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Active suppression of structural chatter vibrations using machine drives and accelerometers

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ABSTRACT

Chatter vibrations coming from the machine tool structure are a major limitation for heavy duty milling operations. Passive and active dampers can be added to increase the productivity. However, the price and required space of these absorbers are important drawbacks for their use in the industry. In this work, the machine tool's own drives are used to suppress chatter with the help of an external accelerometer located close to the tool centre point. The measured acceleration is fed back as an additional control loop. A significant increase in the chatter stability limit is demonstrated experimentally on a large milling machine.

1. Introduction

Chatter vibrations have been a major concern during the last decades, due to their limiting effect on the productivity of machine tools [1]. While chatter can be caused by many sources of flexibility in the machine-workpiece-tooling assembly, in large machines built for heavy duty milling, chatter is usually caused by low frequency structural modes. In this case, passive absorbers are generally too bulky to use and not always robust enough to handle dynamics variations due to posture change. To overcome these limitations, active dampers located close to the tool tip have recently been proposed [2–4]. However, this solution requires an additional actuator which increases cost. The machine tool's feed drives, on the other hand, could also be used to provide damping to the structure. This is the idea proposed in this paper.

Chen and Tlusty [5] simulated, for the first time, the possibility of improving chatter stability using acceleration feedback in the servo drive. Since then, active damping of drive vibrations has been studied extensively [6]. However, most of the feed drive related vibration damping research has focused either on improving the control bandwidth, or achieving higher positioning accuracy under increasing acceleration and jerk values. Thus, the application of acceleration feedback to increase chatter stability has not been demonstrated yet. Closest to this idea, Kakinuma et al. [7] applied force control on a test lathe to suppress chatter.

This paper proposes a new technique for suppressing chatter, by injecting active damping through the feed drives using low-cost industrial accelerometers as sensors. The method is particularly suitable for large machine tools with low frequency modes, near or below the servo bandwidth. A model is developed to predict the effect of this acceleration feedback and to design a robust controller by considering the performance and robustness for an envelope of machine postures.

The implementation of chatter suppression through the drives is not trivial. First of all, the acceleration close to the tool tip should be measured or estimated. Since the machine's drives are used to inject damping, this implies working with a non-collocated system. Therefore, the design of a robust and stable controller is an important point for a real machine with complex dynamics. Also, acceleration measurement noise is an important issue. This was previously addressed using Ferraris sensors or state observers [8]. Here, careful design of the control law, and good electrical grounding, were crucial in overcoming this challenge.

2. Control architecture and associated challenges

Different additional feedback options have been proposed in the literature [9]. The use of an accelerometer to generate additional velocity set points, similar to the approach in [10], has been selected for chatter suppression due to several reasons. First of all, this implementation architecture is very suitable for industrial deployment, as it allows off-the-shelf drives to be used without modification to close the current and velocity control loops. Moreover, it allows having a high value of damping while keeping the proportional velocity loop gain K_p high as well. The CNC kernel, which performs trajectory planning and position control tasks, can thus host the active damping control law; using, for example, Compile-cycles which are available on the Siemens 840D. The velocity commands generated by the position and active damping control laws would be superposed and applied directly to the drives.

Fig. 1a shows the proposed control architecture considering a simple mechanical system with one vibration mode. Fig. 1b and c

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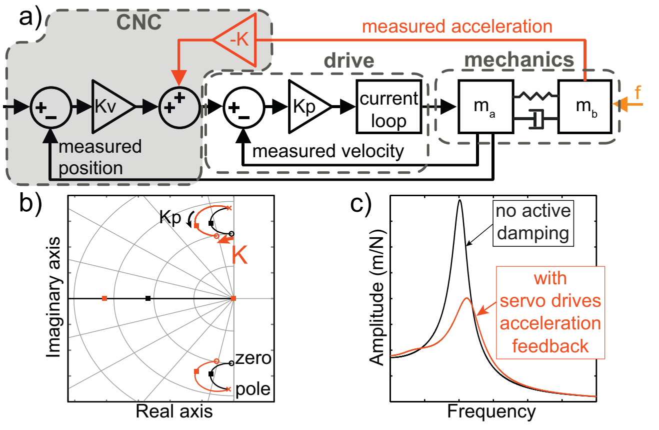

Fig. 1. Vibration damping by velocity setpoint acceleration feedback: (a) control diagram, (b) velocity loop root locus and (c) tool tip compliance.

shows the damping effect acceleration feedback achieves on the vibratory poles, and the resulting load side dynamic compliance.

While the prospect of increasing a machine tool's dynamic stiffness without adding passive or active dampers is appealing, there are certain requirements for a successful implementation:

1. The damping control law should not only damp out the problematic vibration mode(s), but also avoid significantly deteriorating the dynamic stiffness in other frequencies. The destabilization of other modes is an important limitation.
2. Practical limitations, such as sampling delays or bandwidth and flexibility of the servo system, need to be considered, as well as noise issues which are also of major importance.
3. Stability margins need to be retained in the presence of dynamics variations due to machine posture change.
4. The damping strategy should avoid excessive oscillations in the carriage position. These would lead to early wear and damage of the machine tool drive components.
5. Design and commissioning of the damping strategy on industrial machines should be reasonably procedural.

The modelling and controller design presented in Section 4 was developed to address the above requirements.

