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ACTIVE CONTROL OF CHATTER VIBRATIONS IN A TURNING PROCESS USING AN ADAPTIVE FXLMS ALGORITHM

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Abstract

In this paper a 2-DOF lumped mass model is used for dynamic modelling of a turning process. The cutting forces are modelled using predictive machining theory based on the Merchant cutting force model. Cutting parameters such as shear angle are calculated using experimental relations. Variation of cutting angles is also considered. Regarding the governing delayed differential equations, the model is simulated in the time domain using a Matlab/Simulink program. There is a good agreement between simulation results and the simulated and experimental results obtained by Xiao et al. [1]. Then, chatter vibration is suppressed using a single channel feedback FXLMS control method. The adaptive FIR filter generates proper canceller signals for the piezoactuators to reduce tool vibrations. The control method has the advantage of considering the statistical vibration of the cutting process that would be rarely considered in other active control methods. By using adaptive filters, it is possible for the controller to always be in the optimum condition by changing the FIR filter coefficients based on measured error signals. A significant reduction in vibration amplitude of the tool is achieved by using this active vibration control method. The response of the controller is observed for different actuator positioning. In addition, the performance of this control method is evaluated in the presence of disturbances in the vibration signals.

1. INTRODUCTION

Machining operations are often limited by chatter vibration. This kind of vibration has some adverse effects on surface roughness, dimensional accuracy, tool wear, and machine tool life. It also causes undesirable noise in work environment. The most important source of chatter vibration is regenerative vibration between the work piece and the tool holder construction. These vibrations are caused by dynamic interaction of the machine tool structure and the chip formation process. Over the past decades several researches in both modelling of machining process and chatter suppression methods have been done. Liang et al. have reviewed the researches which have been done in machining process monitoring methods and some control techniques in recent years [2].

Many researches have been done on modelling of chatter vibration in machine tools. Most of the analytical models and analysis of chatter vibration in machining are usually based on a number of simplifying assumptions, such as linear cutting force calculation. However, chatter is basically a nonlinear process. An analytical model of chatter vibration in metal cutting has been presented by Tarng et al. [3]. The cutting forces in this model have been calculated using the predictive machining theory based on the shear zone model of chip formation process. Variations of the undeformed chip thickness and the rake angle due to the machine tool vibration have been taken into account in determination of the cutting forces. Nonlinearities such as cutting process damping, variation of cutting angles and the tool jump from work piece have been included in this model. X. Li & H. Li presented a predictive time domain chatter model for the simulation and analysis of chatter vibration in a milling process [4]. In this model, dynamic regeneration effects have been taken into account by considering the effect of tool vibration on the instantaneous undeformed chip thickness.

Several active or passive methods have been used over the years to suppress chatter. Because of the undesirable effects of chatter vibration in machining operation, several researches have been done to suppress the regenerative chatter. Classical chatter theories produced a stability lob diagram to avoid the occurrence of chatter. These stability lobes have been utilized in the selection of chatter free spindle speeds. Al-regib et al. [5], used a spindle speed variation method for the reduction of chatter vibrations. In another work, Xiao et al. [1], used the vibration cutting method to avoid self-excitation in the machining process. The basic principle of these methods is disruption of the regeneration effects. Even though, the need for special DC motors or some other special tool positioners, limit the capability and usage of these methods. Some researchers such as Tarng et al. [6], used vibration absorbers for the suppression of chatter. Since the physical parameters of the absorber are constant, this method is optimum just in a stationary constant. Also in an experimental investigation, Hakansson et al. applied an active controller on a turning machine for suppressing chatter vibrations [7].

In this paper a 2-DOF model of a machine tool structure is used for the simulation of chatter vibrations in a turning process. The cutting forces are calculated using the orthogonal cutting theory, based on the Merchant analytical model. Chatter vibration is suppressed using an adaptive feedback FXLMS control method. The adaptive FIR filter generates proper canceller signals for the piezoactuator to reduce the tool vibrations. In the following sections, the theoretical background and the model of the cutting process are presented. Then, the numerical simulations of the model and control algorithm are described.

