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ORIGINAL PAPER

Adaptive Model-Free Gain Tuning for Active Damping of Machine Tool Vibrations

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Abstract

Purpose Chatter can damage parts. It must hence be avoided and/or suppressed. This paper discusses the use of active vibration control to suppress chatter in a milling process. To apply only as much force as is necessary to stabilize the process, a novel adaptive and model-free gain tuning method is proposed in which gains are adapted to the level of unstable vibrations detected during machining.

Methods Vibrations during the cutting process are monitored using an accelerometer. If/when instabilities are detected, an active damper that is mounted on a flexure is supplied an appropriate control signal based on a velocity feedback control law. The actuator then applies a suitable compensatory force on the flexure to damp the vibrations. Since the amount of force to be applied is governed by the actuator type and by the level of instability detected, efficacy of proposed adaptive gain tuning scheme is tested for its dependence on the time required to update the gain and for its dependence on the levels of gain increments.

Results For slot milling of steel, active damping of unstable vibrations is shown to stabilize the process and improve productivity by up to ~ 300%. With the adaptive gain tuning scheme, higher gain increments with shorter updating times are observed to result in the process being stabilized quicker.

Conclusions Since the proposed scheme is model-free and much simpler to implement than other previous adaptive gain updating schemes found in the literature, and efficacy of the scheme is demonstrated experimentally, it has great potential for industrial use.

Keywords Milling · Chatter · Adaptive controller · Active damping

Introduction

Chatter vibrations degrade machined part surface quality and have the potential to damage elements of the machine. Chatter must hence be avoided. In general, the occurrence of chatter depends on the dynamics of the machine tool system reflected at the cutting point, the tool geometry, the material being cut, and the cutting parameters. Mitigating chatter depends on what causes it [1, 2]. If machine tool structural vibrations are responsible for it, chatter can be avoided by cutting at parameters that lie below model-predicted stability boundaries [3], or by varying and/or selecting spindle speeds such as to avoid regenerative chatter vibrations to develop [4], or using tools with special geometries that disturb the regeneration mechanism [5], or by dynamically stiffening the machine through structural modifications [6] and/or by controlling vibrations by passive [7] and/or active means [8, 9].

Selection of cutting parameters guided by model-predicted stability charts does not always result in chatter being avoided, since predictions are governed by uncertainties in the measured dynamics, by potential nonlinearities in the cutting process, and by approximations in the model. Chatter avoidance by varying spindle speeds may not always be feasible in practice, since variation of spindle speeds requires additional power capacity in the spindle over and above that required for cutting. Changing tool geometry may also not always be possible, since the process may require that only certain types of tools be used. Modifying the machine tool

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elements to dynamically stiffen them is possible only at the design stage and is not suitable for tools and machines already in use. Methods to passively damp structural vibrations to avoid chatter through integration of absorbers, though effective, require new and separate solutions for all machines. In this context, since active damping of vibrations offers a viable solution for avoidance of chatter in machines already in use, this paper focuses its attention only on active vibration control. The modest aim is to propose and demonstrate new methods to detect and actively suppress chatter vibrations.

Active control of vibrations in machine tools has received much attention. Solutions include those focused on active structural chatter suppression, active feed drives, active cutting tools and tool holders, active spindle systems, and active workpiece holders. A succinct review of these is provided in [1, 2]. Though there exist different solutions for active vibration control for machine tools in the literature, the present work is concerned only with suppressing vibrations of machine tool structural components, i.e., the case in which the stiffness of the cutting tool, spindle, and workpiece system is higher than the structural mode of the machine [9–12]. And, as such, we situate our proposed contribution only within the context of other such solutions in the literature that are focused on structural vibration control.

