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ACTIVE CONTROL MACHINE TOOL CHATTER PIEZO CERAMIC ACTUATORS IN TOOL HOLDER SHANK

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Abstract

In the turning operation chatter or vibration is a frequent problem, which affects 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. Adaptive feedback control based on the filtered-x LMS-algorithm, enables a reduction of the vibration by up to 40 dB at 1.5 kHz and by approximately 40 dB at 3 kHz. The active control performed a broadband attenuation of the sound pressure level by up to 35 dB . A significant improvement of the workpiece surface was also observed. In the active control of tool vibration a tool holder construction based on integrated high magnetostrictive actuators was used. However, both the physical features and properties of a active tool holder construction based on high magnetostrictive actuators and the fact that this type of actuators generally have a non-linear behaviour highly reduce its applicability to the general lathe and turning operation. Therefor, a new generation embedded active tool holder shanks based on piezo ceramic actuators have been developed. Based on spectrum estimates, both coherence spectrum and frequency response function estimates has been calculated for both the old tool holder construction and the new generation active tool holder shank. From the results it follows that the phase delay is smaller and the linearity of the new generation active tool holder shank are superior compared to the old technology. It is also obvious that physical features and properties of new generation embedded active tool holder shanks based on piezo ceramic actuators fits the general lathe application.

NOMENCLATURE

$y(n)$	Output signal from adaptive FIR filter
$\mathbf{w}(n)$	Coefficient vector of adaptive FIR filter
$\mathbf{x}(n)$	Vector of input signal samples to the adaptive FIR filter
$e(n)$	Estimation error signal
$d(n)$	Desired signal
$y_C(n)$	Output signal from forward path
μ	Step length of the adaptive algorithm
$\mathbf{x}_{C^*}(n)$	Vector of filtered input signal samples to the adaptive algorithm
M	Length of adaptive FIR filter
I	Length of the FIR filter estimate of the forward path
i	Integer
c_i^*	Coefficient of the FIR filter estimate of the forward path
γ	Leakage factor

1 INTRODUCTION

In the turning operation the tool and tool holder shank are subjected to a 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. Furthermore, the tool life is correlated with the amount of vibrations and the acoustic noise introduced. The noise level is sometimes almost unbearable. 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. A solution to these problems is active control of the tool vibrations. Generally, machine-tool systems are classified as narrow-band systems [1] and as a consequence tool shank vibrations can usually be described as a superposition of narrow-band random processes at each modal frequency. These when added together form a more wide-band random process [1]. The tool vibrations in a turning operation mainly comprise vibrations in two directions: the cutting speed direction and the feed direction [1, 2]. 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 two secondary sources, driven in such a way that the anti-vibrations generated by means of these sources interfere destructively with the tool vibration [3]. However, in external longitudinal turning, most of the vibration energy is usually induced in the cutting speed direction [1, 2]. It is thus likely that the control of tool vibration in the cutting speed direction is an adequate solution to the vibration problem [4, 2, 5] A complication in the turning operation is that the original excitation of the tool vibration the chip formation process cannot be observed directly and thus cannot be used as a feedforward control signal.

