2018 15th International Conference on Control, Automation, Robotics and Vision (ICARCV)
Marina Bay Sands Expo and Convention Centre, Singapore, November 18-21, 2018

EXTENDED MODEL AND CONTROL OF REGENERATIVE CHATTER VIBRATIONS IN ORTHOGONAL CUTTING

Ziv Brand
Dept. of Mechanical Engineering
NRCN, Beer Sheva, Israel
brand.ziv@gmail.com

Shai Arogeti
Dept. of Mechanical Engineering
BGU, Beer Sheva, Israel
arogeti@bgu.ac.il

Abstract — **Turning machining is an important manufacturing process, widely used in industry. Dynamic interaction between the tool and the workpiece may cause regenerative chatter, which is associated with problems of poor surface finish, reduced product quality and low productivity. The demand for high accuracy motivates development of active vibration control methods that are based on realistic dynamical models of the turning process. This paper discusses the development of an active robust control law that is based on an extended regenerative chatter vibration model for orthogonal cutting. Its novelty stems from the way the workpiece elastic behavior is taken into consideration. The presented numerical results show that the vibration level can be reduced significantly, even in the presence of external disturbances, parametric uncertainty and (open loop) unstable machining conditions.**

Index Terms — **regenerative chatter, orthogonal cutting, active control, UDE control, time delay.**

I. INTRODUCTION

Chatter is a common dynamic phenomenon in cutting processes, expressed as self-excited vibrations caused by the interaction of the cutting tool and the workpiece structure. Chatter has several negative effects, for example, poor surface quality, lower productivity, short life span of cutting tools, damage for machine components, disturbing or even dangerous noise and waste of material and energy. For these reasons, chatter avoidance is a topic of wide interest [1]. Chatter may be caused by regeneration of waviness at the workpiece surface. This regenerative effect is the most dominant cause of chatter. The different conditions, which cause a stable cutting processes (i.e. no chatter) or unstable (i.e. with chatter), can be formulated with respect to the width of cut and spindle speed. This representation is known as the stability lobe diagram (SLD). The diagrams allows choosing stable machining conditions [25]. The theory of chatter regeneration has been discussed by many researchers [1]-[7].

One approach to reducing the regenerative effect is by active control methods. Shiraishi et.al [5] designed an optimal control law using the linear time-invariant (LTI) orthogonal cutting model and Padé approximant to describe the regenerative phenomena. They obtained good experimental results, which demonstrated that active chatter control is possible. The authors of [6] have suggested an active control strategy for self-excited vibrations in orthogonal cutting conditions, where the control law was based on linear control and LTI models. Chen et al. [7] presented an active magnetic damping method for boring bars. They designed H_∞ optimal controllers to widen the chatter stability zone. Their dynamic model of the boring bar was achieved through modal testing, where the dynamic behavior of the workpiece was not considered.

This paper suggests a new designing approach to active control of regenerative chatter in orthogonal cutting. This approach is based on an extended dynamical model of the orthogonal turning process, where the model coefficients are influenced by changes of cutting position (i.e., the tool location along the workpiece). In addition, the model is formulated with uncertainties, representing the uncertain part of the model coefficients and an external disturbance. This extended model poses new challenges in the design of active control for regenerative chatter in orthogonal cutting.

To deal with the suggested complex model, the Uncertainty and Disturbance Estimator (UDE) method, proposed in [8]-[9], is utilized for the controller design. The basic principle of this control method lays on a quick disturbance estimation and compensation mechanism. The UDE control algorithm takes advantage of the principle that a signal with a specific bandwidth can be approximated through its low pass filtering. The UDE based control strategy has been successfully applied for robust input-output linearization in [10-13], and was combined with sliding-mode control in [14-16]. Moreover, the method has been extended to uncertain linear [17,18] and nonlinear [9,19] systems with state delays. The UDE control method consists of three design elements: a reference model, an error feedback gain and a filter. The selection of a meaningful reference model is typically straightforward, however only brief guidelines were given in literature for the design of the feedback gain and the filter. The set-point response is determined by the reference model, while the response due to a disturbance is determined by the error feedback gain and filter. In general, the error feedback gain is chosen so that the high-frequency components of the uncertainty and disturbance signal can be attenuated, while attenuation of the low-frequency elements are done by the filter. Hence, the design of the error feedback gain and the design of the filter are decoupled in the frequency domain. The contribution of the approach presented in this paper stems from the suggested complex model and the solution based on UDE, which is implemented in this field, for the first time.