3. Machine description and experimental setup

Fig. 2a shows the large milling machine used in this research. Detailed dynamic verification has been carried out, including modal analysis and chatter tests. These tests showed that, depending on the cutting configuration and position in the workspace, either the mode dominant in the *x*-direction with a chatter frequency between 32 and 44 Hz (Fig. 2b and c), or a secondary mode dominant in the y-direction with 26–29 Hz chatter frequency, were the main contributors to chatter. The proposed active damping architecture has been oriented to damp out both modes.

The shape of the targeted mode in the *x*-direction is composed of: torsion of the column, and bending of the ram in the *x*–*z* plane. These two elements are vibrating in phase. The presence of a secondary mode at 56 Hz, which is close to the main mode, should also be remarked. This mode is similar to the dominant mode, but the elements are vibrating in counter phase which creates a difficult situation for applying non-collocated active control.

There are other problematic modes as well, around 110 and 180 Hz. However, as these frequencies are beyond the bandwidth of the servo drives, control of these modes was not targeted.

The *x*-axis is driven by a rack and double-pinion mechanism, electronically preloaded to minimize backlash. The *y*-axis is driven by a ball screw drive. The spindle head can change orientation to act in different cutting planes. Therefore, the accelerometers (PCB 602D01) were located at the ram tip in *x*- and *y*-directions to facilitate feedback to handle all possible spindle positions. The motors are driven by drives which, for this study, were configured to operate in torque mode.

An Open CNC was developed to implement the control scheme on the machine tool. The motors' velocity and position loops were closed inside a dSpace controller at 4 and 1 kHz, respectively, imitating their industrial counterparts. The active damping controller was also implemented in dSpace at 4 kHz and injects additional commands into the velocity control loop.

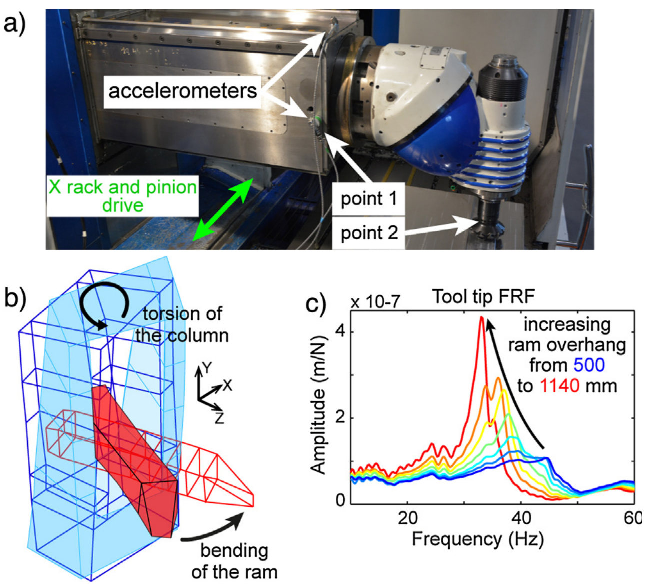

Fig. 2. (a) Large milling machine, (b) 36 Hz vibration mode shape, and (c) variation of tool tip dynamic compliance in XX direction with ram overhang.

4. Active vibration damping controller design

The active damping is applied both to *x*- and *y*-axis but, to keep the presentation simple, the proceeding analysis on tool tip compliance prediction and loop shaping damping design focuses on the dynamics along the *x*-axis.

4.1. MIMO model of the machine tool and damping control law

The machine tool structure, coupled with the servo control system, can be represented as a MIMO plant as shown in Fig. 3:

$$\begin{bmatrix} a_2 \\ a_1 \end{bmatrix} = \begin{bmatrix} G_{a_2 f_2} & G_{a_2 f_1} & G_{a_2 v_r} \\ G_{a_1 f_2} & G_{a_1 f_1} & G_{a_1 v_r} \end{bmatrix} \begin{bmatrix} f_2 \\ f_1 \\ v_r \end{bmatrix} \quad (1)$$

Above, a_2 is the acceleration at the tool tip (i.e., performance point of interest). a_1 is the acceleration at the location of the accelerometer (ram tip). The inputs considered are f_2, f_1 and v_r which are: the cutting force at the tool tip; the measured impact hammer force applied at the ram tip (for model building and compliance prediction); and the additional velocity command injected into the velocity control loop in order to apply active vibration damping. v_r is computed using the following law:

$$v_r(s) = -K(s) \cdot a_1(s) \quad (2)$$

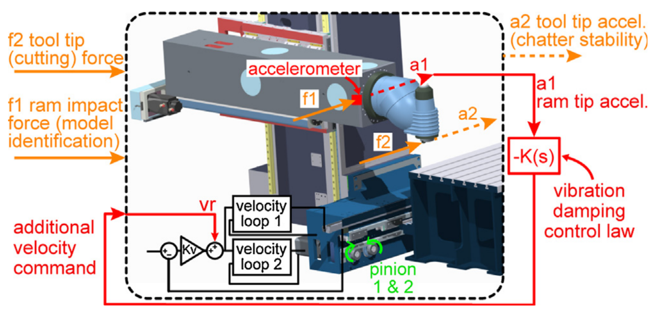

Fig. 3. Implementation of the proposed active damping strategy.

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