2. THEORETICAL MODELLING OF THE CUTTING PROCESS

Machine tool chatter is a self-excited vibration caused by the interaction of chip removal process and the structure of machine tool. The most important type of chatter is regenerative vibration. This kind of vibration usually occur when a favourable phase relationship develops between the inner and the outer modulation, caused by vibration during two consecutive tooth passes. The conventional model of a single degree of freedom machining system is shown in Figure1 [5]. In this model, the resultant cutting force F(t) is proportional to the instantaneous uncut chip thickness h(t) as expressed by:

$$F(t) = K_{ab}\, b\, h(t) \ , \qquad h(t) = h_0 + x(t) - x(t-\tau) \tag{1}$$

Where b is the axial depth of cut and K_{ab} is the static cutting stiffness. In calculation of the cutting forces, the parameter K_{ab} is usually considered as a constant parameter. But in practice, the value of K_{ab} is usually varied due to the variations of other cutting parameters. In this relation the instantaneous uncut chip thickness h(t) composed of the mean uncut chip thickness h_o, the inner modulated cut surface x(t), due to the current tooth pass, and the outer modulated surface $x(t-\tau)$, due to the previous tooth pass. Here τ is the time delay between the two consecutive cuts and represents the regenerative feedback effect. The dynamic equation of the system can be written as a second order differential equation. In this relation ζ is the damping ratio and ω_n is the natural frequency of the machining operation.

$$\ddot{x}(t) + 2\zeta\omega_n \dot{x}(t) + \omega_n^2 x(t) = K_c b[h_0 + x(t) - x(t-\tau)] \qquad (2)$$

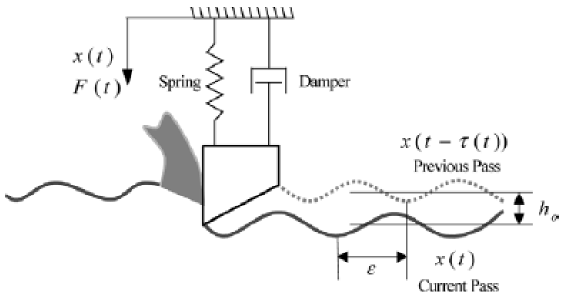


Figure 1. Single degree-of-freedom model of cutting process [5].

In this paper, the cutting system is taken as a 2-DOF vibration system and the mechanical parameters are chosen from the experimental data in reference [1]. This model, which is illustrated in Figure2, provides the tool geometry factors leading to chatter. Also the experimental cutting conditions are presented in Table 1. With the use of these experimental data in analytical model of the cutting process, the simulated results can be verified with the experimental results which are obtained in the same cutting conditions.

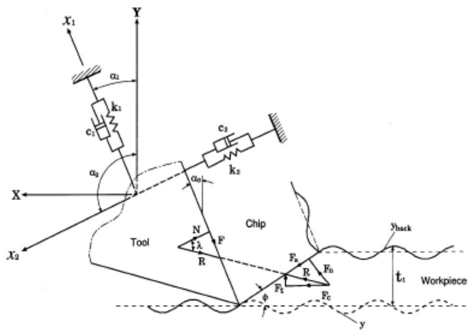


Figure 2. Vibration model of the chip formation in orthogonal cutting [1].

Table 1. Experimental cutting conditions.

Parameter	*Value*	*Parameter*	*Value*
Spindle speed, (S)	460 rpm	Feed rate, (f)	0.051 mm/rev
Cutting speed, (V_S)	58 m/min	Rake angle, (α_0)	3 deg
Depth of cut , (h_0)	0.05 mm	Clearance angle, (γ_0)	7 deg

Equation (3) represents the dynamic equations of motion of the above mentioned system in the principal directions of the tool holder construction. In these equations “m” is the mass coefficient, c_1 and c_2 are the damping coefficients and k_1 and k_2 are the stiffness coefficients.

