Beudaert\textsuperscript{1,a,*}, A. Barrios\textsuperscript{1,b}, K. Erkorkmaz\textsuperscript{2,c}, J. Munoa\textsuperscript{1,d}

\textsuperscript{1}Control and Dynamics, IK4-Ideko, Elgoibar, Basque Country, Spain
\textsuperscript{2}Precision Controls Laboratory, University of Waterloo, Waterloo, ON, Canada
\textsuperscript{a}xbeudaert@ideko.es, \textsuperscript{b}abarrios@ideko.es, \textsuperscript{c}kaane@uwaterloo.ca, \textsuperscript{d}jmunoa@ideko.es

Keywords: chatter, control, damping, milling

Abstract. Chatter vibrations are a major limitation for rough milling operations and lead to productivity reduction, low tool life and poor surface finish. It has been shown recently that the machine tool’s own drives can be used to increase the stability limit related to structural modes of the machine. To damp the low frequency modes, a new feedback loop is added to the classical position, velocity and current cascaded control. The objective of this article is to analyse the limitations of this new chatter suppression technique. Constraints related to the non-collocated control are first studied on a simplified 3-mass model and then experimentally demonstrated on a 3-axis horizontal milling centre. The industrial integration of the new control loop with sampling time constraints and limited drive’s bandwidth is analysed. After determining the appropriate conditions to use this chatter suppression technique, a cutting test demonstrates that the stability limit can be doubled in the low regions of the stability lobes.

Introduction

In heavy duty milling operations, machine tool users are looking for high material removal rates which are mainly limited by two factors: the spindle power and the chatter vibrations. However, chatter vibrations can appear with really small depth of cut reducing drastically the machine’s productivity; hence, it is an important field of investigation for the research community. In general, chatter can come from the flexibility of any element in the force flow: tool, spindle, machine structure, workpiece, fixtures... However, in heavy duty machining, chatter is mostly related to vibration modes of the whole machine structure. This paper focuses on those low frequency chatter problems occurring at frequencies between 15Hz and 200Hz.

Dampers can be added to the machine tool structure to increase the chatter stability limit. A tuned mass passive damper can damp a specific frequency. Unfortunately, the available space is really limited and the frequency of the structural modes are changing with the position of the machine so passive dampers are not able to satisfy the requirements. Active dampers can handle dynamics variations and have been successfully used for chatter suppression [1, 2]. However, these solutions require the installation of new actuators on the machines.

Instead of using additional actuators, the machine’s own drives can be used to increase the damping. The use of machine’s drives to suppress chatter vibrations has been simulated first by Chen and Tlusty [3]. Recently, the increase of the chatter stability related to the low frequency structural modes has been demonstrated on a large milling machine by Munoa et al. [4].

The classical cascaded control with current, velocity and position loops can affect the structural machine’s dynamics. Especially the proportional gain of the velocity loop can be tuned to provide the optimum damping [5, 6]. However, the addition of damping brought by the velocity loop is moderate compared to what can be achieved by the use of an additional feedback loop.

Additional feedback loops have been used in the literature to reduce inertial vibrations coming from the machine movements. Szabat and Orlowska-Kowalska presented a comparative study of the
different feedback possibilities [7]. Dietmair and Verl showed a drive based vibration damping with an additional feedback in the velocity setpoint [8]. Zatarain et al. placed an accelerometer as close as possible to the tool and used a state space observer to improve the dynamic behaviour of a machine [9]. Zirn and Jaeger also used a state space control to feed back the load acceleration in the acceleration setpoint [10]. Hosseinabadi and Altintas damped a ballscrew mode using the linear encoder signal [11]. A comprehensive review of active damping using machine’s drives is presented by Altintas et al. in [12]. However, the research in this field is limited by the difficulty to modify the control loop structure on industrial CNCs, thus open architecture controllers are required to implement new CNC algorithms [13].

