Limiting Factors for Active Suppression of Structural Chatter Vibrations Using Machine′s Drives

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Abstract: Chatter vibrations are a major limitation for rough milling operations, leading to productivity reduction, low tool life and poor surface finish. It has been shown recently that the machine tool′s own drives can be used to increase the stability limit related to structural modes of the machine. To damp the low frequency modes, a new feedback loop is added to the classical position, velocity and current cascaded control. The objective of this study is to analyse the limitations of this new chatter suppression technique. Constraints related to the non-collocated control are first studied on a simplified three-mass model and then experimentally demonstrated on a three-axis horizontal milling centre. The industrial integration of the new control loop with sampling time constraints and limited drive′s bandwidth is analysed. After determining the appropriate conditions to use this chatter suppression technique, a cutting test demonstrates that the stability limit can be doubled in the low regions of the stability lobes.

Key words: chatter; control; damping; milling

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0 Introduction

In heavy duty milling operations, machine tool users are pursuing high material removal rates, which are mainly limited by two factors: the spindle power and the chatter vibrations. However, chatter vibrations can appear with really small depth of cut reducing drastically the machine′s productivity; hence, it is an important field of investigation for the research community. In general, chatter can come from the flexibility of any element in the force flow: tool, spindle, machine structure, workpiece, fixtures, etc. However, in heavy duty machining, chatter is mostly related to vibration modes of the whole machine structure. This work focuses on those low frequency chatter problems occurring at frequencies between 15 Hz and 200 Hz.

Dampers can be added to the machine tool structure to increase the chatter stability limit. A tuned mass passive damper can damp a specific frequency. Unfortunately, the available space is really limited and the frequency of the structural modes are changing with the position of the machine so passive dampers are not able to satisfy the requirements. Active dampers can handle dynamics variations and have been successfully used for chatter suppression [1-2]. However, these solutions require the installation of new actuators on the machines.

Instead of using additional actuators, the machine′s own drives can be used to increase the damping. The use of machine′s drives to suppress chatter vibrations has been simulated first by Chen and Tlusty [3]. Recently, the increase of the chatter stability related to the low frequency structural modes has been demonstrated on a large milling machine by Munoa et al[4].

The classical cascaded control with current, velocity and position loops can affect the structural machine′s dynamics. Especially the proportion-

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al gain of the velocity loop can be tuned to provide the optimum damping [5-6]. However, the addition of damping brought by the velocity loop is moderate compared to what can be achieved by the use of an additional feedback loop.

Additional feedback loops have been used in the literature to reduce inertial vibrations coming from the machine movements. Szabat and Orlowska-Kowalska presented a comparative study of the different feedback possibilities [7]. Dietmair and Verl showed a drive based vibration damping with an additional feedback in the velocity setpoint[8]. Zatarain et al. placed an accelerometer as close as possible to the tool and used a state space observer to improve the dynamic behaviour of a machine [9]. Zirn and Jaeger also used a state space control to feed back the load acceleration in the acceleration setpoint[10]. Hosseinabadi and Altintas damped a ballscrew mode using the linear encoder signal[11]. A comprehensive review of active damping using machine's drives is presented by Altintas et al. in Ref. [12]. However, the research in this field is limited by the difficulty to modify the control loop structure on industrial computer numerical control (CNC), thus open architecture controllers are required to implement new CNC algorithms[13].

CNC manufacturers are also introducing active damping options in their controllers. Fanuc proposes the vibration damping control function that feeds back the difference between the speeds of the motor encoder and of the linear encoder[14]. This function is particularly suited to damp the mode originating from the elasticity of the drive's transmission. Siemens is also proposing a similar solution with the advanced position control (APC)[15] which is analysed by Zirn in Ref. [6]. A patent owned by Mitsubishi Electric Corporation presents the addition of load feedback to the classical cascade loops[16]. Heidenhain offers the active vibration damping (AVD) option to reduce machine's structure oscillations. Moreover, a dedicated chatter suppression option called active chatter control (ACC) is available[17-19]. Spindle drive is also used to suppress chatter vibrations[20]. The spindle speed variation technique is implemented for example by Okuma in the Machining Navi options[21]. This non-exhaustive industrial review proves that the major CNC manufacturers are working to act against vibrations directly with the machine's drives.