$$m\ddot{x}_1 + c_1 h_{x_1} \dot{x}_1 + k_1 x_1 = F_1 \quad ; \quad m\ddot{x}_2 + c_2 h_{x2} \dot{x}_2 + k_2 x_2 = F_2 \qquad (3)$$

The vibration system is excited in the X_1 and X_2 directions by the cutting forces F_1 and F_2, determined by the following expression:

$$\begin{cases} F_1 = F_C Sin(\alpha_1 - \zeta) + F_t Cos(\alpha_1 - \zeta) \\ F_2 = F_C Sin(\alpha_2 - \zeta) + F_t Cos(\alpha_2 - \zeta) \end{cases} \tag{4}$$

The parameter ζ indicates the variation of the tool angles and α_1 and α_2 are the angles of principal directions. In order to determine the cutting force F_c and the thrust force F_t, Tarng et al. used an orthogonal machining theory [3]. Since this theory is extremely complex, an experimental database is applied to predict the cutting force components. With Merchant's analytical model for orthogonal cutting [1], the cutting force F_c and the thrust force F_t can be represented by:

$$\begin{gathered} F_C = R\, Cos(\beta - \alpha), \quad F_t = R\, Sin(\beta - \alpha) \\ R = \frac{K_{ab}\; h\; b}{Sin\,\phi\; Cos(\phi + \beta - \alpha)}\, U(h) \end{gathered} \tag{5}$$

Where R is the total cutting force, α represents the rake angle and U(h) is defined as a unit step (U(h) = 0 for h<0). The shear angle ϕ, the friction angle β and the shear stress K_{ab} may be determined by [1]:

$$\begin{cases} \phi = \exp(\; 0.0587\,v + 1.039h + 0.6742\alpha - 1.2392) \\ \beta = \exp(\; -0.0546\,v - 0.8856h + 0.8923\alpha - 0.2388) \\ K_{ab} = \exp(\; 0.0059\,v - 0.4246h + 0.818\alpha + 6.3211) \end{cases} \tag{6}$$

The cutting database can easily determine the cutting force components. Thus, an available cutting model supporting various materials is presented by using the experimental database.

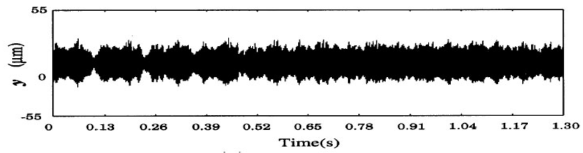
4. NUMERICAL SIMULATION

The equations of Motion, which are used for the modelling of chatter in the cutting process, have been solved numerically in the time domain using a Matlab/Simulink program. Cutting parameters have been chosen the same as experimental cutting conditions in the reference [1]. The simulated hardware parameters are as follows: m=0.19 kg, c_1=52, c_2=63 Ns/m, k_1=0.7, k_2=1.12 MN/m and $\alpha_1 = 20°$, $\alpha_2 = 110°$.

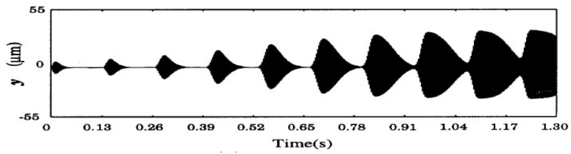
Figure3-a shows the experimental result of the tool displacement, corresponding to the simulation results obtained by Xiao et al., (Figure3-b). Also the simulated result of the presented model in this paper is indicated in Figure3-c. The tool displacement increases at the beginning of the simulation due to the unstable chatter and is constrained to finite amplitude.

5. CONTROL ALGORITHM

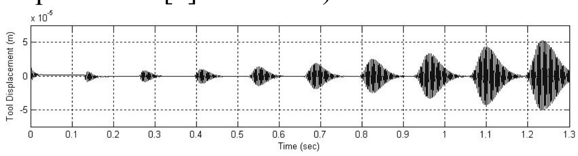
As mentioned before, several methods have been used in last decades to suppress the regenerative vibration in machining operations. Usually, the basic principle of these methods is to prevent the dynamic of machining process from locking on the most favourable phase for chatter. Most of these methods, suppress chatter vibrations by changing the cutting condition and moving the cutting process to a more stable zone.



a) Experimental tool displacement [1]



b) Simulated results of tool displacement [1]



c) Simulation results of machine tool model

Figure 3. Tool displacement vs. time obtained from: a) experimental results, b) simulation results presented in Ref. [1], c) simulation results of the present work.