The efficacy of active vibration control to avoid chatter depends on the type of sensor monitoring the cutting process, on the type of actuator that applies a suitable compensatory force, on observability and controllability of vibration modes to be damped, and on the type of control law. Usually, accelerometers are found to be adequate for monitoring the cutting process [8–10, 13], and hence, it is the preferred transducer in the studies reported in this paper. Of the different actuator types, though piezoelectric [11, 14–16] and electro-hydraulic actuators [17, 18] have been shown to be effective in suppressing machine tool vibrations, this paper prefers to use an actuator of the electromagnetic type. Electromagnetic actuators are usually compact with sufficiently high force-to-weight ratios and desirable force-frequency characteristics, and hence have been preferred by other researchers too for active damping of machine tool vibrations [8–10, 12, 13, 19]. With regards to where to place the sensor and the actuator, though both are recommended to be collocated [20], that may not always be possible in machines where it is desirable and easier to monitor the process close to the cutting zone and where actuators are to be placed on structural components far removed from the cutting zone [8–10, 19]. Since non-collocated control is the more generalized case, this paper too discusses active control for a non-collocated system.

In addition to correctly resolving issues related to the sensor and actuator type and where to place both, the performance of active damping solutions is largely governed by the feedback control strategy. Control laws can either be model-based or model-free. Abele et al. [16] used a model-based robust microcontroller to actively damp vibrations in a gantry portal type milling machine. Zaeh et al. [8] and Kleinwort et al. [19] used a model-based H_∞ controller to suppress chatter in milling and turning processes. Wan et al. [21] used a nonlinear sliding mode controller by applying active damping forces to the rotating spindle through a non-contact electromagnetic actuator, and mitigated chatter vibration in the milling process. Though model-based control strategies have been shown to be effective [8, 16, 19, 21], since they require dynamical models of the machine and the process, they are less suited for industrial implementation, which prefers the simpler to implement model-free control schemes such as acceleration feedback [9, 10], velocity feedback [9, 10, 12, 13, 17], position feedback [9, 10, 22], delayed position feedback [9, 22, 23], delayed acceleration feedback [10, 23], and/or integral force feedback [20]. Since most prior work on model-free implementations of active control of machine tool vibrations has preferred to use the simple and effective direct velocity feedback (DVF) control law [9, 10, 12, 13, 17], this research too uses a DVF control law. Since gains are usually fixed a priori in most model-free DVF implementations, and since controllers with fixed gains cannot dynamically respond to varying levels of disturbances observed during the process, adaptive vibration controllers have also found favor [14, 19, 24].

In prior work on model-free adaptive controllers, filtered-x least mean square (FxLMS) type adaptive controller was generally used, wherein filter parameters were dynamically updated by minimizing the error between a reference signal and the controller output such as to meet some targeted levels of control performance [14, 19, 24, 25]. A model-based adaptive control technique has also been found useful in suppressing machine tool vibrations, wherein a model predictive control technique was used by optimizing multi-objective target functions such as reduced production cost, time, and surface quality [26]. Though these methods have been reported to work well, FxLMS requires the real-time least square error minimization of signals that can be time-consuming, and model predictive controller requires prior knowledge of the model of the system, which is not very effective when the dynamics of the machine tool change considerably. Furthermore, other aspects related to the stability of such adaptive controllers preclude its industrial use.

Since industrial use of active vibration control could benefit from a simple to implement adaptive and model-free gain updating method, this paper offers a new method to dynamically adapt gains to the level of unstable vibrations detected during machining. In doing so, we demonstrate that the actuator supplies only as much force as is necessary to damp the unstable vibrations, and hence is more energy-efficient than the standard methods of supplying a fixed gain.

Our proposed method of dynamically updating gains builds on our prior work [27]. Though our present work leverages learnings from [27], it differs from that work. [27] was concerned with emulating bistabilities in a highly interrupted turning process using a mechatronic hardware-in-the-loop (HiL) simulator, whereas the present work focuses on a real end milling process. Moreover, [27] tested the model-free adaptive gain tuning scheme to respond to small and large perturbations for emulated cutting within the bistable region, whereas the present work is agnostic to any potential nonlinearities in the process. It simply updates gains based on the level of unstable vibrations detected during machining.

The rest of the paper is organized as follows: at first, the experimental setup is described. The second main section discusses the dynamic characteristics of the machine and of the actuator. These characteristics inform signal processing and conditioning for effective model-free active vibration control. The third main section describes the proposed adaptive gain updating scheme. The fourth section discusses experiments with active vibration control and characterizes the proposed scheme's efficacy for its dependence on the time required to update the gain and for its dependence on the levels of gain increments. The main conclusions follow.