The statistical properties of the tool vibration imply a controller which utilizes the statistical correlation of the vibrations [6]. A classical statistical criterion is the mean square error criterion [7]. However, a controller based on this criterion cannot generally solve the control problem, since a such controller is only "optimum" in a stationary environment [8]. The statistical properties of the tool vibrations may vary during the machining process. Changes in cutting data and material properties influence the statistical properties of tool vibrations [1, 2]. Variation within the allowed cutting data interval may also influence the structural response of the tool holder [2] Variations within the excitation and the structural response of the tool holder influence the stability of the adaptive feedback control system. By using the well known leaky filtered-x LMS-algorithm to control the response of a FIR filter controller in the active feedback control of tool vibration, both robust stability and good performance are achieved [4] However, in the active control system for the control of tool vibration a tool holder construction with integrated high magnetostrictive actuators was used. Both the physical features and properties of a active tool holder construction based on high magnetostrictive actuators and the fact that this type of actuators generally have a non-linear behaviour highly reduce its applicability to the general lathe and turning operation. Non-linearities in the response of the forward path or plant under control is likely to degrade the performance of the control system. Therefor, a new generation embedded active tool holder shanks based on piezo ceramic actuators have been developed. Based on spectrum estimates, both coherence spectrum and frequency response function estimates has been calculated for both the old tool holder construction and the new generation active tool holder shank. From the results it follows that the phase delay is smaller and the linearity of the new generation active tool holder shank are superior compared to the old technology. It is also obvious that physical features and properties of new generation embedded active tool holder shanks based on piezo ceramic actuators fits the general lathe application. The tool holder construction with integrated high magnetostrictive actuators is shown in Fig. 2. Fig. 3 shows the new generation embedded active tool holder shanks based on piezo ceramic actuators. This paper discusses the single-channel feedback control of tool vibration in the cutting speed direction as well as a new generation embedded active tool holder shanks based on piezo ceramic actuators.

The controller was based on the well known filtered-x LMS-algorithm and the single channel control system is illustrated in Fig. 1 below.

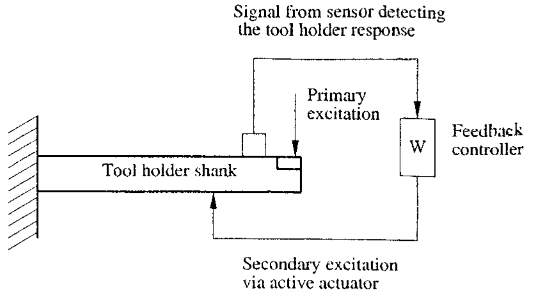

Figure 1: A machine-tool feedback control system[2].

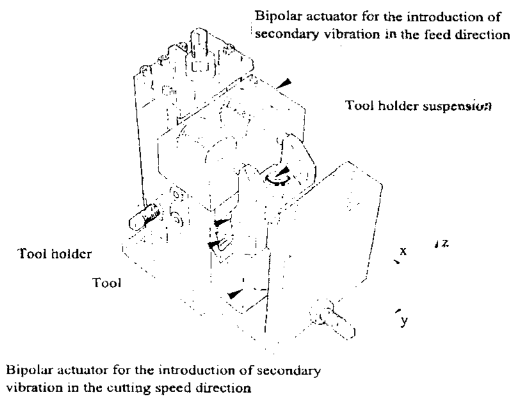

Figure 2: The old tool holder construction with integrated high magnetostrictive actuators.

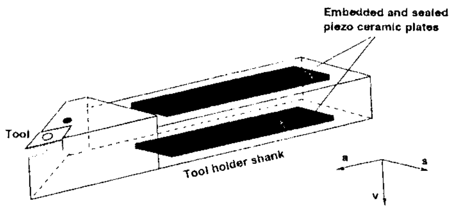

Figure 3: The new generation embedded active tool holder shank with piezo ceramic actuators.

2 MATERIALS AND METHODS

2.1 EXPERIMENTAL SET-UP

2.1.1 CUTTING EXPERIMENTS

The cutting experiments have been carried out on a Köping lathe with 6 kW spindle power using a tool holder construction with integrated actuators [9] and an accelerometer mounted on the cutting tool make it possible to measure the vibrations in the cutting speed direction. The tool holder construction is based on two bipolar actuators. The bipolar design is motivated by a desire to achieve linear behaviour, and is composed of two actuators that work with 180° phase difference. The actuators are based on high magnetostrictive material. In order to operate the bipolar actuator, a large current amplifier (5kW) 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.

2.1.2 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.

Geometry	Cutting speed, v (m/min)	Depth of cut, a (mm)	Feed s (mm/varv)
DNMG 150604-PF 4015	80	0.9	0.25

Table 1: Cutting data and tool geometry.

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 chosen large, over 100 mm. The workpiece vibrations can therefore be neglected.

