Active Control of Machine-Tool Vibration in Cutting Operations in a General Lathe

L. Pettersson†, L. Håkansson†, I. Claesson† and S. Olsson‡

† Department of Telecommunications and Signal Processing
Blekinge Institute of Technology
372 25 Ronneby
Sweden

‡ Active Control, IDEON
223 70 Lund
Sweden

Abstract– **In the general lathe the turning operation chatter or vibration is a frequent problem affecting the result of the machining, and, in particular, the surface finish. Tool life is also influenced by vibration. Severe acoustic noise in the working environment frequently occurs as a result of dynamic motion between the cutting tool and the workpiece. These problems can be reduced by active control of machine-tool vibration.**

In a general lathe the physical features and properties of an active control system usually limit their applicability. By using piezo ceramic actuators some constraints regarding the mechanical construction of the active tool holder can be circumvented.

An active control solution for a general lathe application has been developed. It is based on a standard industry tool holder with an embedded piezo ceramic actuator and an adaptive feedback controller. The adaptive controller is based on the well known filtered-x LMS-algorithm. It enables substantial reduction of the vibration level by up to 40 dB at 3.4 kHz.

I. Introduction

In turning operations the tool and tool holder shank are subjected to dynamic excitation due to the deformation of work material during the cutting operation. The stochastic chip formation process usually induces vibrations in the machine-tool system. Energy from the chip formation process excites the mechanical modes of the machine-tool system. Modes of the workpiece may also influence the tool vibration. The relative dynamic motion between cutting tool and workpiece will affect the result of the machining, in particular the surface finish. Severe acoustic noise is also introduced, the noise level is sometimes almost unbearable to the machine operator. The tool life is also likely to be correlated with the amount of vibrations. It is well known that vibration problems are closely related to the dynamic stiffness of the structure of the machinery and workpiece material. The vibration problem may be solved in part by proper machine design which stiffens the machine structure. In order to achieve further improvements the dynamic stiffness of the tool holder shank can be increased more selectively.

This paper discusses the single-channel feedback control of tool vibration in the cutting speed direction. The single channel control system is illustrated in Fig. 1 below. The tool holder used in

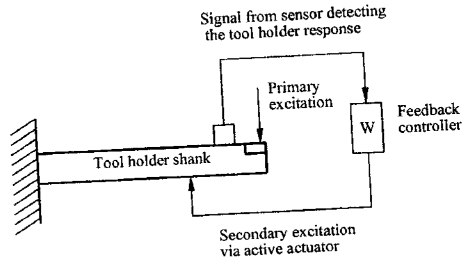

Figure 1: A machine-tool feedback control system[2].

this application has an embedded piezo ceramic actuator, i.e. secondary source, which have been developed at the department of telecommunications and signal processing. The construction of the tool

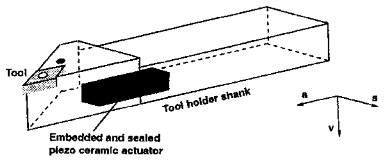

Figure 2: Tool holder with embedded actuator for the control of tool vibration in the metal cutting process.

holder is shown in Fig. 2.
Active control of machine-tool vibration is a solution to these problems. Generally, machine tool systems are classified as narrow band systems and as a consequence tool shank vibrations can usually be described as a superposition of narrow band random processes at each modal frequency [1]. These when added together form a more wide band random process. The tool vibrations in a turning operation mainly comprise vibrations in two directions; the cutting speed direction and the feed direction [1]. Usually, the vibrations in the cutting speed direction and the feed direction are linearly independent, except at some of the eigenfrequencies [1]. Consequently, the control problem involves the introduction of secondary sources driven in such way the anti-vibrations generated by means of these sources will interfere destructively with the tool vibration. However, in external longitudinal turning, most of the vibrations are induced in the cutting speed direction. Thus, the control of the vibrations in the cutting speed direction is an adequate solution to the vibration problem [2]. In order to control the vibrating modes of a tool holder it is essential to select a location for the actuators that enables the introduction of secondary vibration in to these modes. However, the location for the mounting of the actuators must be selected carefully to avoid unnecessary reconstructions and/or performance reductions of the the lathe. By embedding piezo ceramic actuators in a standard industry tool holder the active control of tool vibration is enabled to a general lathe. A complication in the turning operation is that the original excitation of the tool vibration cannot be observed directly and can therefore not be used as a feedforward control signal. A solution to the controller problem is to control the adaptive FIR filter with the leaky version of the well-known filtered-x LMS-algorithm [2].

