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Automatic tuning of active vibration control systems using inertial actuators

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ABSTRACT

Material removal rates of machine tools are mainly limited by chatter, which is caused by the machine's most flexible structural modes. Active damping of the structural modes requires the identification of a machine dynamics model used in tuning the active controller. This complex process has to consider uncertainties, actuator saturation, and the stability of the controller. This paper presents an automated identification of machine dynamics and auto-tuning of the controller using the same actuator and vibration sensor. The proposed method has been experimentally demonstrated with simple and advanced controllers which led to notable increases in chatter free material removal rates.

1. Introduction

The maximum material removal rate of machine tools is determined by either the drive capacity of the spindle and feed drives or by the chatter stability limit. Under certain conditions the machining process can become unstable and chatter occurs, and leads to poor surface finish and large dynamic forces which can damage the machine and the part [1].

Active vibration control (AVC) systems can be used to mitigate chatter caused by the natural modes of the machine tool's structure. Mainly four different actuator designs exist [1] depending on the application: active spindle bearings [2], active tools [3], active workpiece holders [4] and active structural chatter suppression using an inertial actuator [5,6,7]. Piezoelectric actuators can be integrated to the machine structure [8], or the feed drives of the machine tool can be used for active damping of structural modes [9]. The performance of the active damping system with the feed drives can be improved by attaching an accelerometer near the Tool-Center-Point [10]. Piezoelectric actuators can also be used for static compensation [1], but compared to the inertial actuators, the integration takes more effort [5] and is not practical as a retrofit solution. This publication focuses on AVC systems using inertial actuators.

The implementation of an AVC system to a machine tool is a comprehensive, time consuming task and needs expert knowledge [1]. Furthermore, it might be necessary to adapt the controller during its operation, as the tuning does not only depends on the machine tool structure, but also on the machining process. If a new machining process shows higher vibration amplitudes than the one used for tuning, the actuator may saturate unless the controller is re-tuned.

This paper introduces a new methodology for an automatic tuning of AVC systems using inertial actuators. A magnetic actuator and accelerometer are used for both identification and active damping of low frequency machine tool structural modes. The actuator is mounted at a critical location where the mode needs to be damped. The machine is excited by the actuator, and its frequency response function (FRF) is measured. The measured FRF is fitted to obtain a single input single output state space model or the corresponding transfer function, which is used to automatically tune the active damping controller in simulation environment. The tuned controller gain is used for real time active damping control of the machine using the same set-up and actuator.

2. Experimental setup for active vibration control

The experimental active damping setup used in the paper is shown in Fig. 1. The magnetic actuator (SA10-V30 by CSA Engineering) and an accelerometer (KS 813B by MMF) are placed at the antinode location of the machine's structural mode which needed to be damped. The actuator has a bandwidth of 20–1000 Hz, a maximum force output of 45 N, and is driven by an amplifier (BAA 120 by BEAK). The IEPE amplifier of the acceleration sensor has a built-in 2nd order Butterworth low-pass filter with a cut-off frequency of 1 kHz. The active control law is implemented on a real time dSpace control system with a sampling frequency of 10 kHz. The proposed system was experimentally proven on two machine tools.

The transfer function of the machine at the active damping location is experimentally identified. The machine is excited by the same magnetic actuator using Pseudo Random Binary Sequence

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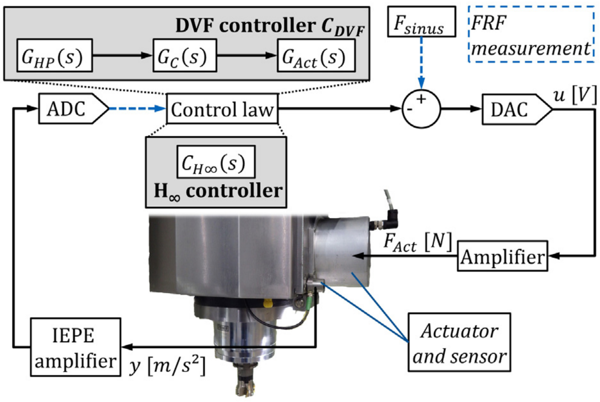

Fig. 1. Machine tool with AVC system and controller block diagram.

(PRBS) load F_{PRBS}, and the Frequency Response Function (FRF) is measured as illustrated in Fig. 2. The FRFs of two machines are measured: Machine A) SPINNER U5-620 Machining Center with a dominant mode at 58 Hz and Machine B) PITTLER PV 630 turning machine with a dominant mode at 46 Hz, which need to be damped. The measured FRFs are curve fitted to obtain a single input single output state space model of the machines using the identification tool–box (n4sid) of MATLAB® as:

$$\dot{x} = A \cdot x + B \cdot u$$

$$y = C \cdot x \tag{1}$$

where the state vector (**x**) contains the impact on the vibration response (acceleration) of each mode, which need to be damped. The state space matrices **A**, **B**, **C** represent the entire dynamics of the machine up to 850 Hz, leading to the following 6th order model of Machine B (Fig. 4):

$$A = \begin{bmatrix} -8.0 & 230.9 & 0 & 0 & 0 & 0 \\ -230.9 & -8.0 & 0 & 0 & 0 & 0 \\ 0 & 0 & -7.7 & 291.5 & 0 & 0 \\ 0 & 0 & -291.5 & -7.7 & 0 & 0 \\ 0 & 0 & 0 & 0 & -6.9 & 381.3 \\ 0 & 0 & 0 & 0 & -381.3 & -6.9 \end{bmatrix}$$

$$B = \begin{bmatrix} -4.5 \\ -3.9 \\ -11.3 \\ -5.9 \\ -0.2 \\ 1.4 \end{bmatrix};$$

$$C = [4.2 \quad -7.0 \quad -5.5 \quad 6.8 \quad 10.1 \quad -2.7]$$

The transfer function of the machine can be derived from the state space model as:

$$G_{u \to y}(s) = C \cdot (s \cdot I - A)^{-1} \cdot B \tag{2}$$

The state space model contains all the dynamic modes, and is used to simulate the performance of active damping controllers.