2.1.3 SYSTEM IDENTIFICATION EXPERIMENTS

The system identification experiments has been carried out on two different forward paths. The first forward path is based on a tool holder construction with integrated high magnetostrictive actuators. The second forward path is based on the new generation embedded active tool holder shanks based on piezo ceramic actuators. To measure the vibrations in the cutting speed direction an accelerometer was mounted on the cutting tool. The new generation active tool holders are based on embedded and sealed piezo ceramic actuators in both the top and the bottom surface of the tool holder shank. The actuators in the top and the bottom surface work with 180° phase difference. The piezo ceramic actuators were powered by a 30 W power amplifier. A two-channel signal analyzer was used to generate a broadband Gaussian distributed excitation signal to the forward path, the frequency band of the used excitation signal was 10-15000 Hz . The excitation signal and the system response were recorded by a DAT recorder.

2.2 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 an accelerometer 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 steered 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. 1. Adaptive digital FIR filters based on the method of steepest descent are popular in various application areas, e.g. active control of sound [6, 8], active control of vibration [10, 2] and in other applications, such as electrical noise cancelling, system identification, adaptive beamforming, etc. [11, 12]. This is due to the simplicity of the implementation and their unimodal error surface in the feedforward application. A feedforward active controller can easily be controlled to converge towards a feasible solution [11]. Usually adaptive FIR filters are used in feedforward control [6, 8] but can also be used in feedback control [3, 13], even though there is no guarantee that the error surface will be unimodal under these conditions [14]. Similar problems can also be observed in feedforward control systems, when the control problem is ill conditioned. A method to improve such systems is to add a leakage factor to the adaptation algorithm [15]. This will also have the effect of limiting the energy in the impulse response of the adaptive filter and it has been observed to improve the robusness in the case of feedback control [4]. Furthermore, the leakage factor will also prevent accumulative build-up of bias in the coefficients of the adaptive filter [16]. The filtered-x LMS-algorithm is an adaptive filter algorithm suitable for active control applications [8] and is developed from the LMS algorithm, where a model of the the dynamic system between the filter output and the estimate, i.e. the forward path is introduced between the input signal and the algorithm for the adaptation of the coefficient vector [8, 11]. The filtered-x LMS-algorithm is given by the following four equations [8, 11];

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

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

$$\mathbf{w}(n+1) = \mathbf{w}(n) + \mu\mathbf{x}_{C^*}(n)e(n) \tag{3}$$

and $\mathbf{x}_{C^*}(n)$ is given by:

$$\mathbf{x}_{C^*}(n) = \begin{bmatrix} \sum_{i=0}^{I-1} c_i^* e(n-i-1) \\ \sum_{i=0}^{I-1} c_i^* e(n-i-2) \\ \vdots \\ \sum_{i=0}^{I-1} c_i^* e(n-i-M) \end{bmatrix} \tag{4}$$

where $c_i^*,\ i \in \{0,\ldots,I-1\}$ is an estimate of the impulse response of the forward (secondary) path. The leaky version of the filtered-x LMS-algorithm is obtained through a modification of the algorithm for the coefficient vector adaptation of the filtered-x LMS-algorithm with a leakage factor γ. Hence, the algorithm for the coefficient vector adaptation of the leaky version of the filtered-x LMS-algorithm is given by [11]:

$$\mathbf{w}(n+1) = \gamma\mathbf{w}(n) + \mu\mathbf{x}_{C^*}(n)e(n) \tag{5}$$

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

$$0 < \gamma < 1 \tag{6}$$

The controller used in the experiments described here is of the feedback type based on the well-known filtered-x LMS-algorithm [2, 13]. A block diagram of the feedback control system with the filtered-x LMS algorithm is shown in Fig. 4:

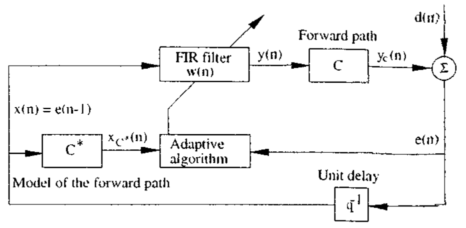

Figure 4: Equivalent block diagram of the feedback control situation with the filtered-x LMS algorithm[2].