The paper is organized as follows. The dynamic model of regenerative chatter vibration in orthogonal cutting is introduced in Section II. Section III presents the implementation of the UDE based control law. Two case studies with numerical results are presented in section IV, and Section V concludes the paper.

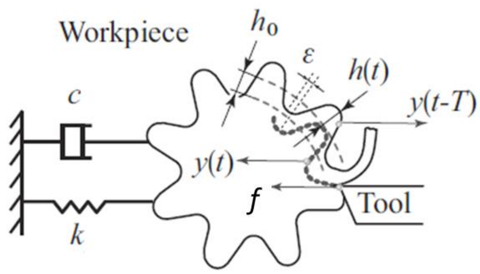

Figure 1. Single–degree-of-freedom orthogonal cutting model[1]

II. DYNAMIC MODELING OF REGENERATIVE CHATTER VIBRATION IN ORTHOGONAL CUTTING

A single–degree-of-freedom (SDOF) orthogonal cutting model is depicted in Fig. 1. Here, f is the radial dynamic cutting force, y is the relative displacement between the tool and the workpiece (in a direction normal to the workpiece surface), $c = c_0 + \vartriangle c$, $k = k_0 + \vartriangle k$ and $m = m_0 + \vartriangle m$ are respectively, the equivalent of damping, stiffness and mass of the workpiece. Assuming that the workpiece motion is approximated as a SDOF in the radial direction, the equation of motion of the system can be expressed as follows,

$$\begin{aligned}(m_0 + \vartriangle m)\ddot{y}(t) + (c_0 + \vartriangle c)\dot{y}(t) + (k_0 + \vartriangle k)y(t) = \\ f(t) + u(t) + \omega(t).\end{aligned} \tag{1}$$

In (1), $u(t)$ is an external dynamic force (to be used later, for control) and $\omega(t)$ is an unknown disturbance. c_0, k_0 and m_0 are respectively, the nominal known equivalents of damping, stiffness and mass of the workpiece. The symbol $\vartriangle$ indicates the unknown part of a model coefficient. The uncertainty here is mainly due the unknown cutting position along of the workpiece (position x). Since the workpiece elasticity is assumed meaningful, the location of the interaction between the tool and the workpiece has strong influence on model coefficients.

The equivalent stiffness k in (1), as a function of the workpiece position x, is defined by,

$$k(x) = \frac{\bar{f}}{w(x)}. \tag{2}$$

Where $w(x)$ is the radial deformation of the workpiece at x (the axial coordinate), due to a radial force $\bar{f}$, acting at the same location of the deformation. Generally, this force represents the cutting and the external control forces. According to mechanics theory, the deflection of a tailstock-supported workpiece at its axial location x [21] is,

$$\begin{aligned} w(x) &= \frac{\bar{f}x^3}{3EI} - \frac{\bar{f}}{2}\frac{x^2(3L-x)}{L^3}\frac{x^2(3L-x)}{6EI} \\ &= \frac{\bar{f}x^3\left[4L^3 - x(3L-x)^2\right]}{12EIL^3}. \end{aligned} \tag{3}$$

Substituting the deflection term (3) in (2) gives the workpiece stiffness $k(x)$ in (1),

$$k(x) = \frac{12L^3EI}{x^3\left[4L^3 - x(3L-x)^2\right]}. \tag{4}$$

The natural frequency of a tailstock-supported workpiece in its first vibration mode can be expressed analytically with clamped-hinged boundary conditions [20], as,

$$\omega_{n1} = 15.4\sqrt{E \cdot I / \rho AL} \tag{5}$$

where E is the elastic modulus, I is the cross-sectional moment of inertia, A is the equivalent cross-section area of the workpiece, L is the length of workpiece and ρ is the material density. The cross-sectional moment of inertia for a cylindrical workpiece is,

$$I = \pi\frac{d^4}{64}.$$

where d is the diameter of the workpiece.