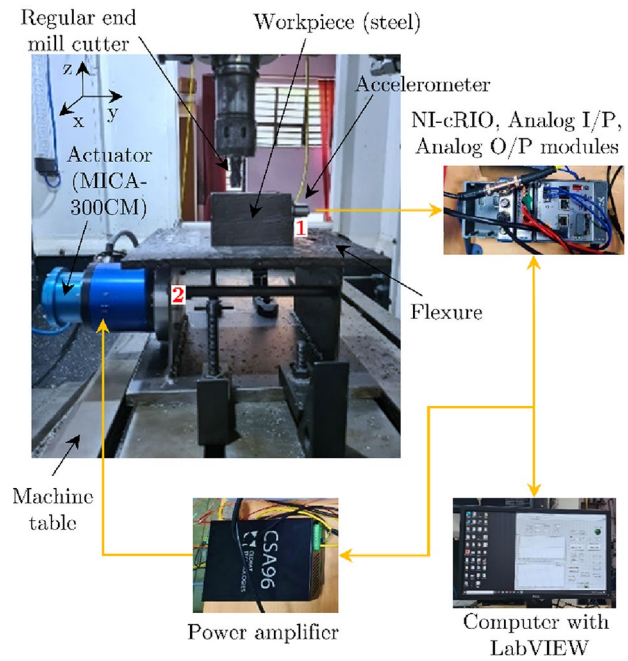

Fig. 1 Experimental setup to demonstrate adaptive gain updating for active damping in a milling process

The Experimental Setup

The experimental setup to demonstrate adaptive model-free gain tuning to actively damp vibrations in a milling process is shown in Fig. 1. Experiments are conducted on a three-axis vertical CNC milling machine (Make: AMSL, Model: ACER) with a four-fluted regular end mill cutter made of carbide that has a 30° helix angle. Since the machine in question is dynamically very stiff, we demonstrate active damping of a specially designed flexure on which a steel workpiece was screw mounted. Such setups are common for testing of active damping strategies [12]. The workpiece holder/flexure was designed to be flexible only in one direction, i.e., in the y-direction. An accelerometer (Make: Wilcoxon, Model: 785A) was magnet mounted on the workpiece to monitor the cutting process. An electromagnetic actuator (Make: Cedrat Technologies, Model: MICA300CM) was screw mounted on the wall of the flexure, as shown in Fig. 1. The actuator applies a compensatory force on the flexure when it vibrates under the action of cutting forces. The system shown in Fig. 1 is not collocated. Also shown in Fig. 1 is a National Instruments (NI)-cRIO 9040 real-time processor along with an analog input (NI9234) and an analog output (NI9263) module. The input module is used to acquire accelerations. The output module is used to provide a controlled signal to the power amplifier (Make: Cedrat Technologies, Model: CSA96) of the actuator (shown in Fig. 1) for it to apply a suitable compensatory force as and when required. Also shown in Fig. 1 is a computer running NI LabVIEW for data acquisition, signal conditioning and processing, and for providing the necessary input to the actuator based on the control law that follows an adaptive gain tuning implementation. Implementation of the proposed adaptive gain updating scheme depends on the dynamic characteristics of the machine-workpiece-flexure system and on the dynamics of the actuator. These are hence characterized as described next.

Characterization of the Dynamics of the Machine and the Actuator

For effective implementation of the adaptive gain tuning scheme, knowledge of the actuator's dynamics described by its force-frequency characteristics is necessary. And even though the proposed model-free gain updating strategy does not strictly require a priori information of the machine's dynamics, to inform system identification of the actuator over the frequency range of interest, knowledge of the dynamics of the machine-workpiece-flexure system is important. Furthermore, since the model-free direct velocity feedback control law to be implemented requires measured accelerations to be integrated, appropriate conditioning and filtering of measured accelerations also require knowledge of the dynamics of the machine-workpiece-flexure system.