Up to now, successful implementations of additional feedback loops have been presented but no information is given about the limitations related to this control strategy. Moreover, specific questions are raised when the drives are used to increase the chatter stability.

The objective of this paper is to study the limitations concerning the use of the machine's drives to increase the chatter stability limit. The first physical restriction analysed here is related to the destabilization of counter-phase modes that limit the possible dynamic behaviour improvement. Then, technical restrictions, like the integration of additional feedback in an industrial CNC and the related sampling time limitations, as well as the dynamical properties of the drives and their limited bandwidth, are taken into account. Finally, the positive and negative effects of the additional feedback loop on the stability are described, and the ability of the drive to suppress chatter vibrations is demonstrated.

First, a simplified model allows the explanation of the problems related to non-collocated control that can destabilize counter-phase modes. Then, the technical requirements related to the CNC integration, the limited sampling time and the machine's drives bandwidth are analysed. Finally, the effect of the additional feedback on chatter stability allows the determination of the appropriate conditions for the use of this new chatter suppression technique. Cutting tests on a three-axis milling machine demonstrate the ability of the drive to suppress chatter vibrations.

1 Limitation Related to Modes' Destabilization

Different load feedbacks are analysed here on a two-mass system presenting a single mode.

However, the main limiting factor which prevents improving further the dynamical response of the system is related to the mode destabilization. Fig. 1 presents the block diagram of a drive with a three-mass system having two vibration modes. The typical control structure with current, velocity and position cascaded loops is used. Due to its high bandwidth, the current loop can be neglected and is modelled here as a perfect amplification. The integral term of the velocity loop controller is not considered here for sake of simplification. Among the different additional feedback strategies presented in the literatures, two acceleration feedback loops are studied here.

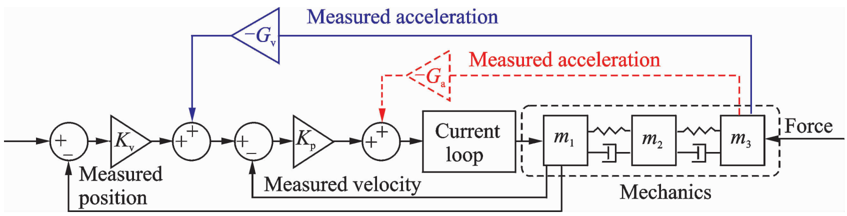


Fig. 1 Block diagram of machine's drive with additional feedbacks

The acceleration measured in the third mass can be fed back to the velocity setpoint with a gain G_v or to the acceleration setpoint with a gain G_a. As the acceleration measurement point and force application point are different, limitations related to non-collocated systems are faced [22].

As shown in the root locus of the velocity loop (Fig. 2(a)), the acceleration feedback in the velocity setpoint is affecting the position of the system zeros whereas the acceleration feedback in the acceleration setpoint is modifying the position of the poles of the system. If only one mode is considered, there is no limit to increase the damping of the system. However, real systems are presenting many modes. With the non-collocated control, some modes can be damped, whereas others are destabilized.

The proportional gain of the velocity loop denoted K_p is positioning the poles of the system on the root locus. If the gain K_p is high, the pole will be close to the zero of the root locus, hence the acceleration feedback in the velocity setpoint has a direct effect on the system pole, whereas the acceleration feedback on the acceleration setpoint has almost no effect. However, playing with the value of the gain K_p, both acceleration feedback strategies can have similar performances.

Fig. 2(b) presents the effect of the acceleration feedback on the compliance of the third mass. The targeted mode is damped but the second mode is destabilized. This example schematically presents the limitation related to the non-collocated control inducing the destabilization of counter-phase modes. Experimental measurements showing this effect are presented further.