The equivalent mass in (1) can be defined as,

$$m(x) = \frac{k(x)}{\omega_{n1}^2} \tag{6}$$

while the equivalent damping c is,

$$c = 2 \cdot \zeta \cdot \rho \cdot A \cdot L \cdot \omega_{n1} \tag{7}$$

and ζ is the damping ratio of the workpiece.

According to regenerative chatter theory, the initial surface of the shaft is smooth without waves during the first revolution, but the tool starts leaving a wavy surface behind. It is caused by the bending vibrations of the shaft in the feed direction $y(t)$, which is in the direction of the radial cutting force f (see Fig.2). In the second revolution (see Fig.1), the surface has waves both inside (represented by $y(t)$) and outside (given by $y(t-T)$ of the cut nominal surface. The latter is due to vibrations during the previous revolution. Hence, the resulting chip thickness is no longer a constant, but it varies as a function of the vibration frequency and the angular speed of the workpiece (i.e., spindle speed). Then, the chip thickness can be expressed as follows [21],

$$h(t) = h_0(t) - \left[y(t) - y(t-T)\right]. \tag{8}$$

where, h_0 is the target chip thickness, which is determined by the feed rate of the machine. Here, T represents a time delay of a single spindle revolution.

[1] From Y.Altintas, *Manufacturing Automation*, Second Edition, Cambridge University Press.

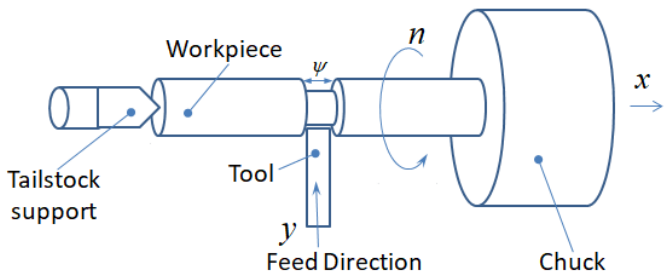

Figure 2. Orthogonal Plunge turning

The feed cutting force $f(t)$ is proportional to the cutting constant in the feed direction (K_f), to the width of cut Ψ and to the chip thickness $h(t)$, hence,

$$f(t) = K_f \Psi h(t). \tag{9}$$

The cutting coefficient K_f may also change, depending on the magnitude of the instantaneous chip thickness and the orientation of the tool and workpiece. These phenomena add complexity to the description of the dynamic cutting process [21]. Accordingly, the cutting stiffness K_f is defined as,

$$K_f = K_{f0} + \vartriangle K_f \tag{10}$$

where as before, the index 0 represents the nominal known value, and the rest stands for the unknown element of the coefficient.

Referring to Fig. 1, the instantaneous depth of the cut, $h(t)$, decreases as the workpiece moves away from the cutting tool, i.e., when $y(t)$ increases. Furthermore, if the vibration is too large (i.e., $y(t) - y(t-T) > h_0(t)$), then the tool jumps out of the cut, thus producing a zero chip thickness and zero cutting force. This phenomenon is undesirable.

Substituting (8), (9) and (10) into (1) yields,

$$\begin{aligned}&(m_0 + \vartriangle m)\ddot{y}(t) + (c_0 + \vartriangle c)\dot{y}(t)\\&+(k_0 + \vartriangle k + K_{f0}\psi + \vartriangle K_f \psi) y(t)\\&-(K_{f0} + \vartriangle K_f)\psi y(t-T) - (K_{f0} + \vartriangle K_f)\psi h_0(t)\\&= u(t) + \omega(t).\end{aligned} \tag{11}$$

The dynamic model in (11), for regenerative chatter vibrations in orthogonal cutting, can be categorized as an uncertain linear time-invariant (LTI) system with an unknown external disturbance and time delay. Its structure is illustrated by the block diagram in Fig. 3 (see the dashed rectangle that represents the controlled process). Two feedback paths can be distinguished; a negative position feedback (which is the primary path) and a positive feedback of the delayed position

$$y(t-T) = y(t) * \delta(t-T); \int_{-\infty}^{\infty} \delta(t-T)dt = 1, \tag{12}$$

which is the regenerative path. In (12), the symbol $*$ stands for a convolution operation, and the dynamics of the workpiece (Σ , in Fig. 3) is,

$$\begin{aligned}\Sigma : &(m_0 + \vartriangle m)\ddot{y}(t) + (c_0 + \vartriangle c)\dot{y}(t)\\&+(k_0 + \vartriangle k + K_{f0}\psi + \vartriangle K_f \psi) y(t) = f_t(t).\end{aligned} \tag{13}$$