Modal analysis of the system

The dynamics of the machine-workpiece-flexure system were measured using an instrumented modal hammer (Make: DYTRAN, Model: 5800B4) and an accelerometer (Make: Wilcoxon, Model: 785A). Since the cutting tool will excite the workpiece, measuring dynamics at the tool-workpiece interaction location, i.e., at location '1' highlighted in Fig. 1 is important. Also, since the system is non-collocated, measuring the dynamics for when the actuator excites the flexure at location '2' shown in Fig. 1 and for which there is a corresponding response at location '1' is also important. Hence, dynamics characterized by direct and cross-frequency response functions (FRFs) were measured accordingly by impacting the workpiece at location '1' and by measuring the response also at location '1' to result in the direct h_{11} FRF, and by subsequently impacting the flexure at location '2' with the actuator mounted on the flexure and measuring the response also at location '1' to result in the cross h_{12} FRF. Measurements were made using CutPRO's MALTF® module [28]. Measured FRFs are shown in Fig. 2.

It is evident from Fig. 2 that the flexure has only a single dominant mode in the frequency range up to 500 Hz. Higher frequency behavior was several orders of magnitude stiffer than the dominant mode and is hence not shown in Fig. 2. Also evident is that the direct FRF is more flexible (higher peak magnitude) than the cross FRF. Reciprocity was checked for and verified to hold true, i.e., the workpiece was excited at location '1' and response was measured at location '2', i.e., the h_{12} FRF was found to be the same as the h_{21} FRF. Furthermore, since the flexure is designed to be flexible only in y-direction, measurements reported in Fig. 2 are limited to that direction. Separate measurements were made of the tool in its x- and y-directions as well as of the flexure in its x-direction, and those FRFs were observed to be dynamically stiffer by at least an order more than the FRFs of the flexure in the y-direction. Those FRFs are hence not shown herein. Modal parameters identified from FRFs in Fig. 2 are listed in Table 1. As is evident, since the direct and cross FRFs have the same natural frequency (f_n) and damping (ζ), their modal mass (m) is naturally different.

Since the actuator must apply a compensatory force to damp the flexural mode at 71.5 Hz, it is important to identify the force-frequency characteristics of the actuator over the frequency range of interest. This is described next.

Characterization of the Actuator

The experimental setup for the force-frequency characterization of the actuator is shown in Fig. 3. The actuator is mounted vertically on a three-axis piezoelectric type force dynamometer (Make: KISTLER, Model: 9257BA), which is in turn mounted on a rigid table. Since the actuator applies a force only along its mounting axis, only one component (Z-component) of the dynamometer is monitored. A sine chirp signal from a signal generator ranging from 10 to 200 Hz with a peak-to-peak voltage of 1 Vpp is supplied to the actuator through its power amplifier. Voltage limits are fixed such as to not damage the actuator. The current drawn by the actuator is measured using the built-in current sensor in the power amplifier of the actuator. The time-series data of the current and the force