The cutting force is a vibratory force with variable amplitude. The frequency and amplitude of this force would be changed due to the changes of other cutting parameters. According to the random behaviour of chatter vibration, classical control algorithms are not very useful for chatter control. So, in these applications, adaptive active controllers are very useful. The FxLMS algorithm is an adaptive filter which is widely used in the active vibration or noise control systems.

The cutting force is generated from the material deformation process. So it can not be directly observed. For this reason, the machine tool vibration control strategy should be as a feedback control method [7]. In vibration control problems, the suppression of undesired signals is usually done by imposing a secondary anti-vibration signal to the system. So in this paper the vibration of tool would be suppressed by generating an external secondary vibration signal which is generated by the control algorithm. A block diagram of this feedback control system is shown in Figure4.

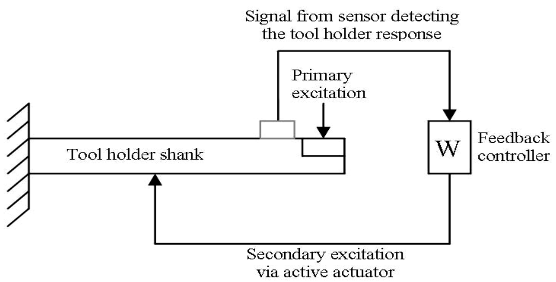


Figure 4. Feedback control system for tool holder vibration.

Adaptive digital FIR filters which work based on the method of steepest descend, are popular in various application areas. Active control of sound, active vibration control systems and other applications like system identification are some examples of adaptive FIR filter usages. These adaptive filters are usually used in the feedforward applications. However, they can be used in a feedback control system. The filtered-x LMS-algorithm is an adaptive filter which is suitable for active noise or vibration control applications. This algorithm is developed from the LMS algorithm. This adaptive filter is a coefficient vector w(n), which acts on the input signal x(n). The filter coefficients are time variable and can be changed by the adaptation law. A schematic model of this adaptive feedback controller is shown in Figure5 [8].

The secondary signal y(n) is the cancellation signal which is generated by the FIR filter and is sent to the actuators:

$$y(n) = \sum_{l=0}^{L-1} w_l(n)\, x(n-l) \tag{7}$$

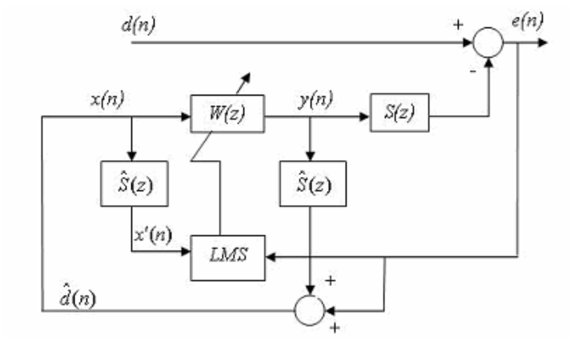


Figure 5. Feedback AVC system using FxLMS algorithm [8].

In this relation, $w_l(n), l=0,1,2,...,L-1$ represent the coefficients of FIR filter and L indicates the length of the filter. These filter coefficients are updated using the LMS algorithm:

$$w_l(n+1) = w_l(n) + \mu \, x'(n-l) \, e(n) \ , \quad l = 0,1,2,...,L-1 \tag{8}$$

In Equation (8), μ is the step length of the adaptive algorithm and e(n) indicates the error signal which should be measured from the displacement sensors. The parameter $x'(n)$ is the filtered input signal which is sampled to the adaptive algorithm.