The maximum allowable axial cut depth, $\psi_{\lim}$, can be represented by the SLD plot; see Fig. 4. In Fig. 4 the Regions below the lobes are stable, while the regions above the lobes are unstable. The maximum axial depth $\psi_{\lim}$ is given ([21, 22]) as,

$$\psi_{\lim} = \frac{-1}{2K_f G(\omega_c)}. \tag{14}$$

where $G(\omega_c)$ represents the real part negative value of the transfer function in (11), and $\omega_c[rad/s]$ (or $f_c[Hz]$) is chatter vibration frequency. As to the maximum spindle speed,

$$N = \frac{60}{\tau}, \quad \tau = \frac{2n\pi + \varepsilon}{2\pi f_c}, \quad n = 0,1,2,3,\dots. \tag{15}$$

where n is the integer number of waves, ε represents the phase difference between the inner and outer modulations, $\tau[s]$ is the spindle period and $N[rev/\min]$ represents spindle speed.

III. IMPLEMINATION OF A UDE BASED CONTROL LAW

The general principles of linear and nonlinear UDE based control laws, for uncertain nonlinear plants with time delay and disturbance, can be found in [8, 9].

The dynamic model of orthogonal cutting (11) can be reformulated as a state space model,

$$\begin{aligned}\dot{x}(t) &= Ax(t) + Bu(t)\\&+F\big(x(t), x(t-\tau(t)), d(t)\big) + \omega(t)\end{aligned} \tag{16}$$

where, $x = [x_1 \quad x_2]^T$ is the state vector (with, $x_1 = y,\ x_2 = \dot{y}$) and u is the control input. The matrices A and B contain the nominal part of the model coefficients, and it equal,

$$A = \begin{pmatrix} 0 & 1 \\ -\dfrac{k_o + k_{f0}}{m_0} & -\dfrac{c_0}{m_0} \end{pmatrix}, \quad B = \begin{pmatrix} 0 \\ \dfrac{1}{m_0} \end{pmatrix}.$$

The term $F\big(x(t), x(t-\tau(t)), d(t)\big)$, on the other hand, is assumed unknown; it includes the influence of the uncertain model coefficients and the delayed state. It is given by,

$$F=\begin{bmatrix}0 & \frac{1}{m_0}\begin{pmatrix}\psi K_f\left(h_0+x_1(t-T)\right)-\triangle m\dot{x}_2(t)\\ -\triangle c x_2(t)-\left(\triangle k+\psi\triangle K_f\right)x_1(t)\end{pmatrix}\end{bmatrix}^T.$$

The desired specifications are determined by the choice of a reference model,

$$\dot{y}_m=A_m y_m+B_m r(t) \tag{17}$$

where $r(t)$ is a reference signal. For the presented application, the reference model is chosen as a second order system, formulated as,

$$\begin{pmatrix}\dot{y}_m(t)\\ \ddot{y}_m(t)\end{pmatrix}=\begin{pmatrix}0 & 1\\ -\omega_m^2 & \zeta_m\omega_m\end{pmatrix}\begin{pmatrix}y_m(t)\\ \dot{y}_m(t)\end{pmatrix}+\begin{pmatrix}0\\ \omega_m^2\end{pmatrix}r(t) \tag{18}$$

where, ω_m and ζ_m determine the closed loop desired dynamics.

The goal of the UDE controller is to make the state error vector,

$$e=\left(y_m-y,\dot{y}_m-\dot{y}\right)^T \tag{19}$$

between the system and the reference model, converge to zero. In other words, to force the error dynamics to behave as,

$$\dot{e}=\left(A_m+K\right)e \tag{20}$$

where $K=diag\left(K_1,K_2\right)$ is called the error feedback gain. Since the reference model is often chosen to be stable, K may be taken as zero. If robust stability is required, then common robust control strategies can be used for the choice of K.