Table 1 Modal parameters of the flexure-workpiece-actuator system

FRF	f_n (Hz)	ζ (%)	m (kg)
h_{11}	71.5	1.3	22.4
h_{12}	71.5	1.3	46.5

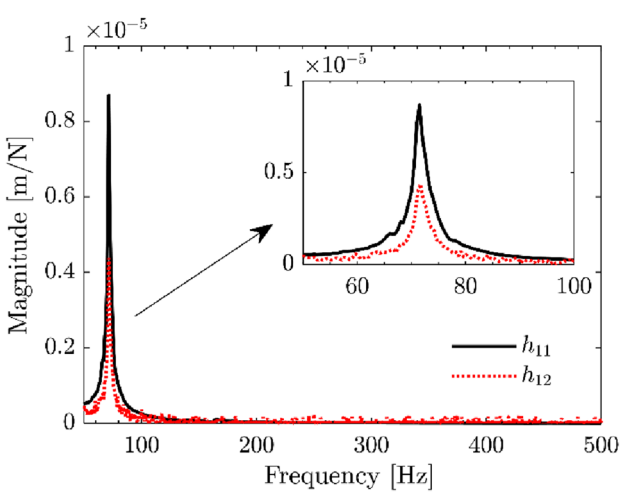

Fig. 2 Measured FRFs of the flexure-workpiece-actuator system

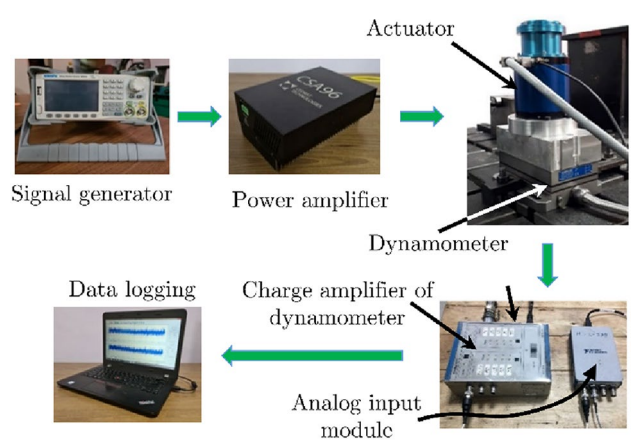

Fig. 3 Experimental setup for identifying the force-frequency characteristics of the electromagnetic actuator

signals are acquired using an analog to digital converter (NI9234) and logged using CUTPRO®'s data acquisition module [28].

The measured time-domain responses are decomposed to the frequency domain, and the resulting frequency response function is shown in Fig. 4. As is evident, the actuator's suspension frequency is ~ 25.6 Hz, and beyond ~ 50 Hz, the actuator applies a constant force over the range of frequencies characterized, with a force constant of $g_{NA} = 13.5$ N/A → $g_{NV} = 27$ N/V, wherein 2 A/V is the current to voltage gain of the power amplifier. As is further evident from the phase–frequency characteristics of the actuator, the phase is negative beyond the actuator's mode, and this may contribute to a delay in the closed-loop system. If delays in the closed-loop system are found to be significant, they can be compensated as detailed in [22].

Knowledge of the actuator's dynamic characteristics together with knowledge of the dynamics of flexure-workpiece-actuator system inform the design of the proposed adaptive model-free gain updating scheme as is detailed next.

Adaptive Model-Free Gain Updating Scheme

A block diagrammatic representation of the proposed adaptive model-free gain updating scheme is shown in Fig. 5. The workpiece vibrates under the action of cutting forces. The accelerometer measures these vibrations, and because we prefer a direct velocity feedback control law, the accelerations are integrated. Since the actuator has a suspension mode at ~ 25.6 Hz, we use a second-order high-pass filter with a cut-off frequency of 40 Hz. We also use a second-order low-pass filter with a cut-off frequency of 2 kHz. Damping ratio of both filters is kept 0.707. For the low-pass filter, the cut-off frequency is chosen such as to not filter out the relevant harmonics of tooth passing frequencies. Since we plan to cut steel with a four-fluted cutter at speeds up to 2400 rpm, the highest expected tooth passing frequency is 160 Hz [four teeth × maximum spindle frequency (2400 rpm/60)]. And a cut-off of 2 kHz will ensure that during stable cutting, even the 10th harmonic is captured. Filtered signals are supplied to the adaptive gain tuning scheme, more on which is discussed subsequently. The output from the gain tuning scheme is converted to a voltage signal using the actuator's identified force constant (g_{NV}) and provided to the actuator if/when instabilities are detected. To ensure safe operation, we implement a force saturation block.