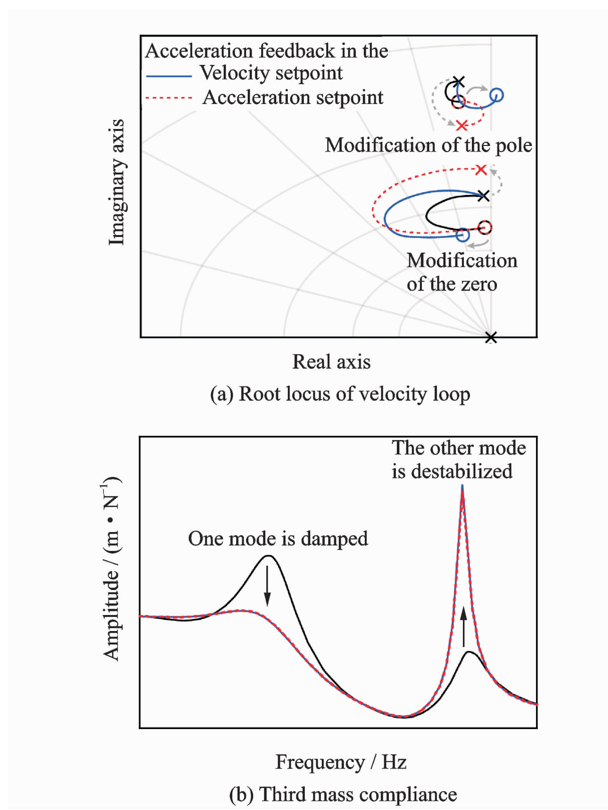


Fig. 2 Effect of additional feedbacks on three-mass model

To counter the effect of mode destabilization, a loop shaping controller has been used in Ref. [4]. Indeed, by selectively adapting the gain and phase for the frequencies of interest, it is possible to avoid the destabilization of the counter-phase modes. However, when the modes are close to each other and considering the dynamical variations due to the machine posture modification, the mode destabilization remains the main element limiting the performance of the active damping strategy when machine's own drives are used.

Another way to avoid destabilizing one of the mode is to feedback the difference of the velocities between m_2 and m_3. This kind of strategy is well adapted for the modes which are in the transmission chain and for which the motor encoder and the linear scale can directly measure both velocities. For structural chatter, the use of this technique results in more difficulties due to the complex mode shape and to the obstacles in measuring the required signals.

Both feedbacks in the velocity and acceleration setpoints can improve the dynamical behaviour of the machine and a deeper study should be carried out to compare them. However, for the rest of the paper, the acceleration feedback in the velocity setpoint is preferred to be able to keep a high value of the velocity loop proportional gain.

A vertical ram-type machining centre is introduced to illustrate the rest of this article, further details about the dynamics, stability lobes and experimental chatter limits of this machine can be found in Ref. [23]. Fig. 3(a) presents the shape of the main mode which has to be damped and which is mainly related to the bending of the ram. The dynamics of this machine is complex and cannot be accurately modelled by the three-mass model presented before but this simple model allows the understanding of the basic behaviour of system. Indeed, in Fig. 3(b) the amplitude of the main peak at 60 Hz is decreased as the gain is increased. However, the modes around 45 Hz and 75 Hz are destabilized due to the counter-phase mode behaviour.

2 Limitation Related to Technical Implementation

In previous work, an Open CNC has been implemented to avoid most limitations related to

