Available online at www.sciencedirect.com

ScienceDirect

Procedia CIRP 46 (2016) 396 – 399

www.elsevier.com/locate/procedia

7th HPC 2016 – CIRP Conference on High Performance Cutting

Comparison of different control strategies for active damping of heavy duty milling operations

Robin Kleinwort[a]*, Martin Schweizer[a], Michael F. Zaeh[a]

[a]*Institute for Machine Tools and Industrial Management, Technical University of Munich (TUM), Boltzmannstraße 15, 85748 Garching, Germany*

* Corresponding author. Tel.: +49-(0)89-289-15571; Fax: +49-(0)89-289-15555. *E-mail address:* Robin.Kleinwort@iwb.tum.de

Abstract

Depending on the machining process chatter might occur at an eigenfrequency of the machine's structure. Electrodynamic proof-mass actuators can be attached to the structure in order to mitigate chatter. This paper gives an overview of different existing control strategies for active damping and compares them with one another. First, the control strategies were implemented and tested in a coupled simulation model. Then, the simulation results were validated by modal tests. For a sample process the analytically predicted depths of cut were finally verified in cutting tests.

Keywords: Machine tool; Chatter; Control; Active Damping

1. Introduction

The maximum material removal rate of machine tools is determined by either the drive capacity of the spindle or - more often - by the stability limit. Under certain conditions the machining process can become instable and chatter occurs. Chatter causes high dynamic forces on the machine's bearings, a poor surface finish and high tool wear.

The active vibration control (AVC) system used in this research is able to mitigate chatter caused by the eigenmodes of the machine tool's structure except the spindle shaft or the tool. This publication focuses on the comparison of several control strategies used for active damping.

A very popular approach is to use collocated control strategies as proposed by [1,2]. Collocated control is characterized by collocated actuator and sensor pairs. One widely-used possibility for collocated control is the direct velocity feedback (DVF) controller, successfully tested in cutting tests by [3,6].

Another method often used for disturbance rejection purposes are model based linear quadratic regulators (LQR), usually in combination with state space observers, as proposed by [1] and successfully deployed on machine tools by [4,5].

The last two control strategies considered in this paper belong to the field of robust control. H_∞-control considers unstructured uncertainties and robust stability demands in the design process [2]. If structured uncertainties occur, μ-synthesis control should be used, which also considers robust performance demands. There exist several approaches for both robust control strategies to use them for active damping of machine tools [2,4].

For the first time, this paper compares the performance of these different control strategies in modal and cutting tests.

2. Active damping system

Fig. 1 shows the collocated placement of the main components: the actuator (SA10-V30 by CSA Engineering) and the acceleration sensor (KS 813B by MMF). They are placed in an antinode of the machine's most critical eigenmode. The actuator is driven by an amplifier (BAA 120 by BEAK) and as a proof-mass actuator it can be attached to the machine's structure at any arbitrary position.

doi:10.1016/j.procir.2016.04.054

The control rule is implemented on a rapid prototyping system (MicroLabBox by dSpace) with a sample frequency of 10 kHz.

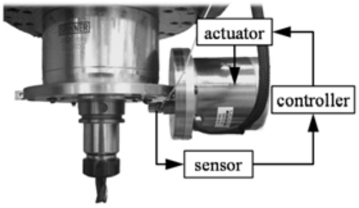

Fig. 1. machine tool with actuator and sensor.

3. Simulation model

The simulation model consists of two components: one model for the mechanical structure of the machine tool and one model for the actuator unit, which includes the actuator itself and the amplifier. Because of its almost ideal transfer characteristics the sensor transfer function was neglected in the modelling process.