CNC manufacturers are also introducing active damping options in their controllers. Fanuc proposes the vibration damping control function that feeds back the difference between the speeds of the motor encoder and of the linear encoder [14]. This function is particularly suited to damp the mode originating from the elasticity of the drive’s transmission. Siemens is also proposing a similar solution with the Advanced Position Control (APC) [15] which is analysed by Zirn in [6]. A patent owned by Mitsubishi Electric Corporation presents the addition of load feedback to the classical cascade loops [16]. Heidenhain offers the Active Vibration Damping (AVD) option to reduce machine’s structure oscillations. Moreover, a dedicated chatter suppression option called Active Chatter Control (ACC) is available [17-19]. Spindle drive is also used to suppress chatter vibrations [20]. The spindle speed variation technique is implemented for example by Okuma in the Machining Navi options [21]. This non exhaustive industrial review proves that the major CNC manufacturers are working to act against vibrations directly with the machine’s drives.

Up to now, successful implementations of additional feedback loops have been presented but no information is given about the limitations related to this control strategy. Moreover, specific questions are raised when the drives are used to increase the chatter stability.

The objective of this paper is to study the limitations concerning the use of the machine’s drives to increase the chatter stability limit. The first physical restriction which is analysed is related to the destabilization of counter-phase modes that limit the possible dynamic behaviour improvement. Then, technical restrictions like the integration of the additional feedback in an industrial CNC and the related sampling time limitations as well as the dynamical properties of the drives and their limited bandwidth are taken into account. Finally, the positive and negative effects of the additional feedback loop on the stability are described and the ability of the drive to suppress chatter vibrations is demonstrated.

The rest of the paper is organized as follows. First, a simplified model allows to explain the problems related to the non-collocated control that can destabilize counter-phase modes. Then, the technical requirements related to the CNC integration, the limited sampling time and the machine’s drives bandwidth are analysed. Finally, the effect of the additional feedback on the chatter stability allows to determine the appropriate conditions for the use of this new chatter suppression technique. Cutting tests on a 3-axis milling machine demonstrate the ability of the drive to suppress chatter vibrations.

**Limitation related to the modes’ destabilization**

In the literature, different load feedbacks are analysed on a 2-mass system presenting a single mode. However, the main limiting factor which prevents improving further the dynamical response of the system is related to the mode destabilization. Fig. 1 presents the block diagram of a drive with a 3-mass system having two vibration modes. The typical control structure with current, velocity and position cascaded loops is used. Due to its high bandwidth, the current loop can be neglected and is modelled here as a perfect amplification. The integral term of the velocity loop controller is not considered here for sake of simplification. Among the different additional feedback strategies presented in the literature, two acceleration feedback loops are studied here.
The acceleration measured in the third mass can be fed back to the velocity setpoint with a gain $G_v$ or to the acceleration setpoint with a gain $G_a$. As the acceleration measurement point and force application point are different, limitations related to non-collocated systems are faced [22].

As shown in the root locus of the velocity loop (Fig. 2a), the acceleration feedback in the velocity setpoint is affecting the position of the zeros of the system whereas the acceleration feedback in the acceleration setpoint is modifying the position of the poles of the system. If only one mode is considered, there is no limit to increase the damping of the system. However, real systems are presenting many modes [23] and with the non-collocated control some modes can be damped whereas others are destabilized.

The proportional gain of the velocity loop denoted $K_p$ is positioning the poles of the system on the root locus. If the gain $K_p$ is high, the pole will be close to the zero of the root locus, hence the acceleration feedback in the velocity setpoint has a direct effect on the system pole whereas the acceleration feedback on the acceleration setpoint has almost no effect. However, playing with the value of the gain $K_p$ both acceleration feedback strategies can have similar performances.

The Fig. 2b presents the effect of the acceleration feedback on the compliance of the third mass. The targeted mode is damped but the second mode is destabilized. This example schematically presents the limitation related to the non-collocated control inducing the destabilization of counter-phase modes. Experimental measurements showing this effect are presented further.

To counter the effect of mode destabilization, a loop shaping controller has been used in [4][3]. Indeed, by selectively adapting the gain and phase for the frequencies of interest, it is possible to avoid the destabilization of the counter-phase modes. However, when the modes are close to each other and considering the dynamical variations due to the machine posture modification, the mode destabilization remains the main element limiting the performance of the active damping strategy when machine’s own drives are used.