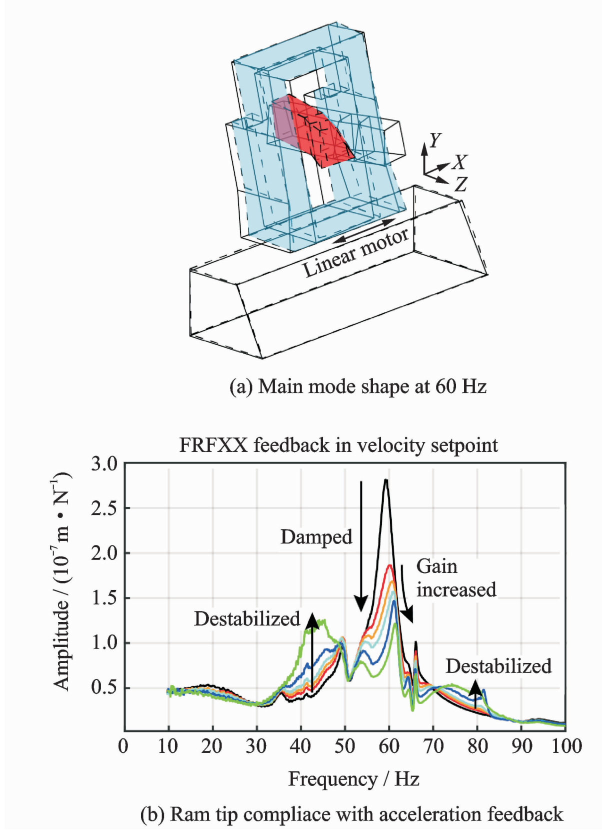


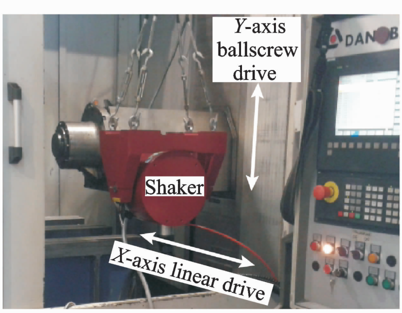
Fig. 3 Main mode shape and ram tip compliance with acceleration feedback at 60 Hz

industrial CNCs[4]. Hence the control structure could be modified easily and the sampling time was low enough to avert a significant phase lag. When using industrial CNCs, the additional feedback loop can be implemented in the programmable logic controller (PLC) program or in the CNC kernel, but in all cases the achievable sampling time is limited. This limitation induces a phase lag and can hamper the possibility of successfully controlling the system.

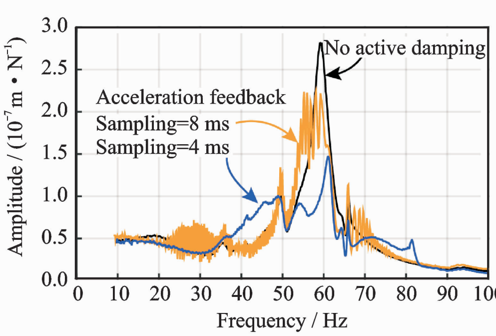
The integration of a new feedback loop in an industrial CNC is not a straightforward task and can be impossible for some CNC's. Machine tool builders have access to the PLC but the scanning time is usually high (1 to 20 ms). Access to the real time part of the CNC kernel is restricted by most CNC manufacturers. In the Siemens 840D CNC, it is possible to introduce additional feedback loops using the Compile-Cycle programming options. Indeed, the Compile-Cycles are software components that are developed by the end-user and are added to the standard system using the open system architecture of the NC kernel [24]. A fast analogue input module is added to the communication bus of the CNC to read the accelerometer signals. This signal is processed internally by the CNC to apply filters and is added to the velocity or acceleration commands. The available sampling time depends on the capacity of the processor of the CNC. The machine used in this paper has a relatively slow CNC processor hence the smallest sampling time which can be used to read the accelerometer signal is 4 ms.

In a first approximation, the sampling effect can be approximated by a first-order filter to take the introduced phase lag into account. This phase lag can be compensated by adjusting the filters of the acceleration feedback controller. However, when the sampling time is too high, the missing information makes impossible to control the system. Experimental tests carried out on a milling machine show that with a sampling time of 8 ms, the compliance Frequency Response Function becomes noisy and the system cannot be controlled properly (Fig. 4).

The tests revealed that even with a relatively high sampling time (4 ms), the additional feed-



(a) Three-axis horizontal milling centre with shaker to measure FRF



(b) Effect of the sampling time on FRF

Fig. 4 Three-axis horizontal milling centre and effect of sampling time on FRF

back loop can damp structural machine tool modes. This is important because it means that the low sampling tasks of CNC can be used to implement the acceleration feedback strategy. Indeed, experiments show that the 60 Hz mode can be damped effectively even if the sampling time reaches 6 ms (166 Hz).

In the CNC control literatures, the problem related to sampling time is recognized and experimental conditions are typically chosen to ensure that the sampling period has minimal negative impact on loop stability. Sometimes, additional measures are also used to mitigate errors due to sensor noise as well; such as Kalman filters[10], state space observers[7] or specific sensors like Ferrari sensors. With the full implementation in the Compile-Cycle, the grounding of the machine is naturally acting against the electrical noise. In our experimental work, standard industrial accelerometers are used and no specific signal processing is required.