Combining (16), (17), (19) and (20), we obtain

$$A_m x+B_m r(t)-Ax-Bu(t)-F-\omega=Ke \tag{21}$$

Then, the control action is determined as,

$$u(t)=B^+\left[A_m x+B_m r(t)-Ax-F-\omega-Ke\right] \tag{22}$$

where $B^+=\left(B^TB\right)^{-1}B^T$ is the pseudo inverse of B. Since $u(t)$, given in (22), is only an approximated solution of (21), the relations in (20) and (21) are not always met. Substituting (22) into (21) leads to the following structural constraint,

$$\left[I-BB^+\right]\left[A_m x+B_m c-Ax-F-\omega-Ke\right]=0 \tag{23}$$

Obviously, if B is invertible, the above constraint is always achievable. If not, the choice of the reference model and the error-feedback gain-matrix is limited. Moreover, structural restrictions on the unknown parts F and ω can be derived such that the above constraint is feasible.

A signal, denoted hereafter by $u_d(t)$, includes the required compensation of the uncertainties and external disturbance. According to the system (16), $u_d(t)$ can be represented as

$$u_d(t)=-B^+\left[F+\omega\right]=B^+\left[-\dot{x}+Ax+Bu\right] \tag{24}$$

Hence, the unknown dynamics and disturbance can be observed by the system states and the control signal. However, it cannot be used in the control law directly. The signal $u_d(t)$ can be accurately approximated by the UDE method, as,

$$ude(t)=u_d(t)*g_f(t) \tag{25}$$

where $g_f(t)$ is the impulse response of a strictly proper low-pass filter $G_f(s)$ with a unit steady state gain and a sufficiently broad bandwidth. In practice, it can be taken as the low-pass filter

$$G_f(s)=\frac{1}{T_f s+1} \tag{26}$$

with $T_f=1/\omega_f$ and ω_f the higher limit of the frequency range of the system dynamics and the external disturbance (in a more general framework that includes measurement noise, also the noise spectrum can be considered here).

The UDE only uses the functions of the state and the control signal to observe the uncertainties and the unknown disturbance. Hence,

$$\begin{aligned}u&=B^+\left[A_m x+B_m c-Ke-Ax\right]+ude\\ &=B^+\left[A_m x+B_m c-Ke-Ax\right]+B^+\left[-\dot{x}+Ax+Bu\right]*g_f.\end{aligned} \tag{27}$$

Using the Laplace transform to solve (27) for $u(t)$, the UDE control law is then,

$$\begin{aligned}u(t)=\;&B^+\mathcal{L}^{-1}\left(I-B^+BG_f\right)^{-1}*\left[A_m x+B_m c-Ke-Ax\right]-\\ &B^+\mathcal{L}^{-1}\left\{sG_f\left(I-B^+BG_f\right)^{-1}\right\}*x+\\ &B^+\mathcal{L}^{-1}\left\{G_f\left(I-B^+BG_f\right)^{-1}\right\}*Ax\end{aligned} \tag{28}$$

where $\mathcal{L}\{\cdot\}$ is the Laplace transform operator. Or, if the dimension of $u(t)$ is one (as with the considered case, and with a little modification), the Laplace form of the control input could be,

$$\begin{aligned}U(s)=\frac{1}{1-G_f(s)}&B^+\left[A_m\,X(s)+B_mR(s)-KE(s)\right.\\ &\left.-sX(s)G_f(s)\right]-B^+AX.\end{aligned} \tag{29}$$

According to the dynamic model of the regenerative chatter vibration in orthogonal cutting, (16), and the UDE controller, (29) is given here as,

$$\begin{aligned}U(s)=\frac{1}{1-G_f(s)}&m_0\left[\left(\omega_m^2-K_1\right)Y(s)+K_2SY_m(s)\right.\\ &-\left(\zeta_m\omega_m+K_2\right)\dot{Y}(s)+\omega_m^2R(s)+K_1Y_m(s)\\ &\left.-SG_f(s)\dot{Y}(s)\right]+m_0\left(k_0+k_{fo}\right)Y(s)-m_0c_0\dot{Y}(s)\end{aligned} \tag{30}$$