Another way to avoid destabilizing one of the mode is to feedback the difference of the velocities between $m_2$ and $m_3$. This kind of strategy is well adapted for the modes which are in the transmission chain and for which the motor encoder and the linear scale can directly measure both velocities. For structural chatter, the use of this technique results to be more difficult due to the complex mode shape and to the difficulty to measure the required signals.

Both feedbacks in the velocity and acceleration setpoints can improve the dynamical behaviour of the machine and a deeper study should be carried out to compare them. However, for the rest of the paper, the acceleration feedback in the velocity setpoint is preferred to be able to keep a high value of the velocity loop proportional gain.

A vertical ram-type machining centre is used to illustrate the rest of this article, further details about the dynamics, stability lobes and experimental chatter limits of this machine can be found in [24]. Fig. 3a presents the shape of the main mode which has to be damped and which is mainly related to the bending of the ram. The dynamics of this machine is complex and cannot be accurately modelled by the 3-mass model presented before but this simple model allows to understand the basic behaviour of the system. Indeed, in Fig. 3b the amplitude of the main peak at 60Hz is decreased as the gain is increased. However, the modes around 45Hz and 75Hz are destabilized due to the counter-phase mode behaviour.
Limitation related to the technical implementation

**Industrial CNC integration, sampling time and noise issues.** In previous work an Open CNC has been implemented to avoid most limitations related to industrial CNCs [4]. Hence the control structure could be modified easily and the sampling time was low enough to not result in significant phase lag. When using industrial CNCs, the additional feedback loop can be implemented in the PLC program or in the CNC kernel, but in all cases the achievable sampling time is limited. This limitation induces a phase lag and can hamper the possibility of successfully controlling the system [25].

The integration of a new feedback loop in an industrial CNC is not a straightforward task and can be impossible for some CNC’s. Machine tool builders have access to the PLC but the scanning time is usually high (1 to 20ms). Access to the real time part of the CNC kernel is restricted by most CNC manufacturers. In the Siemens 840D CNC, it is possible to introduce additional feedback loops using the Compile-Cycle programming options. Indeed, the Compile-Cycles are software components that are developed by the end-user and are added to the standard system using the open system architecture of the NC kernel [26]. A fast analogue input module is added to the communication bus of the CNC to read the accelerometer signals. This signal is processed internally by the CNC to apply filters and is added to the velocity or acceleration commands. The available sampling time depends on the capacity of the processor of the CNC. The machine used in this paper has a relatively slow CNC processor hence the smallest sampling time which can be used to read the accelerometer signal is 4ms.
In a first approximation, the sampling effect can be approximated by a first order filter to take into account the phase lag that is introduced. This phase lag can be compensated by adjusting the filters of the acceleration feedback controller. However, when the sampling time is too high, the missing information makes impossible to control the system. Experimental tests carried out on a milling machine show that with a sampling time of 8ms, the compliance Frequency Response Function becomes noisy and the system cannot be controlled properly (Fig. 4).

The tests revealed that even with a relatively high sampling time (4ms), the additional feedback loop can damp structural machine tool modes. This is important information because it means that the low sampling tasks of the CNC can be used to implement the acceleration feedback strategy. Indeed, experiments show that the 60Hz mode can be damped effectively even if the sampling time reaches 6ms (166Hz).

In the CNC control literature, the problem related to sampling time is recognized and experimental conditions are typically chosen to ensure that the sampling period has minimal negative impact on the loop stability. Sometimes, additional measures are also used to mitigate errors due to sensor noise as well; such as Kalman filters [11], state space observers [8] or specific sensors like Ferrari sensors. With the full implementation in the Compile-Cycle, the grounding of the machine is naturally acting against the electrical noise. In our experimental work, standard industrial accelerometers are used and no specific signal processing is required.