3 Limitation Related to Drive′s Bandwidth

Apart from the non-collocated control, another problem in the application of machine′s drives is their limited bandwidth. Indeed, the drives are not designed for this purpose and the tuning of the velocity and position loop is constrained by the machines′ modes[4-5]. A main challenge is dealing with vibration modes which are outside the response frequency range (i. e., bandwidth) of the feed drives. Fig. 5 shows the bode diagrams of the position and velocity loop of the X axis linear drive. The Bode diagram of the velocity loop shows that the machine has several modes related to the structure of the machine (column, ram, etc.) at low frequency (<80 Hz) and many other modes at higher frequencies are related to smaller components. The mode at 170 Hz is specifically dangerous because it can become unstable if the proportional gain of the velocity loop is increased. The additional acceleration feedback can also destabilize this mode but as this

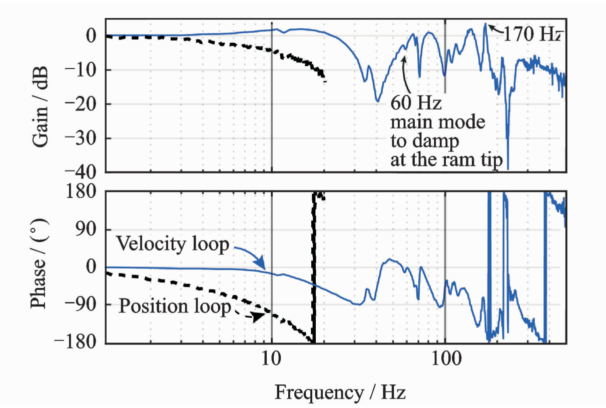


Fig. 5 Bode plot of X-axis linear drive

mode is far from the targeted 60 Hz mode, a loop shaping controller design can avoid exciting this mode.

In Ref. [4], rack and pinion and ballscrew drives were successfully used to damp structural modes of a large milling machine. The limited bandwidth of the drives is preventing using this technique to counter modes which are not coming from the machine structure, like tool or spindle modes which usually have frequencies higher than 300 Hz. However, the drives seems able to damp structural mode and the main problem related to the dynamics of the drive is also the machines' modes destabilization due to the actuator phase lag.

4 Limitation Related to Effect on Stability Lobe

In Fig. 3(b), it is possible to reduce the maximum amplitude of the Frequency Response Function. However, for chatter suppression applications, the effect on the stability lobes diagram should be considered. The non-collocated acceleration feedback strategy can induce the amplification of the counter-phase mode, which can thus limit the cutting capability in some regions of the stability diagram. The effect of the acceleration feedback on the velocity setpoint is shown in the stability lobes presented in Fig. 6. By lowering the amplitude of the maximum peak of the Frequency Response Function, the stability limit is increased for the lower portions of the lobes. On the other hand, the stability limit is reduced for the higher portions of the lobes and the sweet spots may disappear.

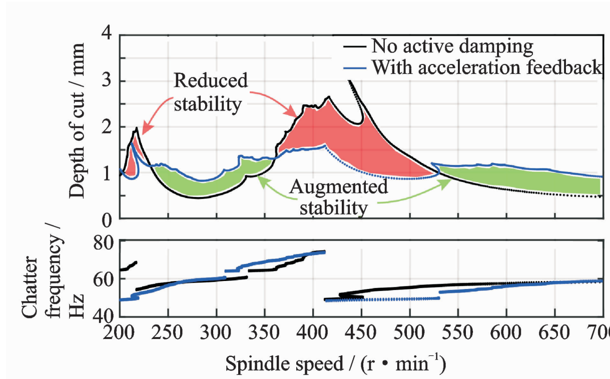


Fig. 6 Zero order stability lobe approximation[25] without and with acceleration feedback in the velocity setpoint

Depending on the spindle speed, the acceleration feedback can improve or worsen the stability. This is an important drawback of this chatter suppression technique using machine's drive. Hence, the additional feedback loop should be used carefully.