In machining process, the only measurable signal is the error signal, which can be obtained by measuring the tool tip displacement. In this case, the separation between the reference signal (cutting forces) and the cancellation signal (actuator forces) is impossible. So there is not any reference signal to be used in the LMS algorithm and for this reason, feedforward controllers can not be implemented on this plant. In the feedback control system, the reference signal is virtually constructed from the measured error signal and the cancellation signal:

$$x(n) = \hat{d}(n) = e(n) + \sum_{m=0}^{M-1} \hat{s}_m \, y(n-m) \tag{9}$$

In this model $S(z)$ is the forward path (actuator) transfer function in Z domain, and $\hat{S}(z)$ is a FIR filter which is used for estimation of the forward path in the controller. By considering the tool as a beam model, the forward path transfer function (in the Laplacian domain) can be obtained from the frequency response solution of the beam model using the finite element method. In the beam model, the dynamics of the actuator and sensor have also been considered. Bilinear transformation has been used in order to obtain the transfer function in the Z domain. Because of the lack of space, the details of the above mentioned modelling procedure of the forward path transfer function has not been presented in this paper.

6. NUMERICAL RESULTS

In section 4 the chatter phenomenon in the cutting process has been simulated and the results have been validated with the previous work presented by Xiao et al. [1]. In this section, the adaptive feedback controller, using the FxLMS algorithm, has been implemented on the plant. The tool displacement results, obtained from the cutting process simulation, have been used as error signal in the controller. The selected adaptive filter is a 15-tap FIR filter and the adaptation law would update the FIR filter coefficients according to the error signal and the constructed reference signal, in order to minimize the measured error signal.

The effect of step length μ on the controller response has been observed by simulating the system with different values of μ. In the adaptive feedback controller using FXLMS algorithm, the step length can be changed from zero to about 0.5 without any controller instability. The results show that the error signal is decreased more rapidly in higher values of the step length (Figure6). For the case $\mu = 0$, the tool displacement has a transient response in the first two seconds and after that the vibration amplitude remains constant. In the transient region, behaviour of the error signal is varying too fast. Therefore, in this region, the FIR filter coefficients can not be updated rapidly and proper canceller signal would not be generated. However, after few seconds, the response of the active tool becomes satisfactory.

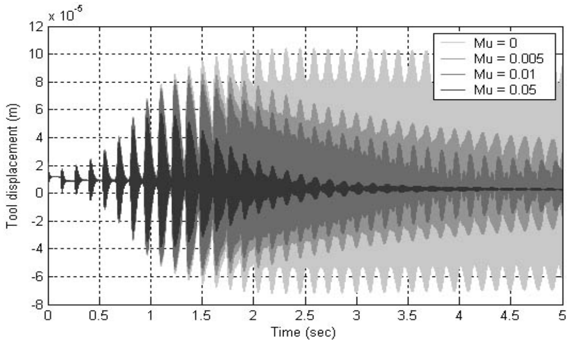


Figure 6. Displacement of active tool holder versus time for different values of μ.

The performance of the active tool holder controller with the presence of additional external disturbance has been observed by introducing a secondary signal as a disturbance to the machine tool model. In practice, the disturbance signal may cause from external sources of vibration like the power transmission system. This signal is considered to be at the range of vibration properties of the main model. The disturbance signal is a sinusoidal wave with the amplitude of 5 microns and frequency of 100 Hz. The generated signal has been added to the simulated model of machine tool vibration. Because of the increment in frequency richness of the final error signal, it would be better to change the FIR filter length from 15-taps to 20 or 25-taps. The simulation results are shown in Figure 7. The obtained results show a decrease in amplitude of tool vibration. It decreases from 5 microns at the beginning of the simulation to about 2 microns at the end of simulation.

Finally, the effect of the actuator positioning on the controller performance has been investigated. For this purpose, the piezoelectric actuator is placed in three different positions on the tool holder body. The positions are near the tool tip (position 3), at the middle of the tool holder (position 2) and near the tool holder support (position 1). In each case, the forward path transfer function S(z), should be obtained from the tool holder model. The active tool holder model is simulated for each actuator position. The simulation results are shown in Figure 8. The results indicate a significant improvement in the controller performance by placing the actuator near the tool holder support.