In the above figure, C represents the dynamic secondary system (forward path) under control, i.e. the electro-mechanic response. The estimate of this path is denoted C^*.
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 16-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 15 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.

3 COHERENCE AND FREQUENCY RESPONSE FUNCTION ESTIMATION

A common measure on to which extent the output signal from a dynamic system can be linearly explaned from the systems input signal is given by the coherence function $\gamma^2_{xy}(f)$ which is given by the relation [17]:

$$\gamma^2_{xy}(f) = \frac{|S_{xy}(f)|^2}{S_{xx}(f)S_{yy}(f)} \tag{7}$$

where $S_{xy}(f)$ is the cross spectral density between the input signal $x(t)$ and the output signal $y(t)$, and S_{xx} and $S_{yy}(f)$ is the power spectral density for respective signal. Furthermore, a least-squares estimate of the frequency response function for a dynamic system can be obtained through [17]:

$$H_{xy}(f) = \frac{S_{xy}(f)}{S_{xx}(f)} \tag{8}$$

The power spectral densities for the input signal $x(t)$ and the output signal $y(t)$ as well as the cross power spectral density between them where estimated using Welch's method [18]. Based on these spectrum estimates both coherence function and frequency response function estimates were calculated.

4 RESULTS-SYSTEM IDENTIFICATION OF FORWARD PATHS

In Fig. 5 the estimate of the Coherence function for the forward path based on the tool holder construction with integrated high magnetostrictive actuators is shown. Fig. 6 shows a coherence function estimate for the forward path based on the new generation embedded active tool holder shank with piezo ceramic actuators. From Figs. 5 and 6 it is obvious compared with the tool holder construction with integrated high magnetostrictive actuators that the new generation embedded active tool holder shanks based on piezo ceramic actuators can to a much greater extent be explained by a linear system. In

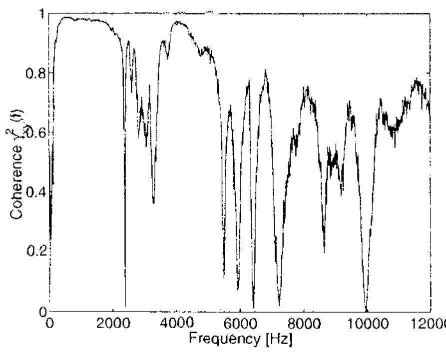

Figure 5: Coherence function estimate for the forward path based on the tool holder construction with integrated high magnetostrictive actuators.

Fig. 7 the estimated Bode diagram for the forward path based on the tool holder construction with integrated high magnetostrictive actuators is presented. Fig. 8 shows an estimate of Bode

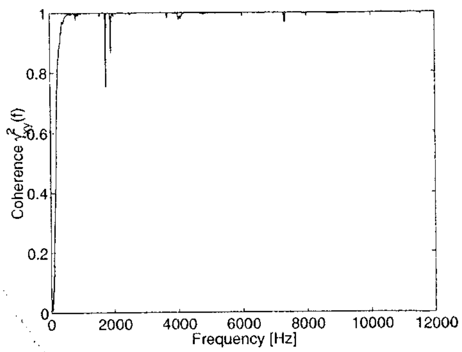

Figure 6: Coherence function estimate for the forward path based on the new generation embedded active tool holder shank with piezo ceramic actuators.

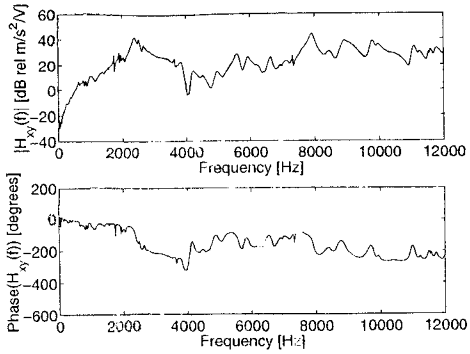

Figure 8: Frequency response function estimate for the forward path based on the new generation embedded active tool holder shank with piezo ceramic actuators.