II. Experimental Set-up

The cutting trials have been carried out in a Mazak SUPER QUICK TURN - 250M CNC turning centre with 18.5 kW spindle power, maximal machining diameter 300mm, 1007 mm between the centres. The tool holder construction is based on an embedded design with an piezo ceramic stack actuator and an accelerometer mounted on the cutting tool to make it possible to measure the vibrations in the cutting speed direction. In order to operate the piezoelectric stack actuator a custom designed amplifier was used. A digital signal processor controller was used and the measurements were carried out on a two-channel signal analyzer. Furthermore, a two channel low-pass filter was used to adjust the input level to the A/D converter and the output level from the D/A converter.

III. Work Material - Cutting Data - Tool Geometry

The workpiece material SS 2541-03, chromium molybdenum nickel steel [1], was used in the experiments. This work material excites the machine-tool-system with a narrow bandwidth in the cutting operation. After a preliminary set of trials a suitable combination of cutting data and tool geometry was selected, see table 1. The combination was selected to cause significant tool vibrations which resulted in an observable deterioration of the workpiece surface and severe acoustic noise. The diameter of the workpiece was deliberately chosen large (over 100 mm), in order to render the workpiece vibrations negligible.

Geometry	Cutting speed, v (m/min)	Depth of cut, a (mm)	Feed s (mm/rev)
DNMG 150508-SL TN 7015	80	0.9	0.25

Table 1: Cutting data and tool geometry.

IV. Active Tool Holder

In order to control the vibrating modes of a tool holder it is essential to select a location for the actuators that enables the introduction of secondary vibration in to these modes. However, the location for the mounting of the actuators must be selected carefully to avoid unnecessary reconstructions and/or performance reductions of the the lathe. The tool vibration or bending motion in the tool holder, introduced by the stochastic chip formation process may be attenuate by introducing a opposite bending moment in the tool holder. By

mounting the actuator in the area of peak modal strain and optimising the actuator offset distance to the centre axis of the tool holder, a suitable control force my be introduced by a voltage induced actuator strain. Hence, the bending deformation of the tool holder introduce a axial deformation of the actuator and by producing an equal and opposite control force, the tool vibration is reduced.
The principle of the active tool holder with embedded actuator is illustrated in Fig. 2.

V. Adaptive Control of Tool Vibration

The original excitation of the tool vibrations, originating from the material deformation process, cannot be directly observed. Consequently, the controller for the control of machine-tool vibration is based on a feedback approach. The response of the tool holder can be measured with a sensor mounted on the machine-tool. By introduction of secondary anti-vibrations with a secondary source, actuator, the response of the tool holder can be modified [2]. The actuator is governed by a controller which is fed with the accelerometer signal sensing the vibrations of the tool holder. A block diagram of the feedback control system is shown in Fig. 3.
The objective of the control is to minimize the mean square error. The use of the error signal as input signal to the adaptive FIR filter controlling the plant, will cause the adaptive FIR filter to act as a feedback controller. This will complicate the relation between the mean square error and the filter coefficients, i.e. the mean square error will not be a quadratic function of the filter coefficients. In fact the mean square error function may be multimodal in the filter coefficients [3]. The search for a minimum on the mean square error surface can be performed by the well-known filtered-x LMS algorithm defined by [2]:

$$y(n) = \mathbf{w}^T(n)\mathbf{x}(n) \tag{1}$$

$$e(n) = d(n) - y_C(n) \tag{2}$$

$$\mathbf{x}_{C^*}(n) = \mathbf{c}^{*T}\mathbf{x}(n) \tag{3}$$

$$\mathbf{w}(n+1) = \mathbf{w}(n) + \mu\mathbf{x}_{C^*}(n)e(n) \tag{4}$$

where $\mathbf{x}_{C^*}(n)$ is the filtered reference signal vector. A block diagram of the feedback control system with the filtered-x LMS algorithm is shown in Fig. 3. In Fig. 3 the box with the unit delay operator q^{-1} at the input to the controller handle the fact that we are dealing with an adaptive digital filter in a feedback control system. Observe the feedback relation from $x(n) = e(n-1)$. Furthermore, C represents the dynamic secondary system (forward path) under control, i.e. the electro-mechanic response. The estimate of this path is denoted C^*. It is in practice customary to use an estimate of the

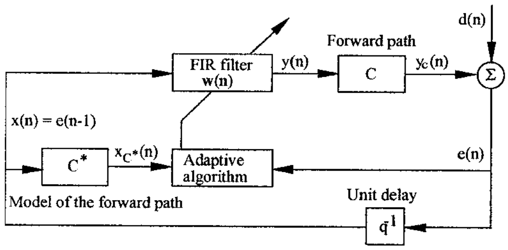

Figure 3: Equivalent block diagram of the feedback control situation with the filtered-x LMS algorithm[2].

impulse response for the forward path. As a result, the reference signal $\mathbf{x}_{C^*}(n)$ will be an approximation, and differences between the estimate of the forward path and the true forward path influence both the stability properties and the convergence rate of the algorithm [4, 5, 6, 7].
The estimation error can be expressed as follows [2];

$$e(n) = \frac{1}{1 + C(q)W(n,q)q^{-1}}\, d(n) \tag{5}$$

where delay-operator notation is used.
From this expression it is obvious that the poles of filter, i.e. the poles of the transfer function between the desired signal $d(n)$ and the estimation error $e(n)$, are affected by the controller response. The stability of the feedback control systems thus depends on the ability of the filtered-x LMS-algorithm to control the adaptive FIR filter, the time varying controller response, without violating the closed loop stability requirements, i.e. the Nyquist stability criterion [8]. In feedback control, limiting the energy in the control signal to the plant yields a more robust behaviour. By introducing leakage in the filtered-x LMS-algorithm the "memory" of the adaptive algorithm is reduced thereby reducing the energy in the response of the adaptive FIR filter and also the energy in the control signal to the plant.
The leaky version of the filtered-x LMS-algorithm is obtained through a modification of the algorithm for the coefficient vector adaption of the filtered-x LMS-algorithm with a leakage factor γ. As a result, the algorithm for the coefficient vector adaption of the leaky version of the filtered-x LMS-algorithm is given by [7]:

$$\mathbf{w}(n+1) = \gamma\mathbf{w}(n) + \mu\mathbf{x}_{C^*}(n)e(n) \tag{6}$$

The leakage factor γ is a real positive parameter which satisfies the condition:

$$0 < \gamma < 1 \tag{7}$$

The identification of the secondary path was performed in an initial phase. A second adaptive FIR filter was governed by a LMS algorithm while the system was excited with a broadband Pseudo Random, PN, signal. The fixed FIR filter estimate of the forward path was subsequently used to prefilter the input signal to the algorithm for the adaptation of the coefficient vector in the filtered-x LMS algorithm. For the control of tool vibration a 20-tap adaptive FIR filter was used together with a 35-tap FIR filter estimate of the secondary path [2]. These filter lengths were at the limit for the processing capacity of the signal processor used. A 16 kHz sampling rate was chosen for the digital filter. In order to minimize delay in the loop, no anti-aliasing or reconstruction filters were used. Obviously, this necessitates extra care being taken in order to avoid aliasing.