7. CONCLUSIONS

In this paper, the active vibration control of a machine tool with the presence of chatter instability has been investigated. The single channel feedback controller based on the FxLMS algorithm has been implemented on the 2-DOF model of the machining process. The model of the machine tool vibration has been validated with the experimental results of the previous works. The adaptive controller has effective influence on chatter reduction in the turning model. This method of chatter suppression does not affect the cutting parameters and it may allow an increase of the material removal rate by reducing the tool vibration. On the other hand, reduction in the tool vibrations will lead to an increase in tool life. The performance of the active tool holder in the presence of external disturbance in tool vibration has also been observed and acceptable reduction in both machine tool chatter and disturbance has been obtained. In addition, the near optimum location for the actuator has been obtained by investigating the performance of the controller in different actuator positions.

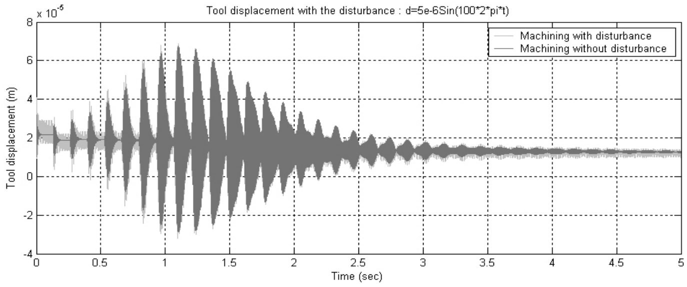


Figure 7. Active control of tool vibration with the presence of external disturbance.

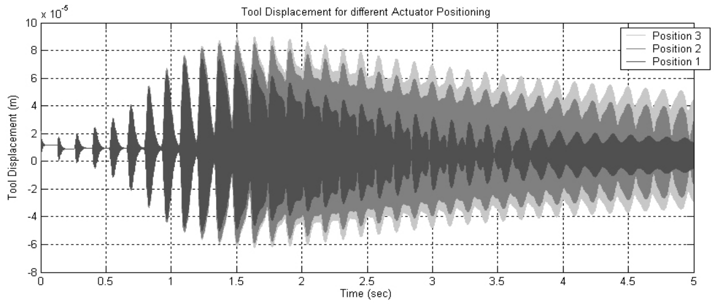


Figure 8. Tool displacement vs. time for different actuator positions.

REFERENCES

[1] M. Xiao, S. Karube, T. Soutome and K. Sato, "Analysis of chatter suppression in vibration cutting", *International Journal of Machine Tools Manufacture* **42**, 1677-1685 (2002).

[2] S.Y. Liang, R.L. Hecker and R.G. Landers, "Machining process monitoring and control: The State-of-the-Art", *Journal of Manufacturing Science and Engineering* **126**, 297-310 (May 2004).

[3] Y.S. Tarng, H.T. Young and B.Y.Lee, "An analytical model of chatter vibration in metal cutting", *International Journal of Machine Tools Manufacture* **34-2**, 183-197 (1994).

[4] H. Li and X. Li, "Modelling and simulation of chatter in milling using a predictive force model", *International Journal of Machine Tools Manufacture* **40**, 2047-2071 (2000).

[5] E. Al-Regib, J. Ni and S.H. Lee, "Programming spindle speed variation for machine tool chatter suppression", *International Journal of Machine Tools Manufacture* **43**, 1229-1240 (2003).

[6] Y.S. Tarng, J.Y. Kao and E.C. Lee, "Chatter suppression in turning operations with a tuned vibration absorber", *Journal of Materials Processing Technology* **105**, 55-60 (2000).

[7] L. Hakansson, I. Claesson, L. Pettersson and T. Lago, "Active control machine tool chatter piezo ceramic actuators in tool holder shank", *Proceeding of ASME Design Engineering Technical Conferences*, Las Vegas, Nevada, Sep 1999, pp. 2675-2683.

[8] S.M. Kuo and D.R. Morgan, *Active noise control systems, algorithms and DSP implementations*, John Wiley & Sons, 1996.