diagram for the forward path based on the new generation embedded active tool holder shank with piezo ceramic actuators. From Figs. 7

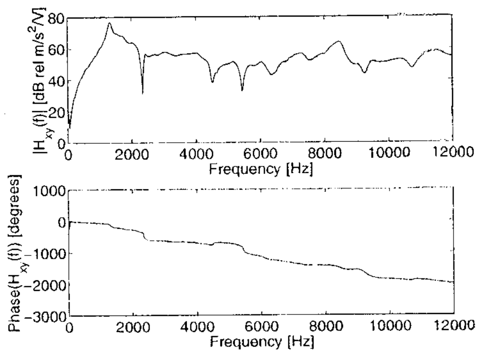

Figure 7: Frequency response function estimate for the forward path based on the tool holder construction with integrated high magnetostrictive actuators.

and 8 it follows directly that the new generation embedded active tool holder shanks based on piezo ceramic actuators introduce a significantly smaller phase delay.

5 RESULTS-ACTIVE CONTROL OF TOOL VIBRATION

The tool shank vibrations considered in this paper originate from the cutting speed direction of the tool holder shank in the tool holder construction with actuators based on high magnetostrictive material. 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 given. Fig. 9 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 simultaneously at 1 5 kHz and 3 kHz.

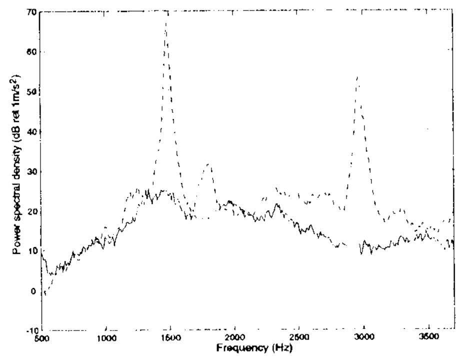

Figure 9: The power spectral density of tool vibration with 20-tap FIR filter feedback control (solid) and without (dashed). Step length $\mu = 0.05$, cutting speed $v = 80$ m/min, cut depth $a = 0.9$ mm, feed rate $s = 0.25$ mm/rev, tool DNMG 150604-PF, grade 4015 [4].

Fig. 10 shows the vibration spectrum obtained using four different settings of the leakage factor in the adaptive algorithm controlling the 20-tap FIR filter feedback controller.

The sound pressure was measured in the operator area for the lathe and a typical result obtained with adaptive feedback control of tool-vibration is shown in Fig. 11. It performs a

broad-band attenuation of the sound pressure in the frequency band 1.5 kHz to 25 kHz, with up to approximately 35 dBSPL at 3 kHz.
In the experiments, it was observed that the adaptive feedback control lead to a significant improvement of the workpiece surface. In Fig. 12 a photo of the workpiece used in the experiments is shown.

6 CONCLUSIONS AND FUTURE WORK

It is clear that tool vibrations in a lathe during metal cutting can be controlled using an active control system. The tool holder shank vibrations are fed into an actuator via a digital controller. Further, the well-known filtered-x LMS-algorithm, traditionally used as a feedforward controller, seems to have great potential with respect to feedback control of tool vibrations in the turning operation. The adaptive feedback control performs a broad-band attenuation of the tool-vibrations, and is able to reduce the vibration level by up to approximately 40 dB simultaneously at 1.5 kHz and 3 kHz (see Fig. 9). Furthermore, in the operator area for the lathe, the vibration control results in a broad-band attenuation of the sound pressure in the frequency band 1.5 kHz to 25 kHz, with up to approximately 35 dBSPL at 3 kHz (see Fig. 11).

The introduction of a leakage factor or a "forgetting factor" in the recursive coefficient adjustment algorithm will induce bias in the coefficient vector and thereby cause a somewhat reduced attenuation of the tool-vibration, see Fig. 10. By controlling the FIR filter controller with the leaky filtered-x LMS-algorithm good performance and robust control of tool vibration are achieved [4]. In comparison with the filtered-x LMS-algorithm, the leaky filtered-x LMS-algorithm introduce a substantial improvement in the robustness of the feedback control of tool vibration in the Nyquist sense [4].