**Limitation related to drive’s bandwidth.** Apart from the non-collocated control, another problem of using the machine’s drives is their limited bandwidth. Indeed, the drives are not designed for this purpose and the tuning of the velocity and position loop is constrained by the machines’ modes [6], [5]. A main challenge is dealing with vibration modes which are outside the response frequency range (i.e., bandwidth) of the feed drives. Fig. 5 shows the bode diagrams of the position and velocity loop of the X axis linear drive. The Bode diagram of the velocity loop shows that the machine has several modes related to the structure of the machine (column, ram, etc.) at low frequency (<80Hz) and many other modes at higher frequencies are related to smaller components. The mode at 170Hz is specifically dangerous because it can get unstable if the proportional gain of the velocity loop is increased. The additional acceleration feedback can also destabilize this mode but as this mode is far from the targeted 60Hz mode, a loop shaping controller design can avoid exciting this mode.
In a previous work [3], rack and pinion and ballscrew drives were successfully used to damp structural modes of a large milling machine. The limited bandwidth of the drives is preventing using this technique to counter modes which are not coming from the machine structure, like tool or spindle modes which usually have frequencies higher than 300Hz. However, the drives seems able to damp structural mode and the main problem related to the dynamics of the drive is also the machines’ modes destabilization due to the actuator phase lag.

**Limitation related to the effect on the stability lobe**

It has been shown in Fig. 3b that it is possible to reduce the maximum amplitude of the Frequency Response Function. However, for chatter suppression applications, the effect on the stability lobes diagram should be taken into consideration. The non-collocated acceleration feedback strategy can induce the amplification of the counter-phase mode, which can thus limit the cutting capability in some regions of the stability diagram. The effect of the acceleration feedback on the velocity setpoint is shown in the stability lobes presented in Fig. 6. By lowering the amplitude of the maximum peak of the Frequency Response Function, the stability limit is increased for the lower portions of the lobes. On the other hand, the stability limit is reduced for the higher portions of the lobes and the sweep spots may disappear.

Depending on the spindle speed, the acceleration feedback can improve or worsen the stability. This is an important drawback of this chatter suppression technique using machine’s drive. Hence, the additional feedback loop should be used carefully.

![Figure 5: Bode plot of the X-axis linear drive.](image)

![Figure 6: Zero order stability lobe approximation [27] without and with acceleration feedback in the velocity setpoint](image)
Experimental demonstration of the chatter suppression

The worst position on the stability lobe has been selected to demonstrate the ability of the drive to suppress chatter vibrations. A tool of diameter 125mm with 8 teeth is used with a spindle speed of N=300rpm and a feed per tooth fz=0.2mm/z. The radial depth of cut is 100mm which represents 80% of the tool diameter, the axial depth of cut is 1.5mm. Fig. 7 presents the measured acceleration signal in time and frequency domain. Strong chatter appears at 61Hz related to the bending mode of the ram in the X direction. When the acceleration feedback is activated, it is possible to see that the amplitude of the vibration is immediately reduced and the frequency spectrum shows that the cutting process is stable. The stability limit is predicted considerably accurately when no active damping is used, however, with the drive’s acceleration feedback activated only the main trends are reliable in the stability lobes presented in Fig. 6. For the selected cutting conditions, the stability limit can be doubled which is a huge improvement that can be obtained at a relatively low cost.

Conclusion

This article investigates the use of the machine’s own drives to damp the structural vibration mode of a machine tool and hence increase the chatter stability. The main contribution of this article is the analysis of the limiting factors related to this new chatter suppression technique. It has been shown that the most problematic issue is the destabilization of other modes of the machine. A specific effort to design the feedback controller can improve the behaviour but it is still the most critical element. It has been validated experimentally that the integration in a Siemens 840D CNC is possible and that the sampling time requirements are not strong. Indeed, 4ms sampling time is enough to control the system and a 61Hz chatter could be removed. Another possible limiting factor is the machine drive bandwidth. However, rack and pinion, ball screw and linear drives have been successfully used to damp structural machine tool modes. Finally, the stability lobe diagram and cutting tests show that this additional feedback can provide a huge improvement in the depth of cut for the minimum regions of the lobes. However, the additional feedback can worsen the stability for other areas of the stability lobes. To conclude, this article provides a clearer view on the limiting elements that should be taken into account when the machine’s drives are used to suppress chatter vibrations.

References