5 Experimental Demonstration of Chatter Suppression

The worst position on the stability lobe has been selected to demonstrate the ability of the drive to suppress chatter vibrations. A tool of diameter 125 mm with 8 teeth is used with a spindle speed of $N=300$ r/min and a feed per tooth $f_z=0.2$ mm/z. The radial depth of cut is 100 mm which represents 80% of the tool diameter, the axial depth of cut is 1.5 mm. Fig. 7 presents the measured acceleration signal in time and frequency domains. Strong chatter appears at 61 Hz related to the bending mode of the ram in the X direction. When the acceleration feedback is activated, it is possible to see that the amplitude of vibration is immediately reduced and the frequency spectrum shows that the cutting process is stable. The stability limit is predicted considerably accurately when no active damping is used, however, with the drive's acceleration feedback activated only the main trends are reliable in the stability

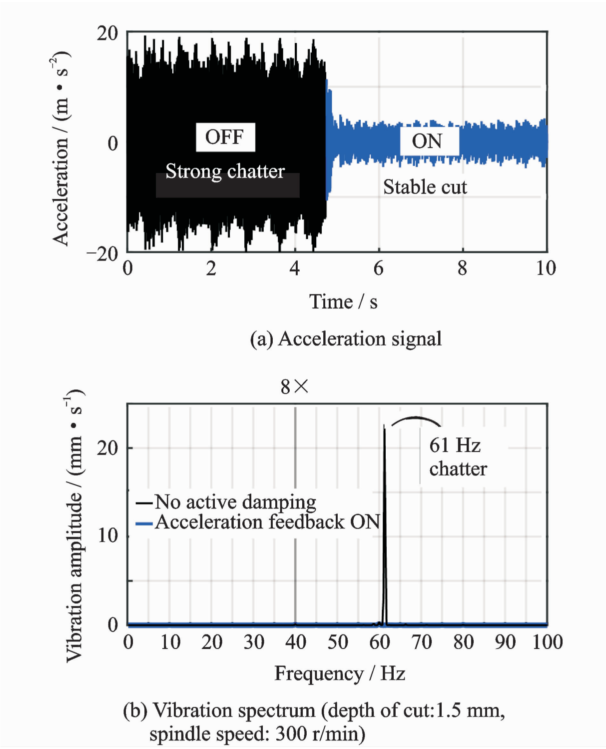


Fig. 7 Effect of activation of acceleration feedback

lobes presented in Fig. 6. For the selected cutting conditions, the stability limit can be doubled, which is a huge improvement at a relatively low cost.

6 Conclusions

Here, we investigates the use of the machine's own drives to damp the structural vibration mode of a machine tool and hence increase the chatter stability. The main contribution of this work is the analysis of the limiting factors related to the new chatter suppression technique. It has been addressed that the most problematic issue is the destabilization of other modes of the machine. A specific effort to design the feedback controller can improve the behaviour but it is still the most critical element. It has been validated experimentally that the integration in a Siemens 840D CNC is possible and that the sampling time requirements are not strong. Indeed, the 4 ms sampling time is enough to control the system and a 61 Hz chatter could be removed. Another possible limiting factor is the machine drive bandwidth. However, rack and pinion, ball screw and linear drives have been successfully used to damp structural machine tool modes. Finally, the stability lobe diagram and cutting tests show that this additional feedback can significantly improve the depth of cut for the minimum regions of the lobes. However, the additional feedback can worsen the stability for other areas of the stability lobes. To conclude, this work provides a clearer view on the limiting elements that should be taken into account when the machine's drives are used to suppress chatter vibrations.

References:

[1] MUNOA J, MANCISIDOR I, LOIX N, et al. Chatter suppression in ram type travelling column milling machines using a biaxial inertial actuator[J]. CIRP Annals—Manufacturing Technology, 2013, 62: 407-410.

[2] BRECHER C, SCHULZ A, WECK M. Electro-hydraulic active damping system[J]. CIRP Annals—Manufacturing Technology, 2005, 54:389-392.

[3] CHEN Y, TLUSTY J. Effect of low-friction guideways and lead-screw flexibility on dynamics of high-speed machines [J]. CIRP Annals—Manufacturing Technology, 1995, 44:353-356.