VI. Results

The tool shank vibrations considered in this paper originate from the cutting speed direction of the tool holder shank. To illustrate the effect of feedback control of tool vibration in the cutting speed direction, the spectral densities of the tool vibrations with and without feedback control are shown in Figs. 4 and 5. Figure 4 shows a typical result obtained with adaptive feedback control of tool-vibration. It performs a broad-band attenuation of the tool-vibration and manage to reduce the vibration level with up to approximately 40 dB at 3.4 kHz. If a leakage factor is introduced in to

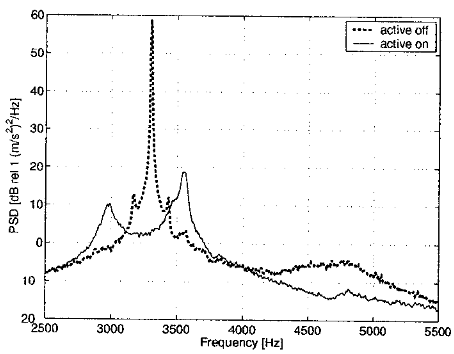

Figure 4: The spectral density of tool vibration with 20 tap FIR filter feedback control (solid) and without (dashed). Step length μ = -1, cutting speed v = 80 m/min, cut depth a = 0.9 mm, feed rate s = 0.25 mm/rev, tool DNMG 150508-SL , grade 7015. without leakage factor.

the filtered-x LMS-algorithm the performance of the adaptive control system will be reduced. This is illustrated by the spectral density given in Fig. 5.

In the experiments, it was observed that the adaptive feedback control of tool vibration resulted

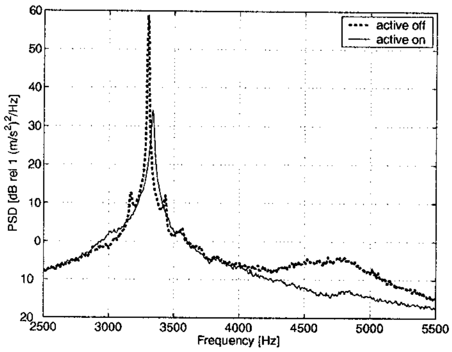

Figure 5: The spectral density of tool vibration with 20 tap FIR filter feedback control (solid) and without (dashed). Step length μ = -1, cutting speed v = 80 m/min, cut depth a = 0.9 mm, feed rate s = 0.25 mm/rev, tool DNMG 150508-SL , grade 7015. with leakage factor $\gamma = 0.9999$.

in a significant improvement of the workpiece surface. In Fig. 6 a photo of the workpiece used in the experiments is shown. In order to illustrate

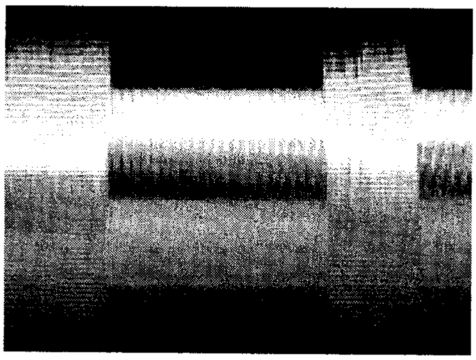

Figure 6: The workpiece surface with and without adaptive feedback control

the influence of leakage on the stability of the feedback control system, the significant part of the Nyquist plot, i.e. the part of the Nyquist plot closest to the point $(-1, 0)$, is given for estimates of the open loop frequency response with and without leakage in the filtered-x LMS algorithm. Fig. 7 shows the Nyquist plot for the case of no leakage; and Fig. 8 shows the Nyquist plot for the case of leakage in the filtered-x LMS algorithm.

VII. Conclusions and Future Work

It is clear from the results presented that tool vibrations in a lathe during metal cutting can be controlled by using a tool hoder with an embedded piezoelectric actuator and a adaptive FIR filter

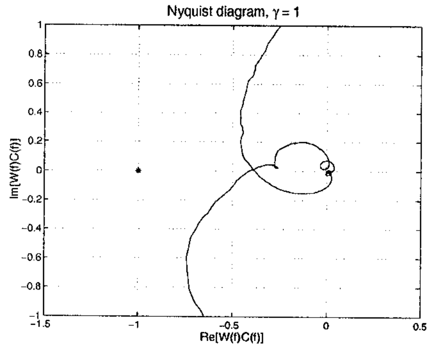

Figure 7: Nyquist diagram for the feedback control system with and without leakage, step length $\mu =$ -1, $\gamma = 1$.