IV. SIMULATION RESULTS

The suggested UDE control law for regenerative chatter vibration in orthogonal cutting is demonstrated in this subsection. The simulation considers the typical turning process of a tailstock-supported workpiece. The workpiece is made of aluminum 1020, and its physical parameters are given as follows. The elastic modulus E is $7\times10^{10}\,Pa$, the density ρ is $2770\,Kg/m^3$, the damping ratio ξ is 0.029, the length L is $0.3m$ and the diameter d is $8mm$. As to the coefficients of the turning process, the nominal cutting constant K_{f0} is $1800MPa$ with an uncertain part chosen as $\vartriangle K_f = 50\% K_{f0}$, the spindle speed is 1500rpm, the width of cut Ψ is taken as 5mm and the target chip thickness is $h_0 = 1.5mm$. The natural frequency at the first vibration mode (in (5)) is 273.5Hz. The SLD plots with the nominal coefficients are shown in Fig. 4. The star sign (i.e., *) on Fig. 4 represents unstable cutting conditions.

The reference model (18) is chosen to be stable with $\omega_{mn} = 300\,rad/\sec$ and $\zeta_m = 0.05$, hence the error feedback gain K can be chosen as zero. The low-pass filter in (26) has been designed with $T_f = 1/f_{n1} = 1/273.5$. The unknown external disturbance ω is simulated as a zero mean white-noise signal, with noise power of 0.01 and sample time of 0.005 sec.

The UDE control low for regenerative chatter vibration in orthogonal cutting was simulated by Simulink™. The simulated closed loop block diagram is shown in Fig. 3, and the results refer to two cases, with the following details.

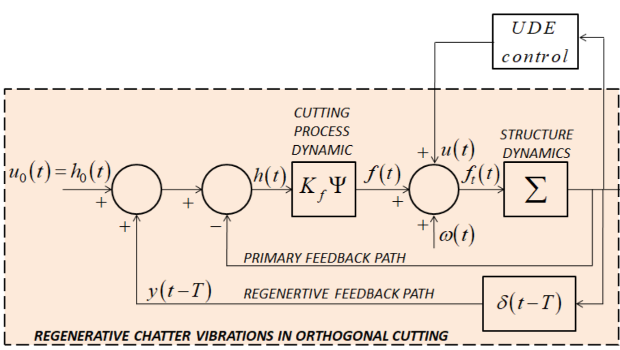

Figure 3. Block diagram of model and ude control of regenerative chatter in orthogonal cutting.

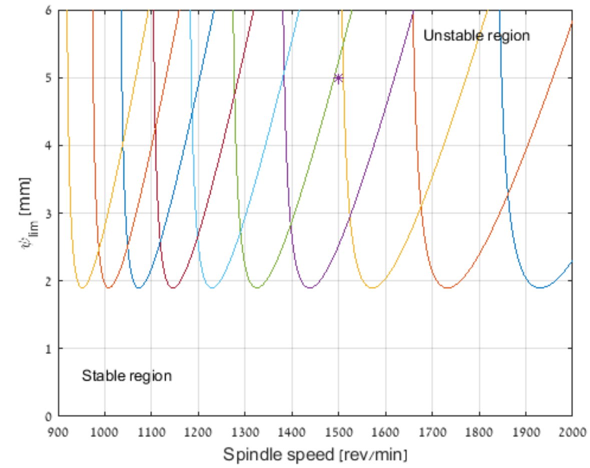

Figure 4. Stability Lode Diagrams for K_{f0} and ξ=0.029

Case 1, for an unstable process and known model coefficients is shown in Fig. 5. Then, case 2, shown in Fig. 6, describes an unstable process with nominal cutting position at *0.5L* and a few levels of uncertainty. Fig. 5 includes three stages. Stage I, with no closed loop control (i.e., the controller is off), is from initial time to 2 [s]. Stage II is from 2 [s] until 4 [s], with closed loop control but without external disturbance. Finally, stage III that takes place from 4 [s] to the end of the simulation, includes the control action with the influence of white-noise external disturbance.