The model of the mechanical structure of the machine tool was obtained via experimental modal analysis. It includes the first five eigenmodes of the machine tool. The behavior of the machine tool is described by a second order differential equation, where $[M]$, $[D]$ and $[K]$ are the mass, damping and stiffness matrix respectively, $\{q\}$ the displacements in Cartesian coordinates and $\{F\}$ the applied forces:

$$[M]\{\ddot{q}\}+[D]\{\dot{q}\}+[K]\{q\}=\{F\}. \tag{1}$$

Following [1], the transfer function of a proof-mass actuator can be described in the form of:

$$G_A\,(s)=g_A\;\frac{s^2}{s^2+2\zeta_A\omega_A s+\omega_A^2}\,, \tag{2}$$

with g_A being the gain factor, ζ_A the damping factor and ω_A the natural eigenfrequency of the actuator. In order to identify the parameters in eq. (2) the transfer function of the actuator unit was determined experimentally. As fig. 2 depicts, not only the transfer function of the actuator itself but the transfer function of the actuator unit consisting of actuator and amplifier could be approximated very well in the relevant frequency range (20-200 Hz) after adjusting the parameters.

Both models for the mechanical structure of the machine tool and the actuator unit were consolidated in a state space model of the form:

$$\begin{aligned}\{\dot{x}\}&=[A]\{x\}+[B]\,u,\\ y&=[C]\{x\}+[D]\,u,\end{aligned} \tag{3}$$

where $[\boldsymbol{A}]$, $[\boldsymbol{B}]$, $[\boldsymbol{C}]$ and $[D]$ are the system matrix, input matrix, output matrix and transition matrix respectively. u is the input quantity, in this case the control voltage at the actuator unit. On the other hand, y is the output quantity, which is the measured acceleration at the sensor position. The state vector $\{x\} = \{\{\boldsymbol{q_m}\}, \{\dot{\boldsymbol{q}}_{\boldsymbol{m}}\}, \{\boldsymbol{x_A}\}\}$ includes the states of the mechanical structure in modal coordinates $\{\boldsymbol{q_m}\}$ and the states of the actuator unit $\{\boldsymbol{x_A}\}$.

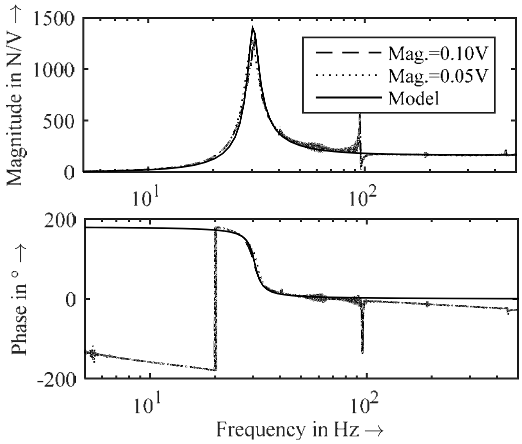

Fig. 2. actuator transfer function at two magnitudes and the identified model.

4. Control strategies for active damping of machine tools

In the following section the basics of collocated, optimal and robust control are presented.

4.1. Collocated control

Compared to other collocated control strategies, the DVF controller shows the best performance on machine tools [6]. Collocation between the actuator and the sensor leads to higher stability of the control loop [1]. A velocity feedback controller acts as a viscous damper by applying an actuator force F_{act} proportional to the measured velocity signal to the system:

$$F_{act}=-K_C\dot{x}\,, \tag{4}$$

where the gain K_C is the only variable, which has to be adapted in a way that the controller remains stable.

4.2. Optimal control

Optimal control is characterized by the desire to control a dynamic system at minimum cost. If the system can be described by a set of linear differential equations and the cost by a quadratic equation, the problem can be solved by the linear-quadratic regulator (LQR). The cost function, which has to be minimized, is:

$$J=\frac{1}{2}\int_0^\infty x^T Q\,x+u^T R\,u\,dt\,, \tag{5}$$

where Q and R are - usually diagonal - weight matrices. It has two contributions: The first one is the term $x^T Qx$, which makes sure that the controlled state space vector entries approach zero after an initial displacement in a speed corresponding to their