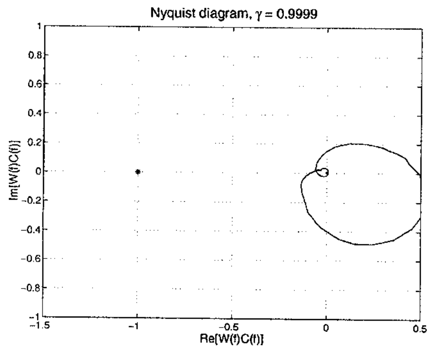

Figure 8: Nyquist diagram for the feedback control system with and without leakage, step length $\mu =$ -1, $\gamma = 0.9999$.

feedback controller. Furthermore, by embedding the actuator in the tool holder, the active control of tool vibration is likely be applicable in an arbitrary lathe.

The adaptive feedback control performs a broadband attenuation of the tool-vibration, and is able to reduce the vibration level by almost 40 dB at 3.4 kHz (see Fig. 4).

It is thus essential that the adaptive feedback control of machine-tool vibration handles the time varying environment. And indeed, the leaky filtered-x LMS algorithm appears to have great potential with respect to the feedback control of tool vibrations in the turning operation. It has been reported that the leaky version of the feedback filtered-x LMS-algorithm is robust to large variations in the spectral properties of tool vibration [2].

From a manufacturing point of view the improvements of the surface is of great importance. The surface is a result of a more stable cutting operation due to adaptive control system. The reduced noise level around the lathe is also important. Today, the industry are facing more and more regulations concerning the noise level in the working area of the employees . It is also interesting to note that the adaptive technique does not affect the cutting data, it may even allow an increase of the material removal rate. It is also likely that the tool life, which is an important cost to the manufactures, will increase.

The leakage factor reduce the magnitude of the frequency response of the adaptive FIR filter, i.e. causes the loop gain of the control system to be reduced and is so doing increases the distance between the trajectory of the open loop frequency response and the point $(-1, 0)$. This can be observed by comparing the Nyquist diagram for the open loop response for the control system when the filtered-x LMS algorithm is used to control the response of the adaptive FIR filter shown in Figs. 7 and 7. The control system is thus likely to be more robust in the Nyquist sense when the leaky filtered-x LMS algorithm controls the response of the adaptive FIR filter.

Future work will focus on to transfer the active vibration control technique to boring operations. Further study of different control algorithms is also urgent.

Acknowledgment

The project is sponsored by The Foundation for Knowledge and Competence Development.

References

[1] P-O. H. Sturesson, L. Håkansson, and I. Claesson. Identification of the statistical properties of the cutting tool vibration in a continuous turning operation - correlation to structural properties. Journal of Mechanical Systems and Signal Processing, Academic Press, 11(3), July 1997.

[2] I. Claesson and L. Håkansson. Adaptive active control of machine-tool vibration in a lathe. IJAV-International Journal of Acoustics and Vibration, 3(4), 1998. Invited.

[3] S.J. Elliott. Active control using feedback. Technical Report 732, Institute of Sound & Vibration Research, University of Southampton, January 1994.

[4] S.J. Elliott and P.A. Nelson. Active noise control. IEEE signal processing magazine, pages 12–35, October 1993.

[5] P.A. Nelson and S.J. Elliott. Active Control of Sound. Academic Press, Inc, 1992.

[6] B. Widrow and S.D. Stearns. Adaptive Signal Processing. Prentice-Hall, 1985.

[7] S.M. Kuo and D.R. Morgan. Active Noise Control Systems. Telecommunications and Signal Processing. Weiley, 1996.

[8] K.J. Åström and B. Wittenmark. Computer Controlled Systems, Theory and Design. Prentice Hall, 1984. ¡