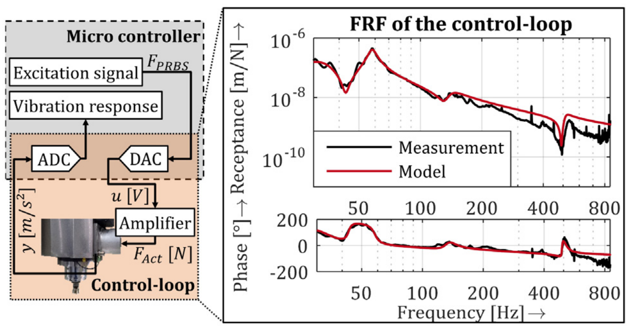

Fig. 2. Identification of SPINNER U5-620 Machine.

The following natural frequencies of the machines were included in the modal model of the machine during simulations.

$$f_n(SPINNER) = [33.5; 58.0; 138.9; 507.9]\text{Hz}$$

$$f_n(PITTLER) = [36.7; 46.4; 60.7]\text{Hz}$$

3. Automatically tuned active damping controller

There has been several active damping control strategies reported in the literature for both: non-model-based controller [7] and model-based controller [6]. A non-model-based Direct Velocity Feedback (DVF) controller, which is simple to apply when the control gain is automatically tuned, is proposed in this paper. It is also compared against more sophisticated, model-based H∞ robust controller. The robustness of H∞ controller against system dynamics changes was analyzed in Ref. [5]. The DVF controller generates an actuator force $F_{Act}(t)$ proportional to the vibration velocity ($\dot{x}(t)$) which leads to 180° phase shift between the disturbance and the actuator forces at the natural frequencies of the machine. The actuator force can be expressed in Laplace (s) domain as:

$$F_{Act}(s) = K_{v,stat} \cdot \dot{x}(s) = K_{v,stat} \cdot x(s)/s \tag{3}$$

where the proportional gain $K_{v,stat}$ needs to be tuned. The vibration velocity ($\dot{x}$) is obtained by integrating (1/s) the acceleration measurement (x), and used as a feedback signal by the controller after series of cascaded signal processing filters. The DVF controller works only around the natural frequencies of the machine's structure, and it is quite sensitive to low-frequency noise [7], which mainly occurs during high acceleration motion in the working direction of the actuator. The amplification of low-frequency noise is reduced by weakening the integrator by a 2nd order high pass filter $G_{weak}(s)$ with the cut-off frequency ω_w leading to the following transfer function

$$G_C(s) = K_{v,stat} \cdot \frac{1}{s} \cdot G_{weak}(s)$$

$$\rightarrow G_{weak}(s) = \left(1 - \frac{\omega_w^2}{s^2 + 2 \cdot \zeta_w \cdot \omega_w \cdot s + \omega_w^2}\right) \tag{4}$$

A low damping ratio (ζ_w) sharpens the phase jump but also increases an unwanted amplification around ω_w.

The DVF controller needs to filter the low-frequency noise (Fig. 1) further by adding an additional 2nd order high-pass filter $G_{HP}(s)$ and a compensation filter ($G_{Act}(s)$) for the transfer function of the inertial actuator, which extends the working bandwidth of the controller to lower frequencies:

$$G_{HP}(s) = \left(\frac{s}{s + \omega_{HP}}\right)^2; \; G_{Act} = \frac{s^2 + 2\zeta_A \cdot \omega_A s + \omega_A^2}{s^2} \cdot G_{weak}^2(s) \tag{5}$$

All filter parameters are automatically determined by the lowest critical natural frequency (f_n) of the machine and the actuator's bandwidth as listed in Table 1. For interrupted cutting operations like milling, adaptive band rejection filters are used to filter out the tooth passing frequency and its harmonics, because the DVF controller cannot damp the forced vibrations. However, the vibration amplitudes are reduced due to increased dynamic stiffness of the modes. All unfiltered natural frequencies of the machine, which are between the high-pass filter (i.e. 15% of the

Table 1
Controller and filter parameters of the DVF controller: (A) SPINNER U5-620, (B) PITTLER PV 630.

f_n	(A) 58.0 Hz; (B) 46.4 Hz;	$K_{v,stat}$	(A) 407; (B) 613
ω_w	$0.15 \cdot 2\pi \cdot f_n$	ζ_w	0.5
ω_{HP}	$0.04 \cdot 2\pi \cdot f_n$	Actuator data sheet:	
T	8.5 ms	ζ_A	(A) 0.06; (B) 0.07
$F_{Act,max}$	45 N	ω_A	(A) 214,9 rad/s; (B) 208,9 rad/s