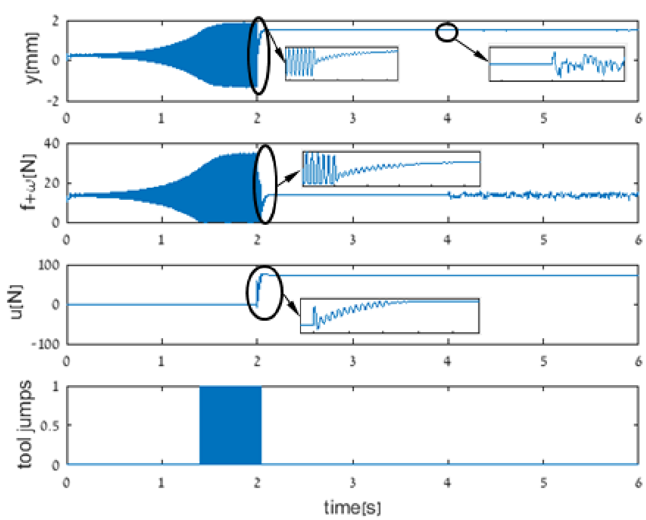

Figure 5. UDE Control for unstable process

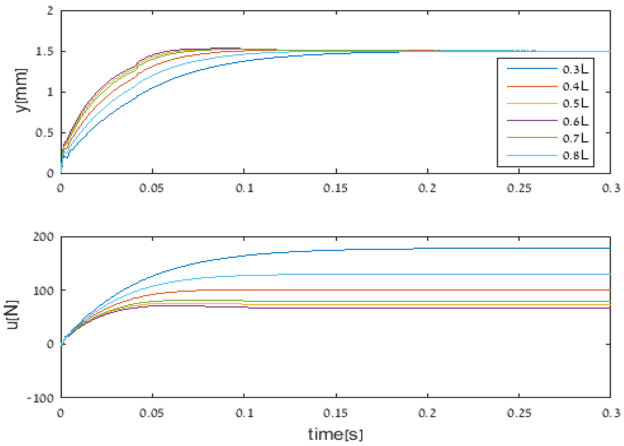

Figure 6. UDE Control for unstable process and uncertainty

The upper plot in Fig. 5 shows the relative displacement between the tool and the workpiece, normal to the machined surface. The second plot presents the chatter force with the external disturbances. The control signal is shown in the third, and the plot at the bottom of the figure indicates the time of tool jumps (out of the cut). It shows clearly that as a result of the stability loss, the tool jumps out, causing loss of cutting. Fig. 6 shows the performance of the UDE control for unknown changes in the cut-off position from the nominal work point (i.e., $x = 0.5L$), indicating the ability of UDE to compensate for uncertainty and its robustness. As mentioned earlier, changing the location of the cut causes changes to the values of equivalent stiffness (4) and mass (6). These changes are assumed uncertain in this study.

CONCLUSIONS

This paper has presented a new control strategy for the problem of self-excited vibrations in orthogonal cutting machines. The standard orthogonal cutting model was extended with dynamic coefficients that depend on the position of the cut. In addition, this dynamic model includes uncertainty and disturbances; conditions that are typical for actual machines. The control law was developed based on principles of the uncertainty and disturbance estimator (UDE) control. The achieved numerical results describe the performance of the suggested controller at two cases. One is of unstable process conditions with unknown external disturbance, and the other is when the process is open-loop unstable and with uncertain model coefficients. Both presented cases have shown significant reduction of chatter, and demonstrate the potential of UDE approach to active vibration control in turning processes.

REFERENCES

[1] G. Quintana, J. Ciurana, “Chatter in machining process: A review,” International Journal of Machine Tools & Manufacture, vol. 51, pp. 363–376, January, 2011.

[2] R. S. Hahn, "On the theory of regenerative chatter in precision grinding Operations", Trans. A.S.M.E., 76 (1954) 593-597.

[3] J. Trustly and M. Polacek, "The stability of machine tools against self-excited vibrations in machining", International Research in Production Engineering, ASME, New York (1963) 465- 474.

[4] S. A. Tobias, "Machine tool vibration", BLACKIE, London (1965).