From coherence function estimates in the Figs. 5 and 6 it follows that the linearity of the new generation active tool holder shank is superior compared to the old technology. It is also obvious from Figs. 7 and 8 that the new generation active tool holder shank introduce a significantly smaller phase delay than the old tool holder construction with integrated high magnetostrictive actuators. Both the fact that the phase delay is smaller and that the linearity of the new generation active tool holder shank are

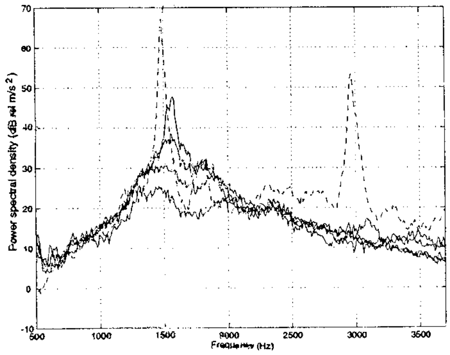

Figure 10: The power spectral densities of tool vibrations with 20-tap FIR filter feedback control and four different leakage factors (solid), and spectral density without feedback control (dashed). Leakage factors $\gamma = 1, 0.9999, 0.999, 0.99$, step length $\mu = 0.05$, cutting speed $v = 80$ m/min, cut depth $a = 0.9$ mm, feed rate $s = 0.25$ mm/rev, tool DNMG 150604-PF, grade 4015 [4].

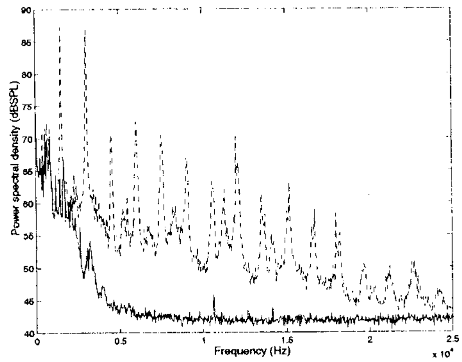

Figure 11: The Power spectral density of sound pressure with 20-tap FIR filter feedback control (solid) and without (dashed). Step length $\mu = 0.05$, cutting speed $v = 80$ m/min, cut depth $a = 0.9$ mm, feed rate $s = 0.25$ mm/rev, tool DNMG 150604-PF, grade 4015.

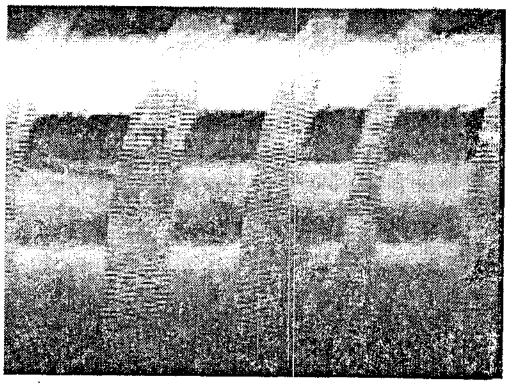

Figure 12: The workpiece surface with and without adaptive feedback control[4].

superior compared to the old technology is advantageous for the feedback control of tool vibration. It is also obvious that physical features and properties of new generation embedded active tool holder shank based on piezo ceramic actuators are superior compared to the old technology in the application.

From a manufacturing engineering point of view, the significant improvement of the workpiece surface, see Fig. 12, achieved with the adaptive feedback control of the tool vibration, is of great importance. The reduction of the noise introduced by the tool vibrations is also an important feature. 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, e.g. increase the productivity. Further, it is well known that there exists a correlation between tool vibrations and tool life. It is therefore likely that the adaptive feedback control of the tool vibration extends the tool life.

Future work includes for example the investigation of Internal Model Control based on the filtered-x LMS algorithm in the application. A theoretical foundation for the behaviour of the filtered-x LMS algorithm in this application is also urgent.

ACKNOWLEDGEMENT

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