[4] MUNOA J, BEUDAERT X, ERKORKMAZ K, et al. Active suppression of structural chatter vibrations using machine drives and accelerometers[J]. CIRP Annals-Manufacturing Technology, 2015, 64:385-388.

[5] GROβ H, HARMANN J, WIEGÄRTNER G. Electrical feed drives in automation [M]. SIEMENS: MCD corporate Publishing, 2001.

[6] ZIRN O. Machine tool analysis-modelling, simulation and control of machine tool manipulators [D]. [S. l.]: Department of Mechanical & Process Engineering ETH Zürich, 2008.

[7] SZABAT K, ORLOWSKA-KOWALSKA T. Vibration suppression in a two-mass drive system using PI speed controller and additional feedbacks—Comparative study[J]. IEEE Transactions on Industrial Electronics, 2007, 54:1193-1206.

[8] DIETMAIR A, VERL A. Drive based vibration reduction for production machines [J]. Modern Machinery Science Journal, 2009, 3:129-133.

[9] ZATARAIN M, DE ARGANDONA I R, ILLARRAMENDI A, et al. New control techniques based on state space observers for improving the precision and

dynamic behaviour of machine tools[J]. CIRP Annals—Manufacturing Technology,2005,54:393-396.

[10] ZIRN O, JAEGER C. Vibration damping for machine tool servo drives by load acceleration feedback [J]. IEEE International Symposium on Industrial Electronics (ISIE),2010(3): 3264-3269.

[11] HOSSEINABADI A H, ALTINTAS Y. Modeling and active damping of structural vibrations in machine tools[J]. CIRP Journal of Manufacturing Science and Technology, 2014,7:246-257.

[12] ALTINTAS Y, VERL A, BRECHER C, et al. Machine tool feed drives[J]. CIRP Annals—Manufacturing Technology,2011, 60:779-796.

[13] BEUDAERT X, LAVERNHE S, TOURNIER C. Direct trajectory interpolation on the surface using an open CNC[J]. The International Journal of Advanced Manufacturing Technology, 2014, 75:535-546.

[14] FANUC. Fanuc30i Alpha-i Parameter manual B-65270EN/07[M]. Japan: Fanuc Company,2010.

[15] SIEMENS. Function manual drive functions—FBA [M]. [S. l.]:Siemens,2009.

[16] MARUSHITA Y, IDEKA H, MATSUMOTO K. Electric motor control apparatus: Mitsubishi, U. S. Patent 7868577B2[P]. 2011-05-21.

[17] HEIDENHAIN. Technical manual TNC 640[M]. [S. l.]:Heidenhain, 2015.

[18] KERNER N. Closed-loop control structure for damping low-frequency vibrations: US20130204404 A1 [P]. 2013-03-11.

[19] KERNER N. Parametrization procedures attenuation of chatter vibrations in the control loop of a numerical control: DE 102013202408 A1[P]. 2014-03-11.

[20] BEDIAGA I, EGAÑA I, MUNOA J. Reducción de la inestabilidad en cortes interrumpidos en fresado a alta velocidad mediante variación de la velocidad del husillo[C]// XVI Congreso de Máquinas-Herramienta y Tecnologías de Fabricación. San Sebastián, Spain:[s. n.],2006.

[21] NISHIMURA K, YOSHINO K, ISHII H, et al. Machine tool, Nagoya University—Okuma Corporation: US20120109360A1[P]. 2012.

[22] PREUMONT A. Vibration control of active structures: An introduction[M]. [S. l.]: Kluwer Academic Publishers, 2002.

[23] IGLESIAS A, MUNOA J, CIURANA J. Optimisation of face milling operations with structural chatter using a stability model based process planning methodology[J]. The International Journal of Advanced Manufacturing Technology,2014,70:559-571.

[24] Sinumerik 840D. OEM-package NCK, User's Manual[M]. Edition 11/00,2005.

[25] ALTINTAS Y, BUDAK E. Analytical prediction of stability lobes in milling[J]. CIRP Annals—Manufacturing Technology,1995,44:357-362.

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