[5] M. Shiraishi, K. Yamanaka, H. Fujita, "Optimal control of chatter in turning", Int. J. Tools Manufact, vol. 31, No. 1, pp. 31-43, 1991.

[6] A. S. White, "Simulation of active control of chatter vibrations", International Journal of recent development in engineering and technology, vol. 3, issue 4, October 2014.

[7] F. Chen, M. Hanifizadegan, "Active damping of boring bar vibration with a magnetic actuator", IEEE/ASME TRANSACTION ON MECHATRONICS, vol. 20, no. 6, December 2015.

[8] Q. Zhong, D. Rees, "Control of uncertain LTI systems based on uncertainty and disturbance estimator", Journal of dynamic system, measurement and control, ASME, vol. 126, pp. 905-910, 2004.

[9] A. Kuperman, Q. Zhong, "Robust UDE based control of uncertain nonlinear state delay system", IEEE eurocon, 2009.

[10] A. Patel, R. Neelgund, A. Kolhe, M. Kuber, S. Talor, "Robust control of flexible joint robot manipulator", Proceedings of the IEEE International Conference on Industrial Technology (ICIT 2006), Mumbai, 2007, pp. 649-653.

[11] S. Talole, S. Phadke, "Robust input-output linearization using UDE", Proceedings of the International Conference on Advances in Control and Optimization of dynamical System (ACODS2007), Bangalore, India, 2007, pp. 79-82.

[12] S.Talole, S.Phadke, "Robust input-output linearization using uncertainty and disturbance estimation", International Journal of Control, 2010, vol. 82(10), pp. 1974-1803.

[13] Q. C. Zhong, A. Kuperman, R. K. Stobart, "Design of UDE-based controllers from their two-degree-of-freedom nature", International Journal of Robust and Nonlinear Control, 2011, vol. 21, pp. 1994-2008.

[14] P. Shendge, B. Patre, "Robust model following load frequency sliding mode controller based on UDE and error improvement with higher order filter", IAENG International Journal of Applied Mathematics, 2007, vol. 37(1).

[15] T. Chandrasekhar, L. Dewan, "Sliding mode control based on TDC and UDE", International Journal of Informatics and systems sciences, 2007, vol. 3(1), pp. 36-53.

[16] S.Talole, S.Phadke, "Model following sliding mode control based on uncertainty and disturbance estimation", Journal of dynamic systems, measurement, and control", Transaction of the ASME 2008, 130.

[17] A. Kuperman, Q. C. Zhong, "Control of uncertain linear systems with a state delay based on an uncertainty and disturbance estimator", Proceedings of the Sixth IFAC Symposium on Robust Control Design, Haifa, Israel, 2009.

[18] R. K. Stobart, A. Kuperman, Q. C. Zhong, "Uncertainty and disturbance estimator (UDE)-based control for uncertain LTI-SISO systems with state delays, Journal of dynamic systems measurement and control – Transactions of the ASME 2011, 133.

[19] A. Kuperman, Q. C. Zhong, "Robust control of uncertain nonlinear system with state delays based on an uncertainty and disturbance estimator", International Journal of Robust and Nonlinear Control 2010.

[20] Allan G. Piersol, Thomas L Paez, Harris' Shock and Vibration Handbook, McGraw Hill Professional, 2009.

[21] Y. Altintas, Manufacturing Automation: Metal Cutting Mechanics, Machine Tool Vibrations, and CNC Design, Cambridge University Press, 2012.

[22] E. Turkes, S. Orak, S. Neseli, S. Yaldiz, " Linear analysis of chatter vibration and stability for orthogonal cutting in turning", Int. Journal of Refractory Metals and Hard Materials 29 (2011) 163–169.

[23] L.N. Lopez de Lacalle M.E. Guitierrez G. Urbikain, A. Fernandez. Stability lobes for

[24] general turning operations with slender tools in the tangential direction. International

[25] L.N. Lopez de Lacalle M.E. Guitierrez G. Urbikain, A. Fernandez. "Stability lobes for general turning operations with slender tools in the tangential direction". International, Journal of Machine Tools and Manufacture, 67:35–44, Apr 2013.