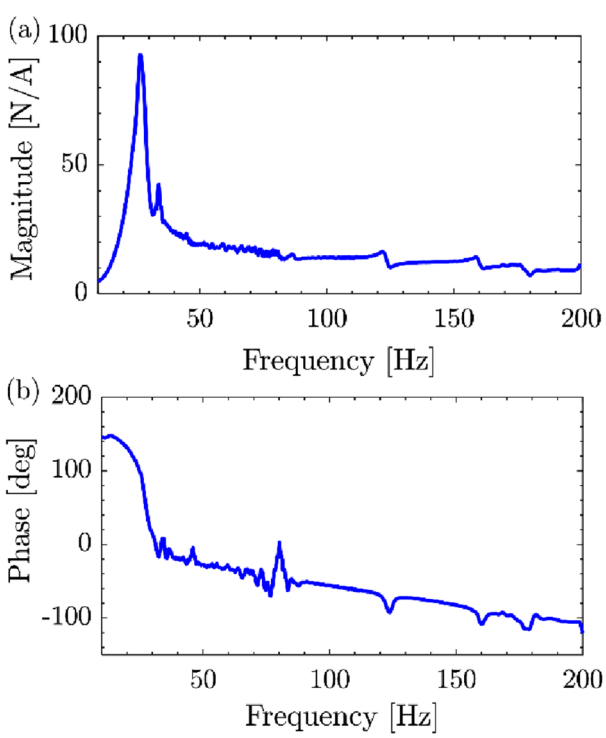

Fig. 4 Measured force/current frequency characteristics of the MICA. **a** Magnitude. **b** Phase

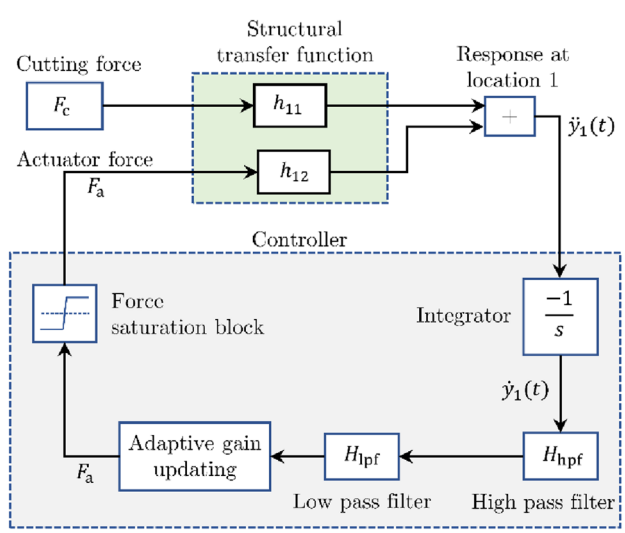

Fig. 5 Block diagrammatic representation of closed-loop active damping scheme

Adaptive Gain Updating Scheme

In an active damping system, ideally, the control gain should be tuned to the level of unstable vibrations detected. This would ensure that only as much force as necessary is supplied by the actuator and that the actuator is not always operated at its full capacity. To do so, we propose an adaptive gain tuning method that adaptively changes the closed-loop control gain as/when required. A flowchart outlining the proposed method is shown in Fig. 6.

The adaptive tuning method works as follows: with the start of the cutting process, the control gain K_{dvf} is initialized to zero, so that the actuator does not apply any force to the system. During the cutting process, accelerations are monitored, and the frequency content (via a real-time FFT) of the data is used to decide whether the cut is stable or unstable. The cutting process is considered stable if the dominant frequency in the frequency spectra, i.e., $f_{A_{\max}}$ is a harmonic of the spindle frequency (f_s). The spindle frequency is monitored instead of monitoring the tooth passing frequencies to account for any potential runouts in the system. If the process is stable, active damping is not necessary, and no voltage is supplied to the actuator. If, on the other hand, the frequency spectra of the cutting signal contain content at frequencies that are not harmonics of the spindle frequency, i.e., $f_{A_{\max}} \neq nf_s$ for $n \in 1:10$, then the cut is deemed unstable. The choice of n will vary depending upon the natural frequencies of the structure under test and operating range of spindle speed. In such an unstable case, the controller is turned on, and the default gain supplied is that with which the system is initialized. If this gain does not help stabilize the process, an adaptive gain K_{adapt} is incremented with dK, and since the force applied for the actuator is proportional to the gain, the actuator applies a higher force to stabilize the cutting process.

A saturation block is implemented, so that the updated gain value does not exceed K_{sat} to ensure the safety of the actuator and the machine tool structure. For sufficiently low values of dK, the gain updation is very slow, and the actuator will take time to mitigate chatter vibrations, and in that time, the machine might get damaged, which is not desirable. On the other hand, for the higher values of dK, although the actuator might be fast enough to suppress the vibrations, in doing so, it may apply an impulse force and the updated gain may reach the saturation value and therefore, the actuator may saturate in force, and there is no scope of further increasing the control gain. Also, with higher than necessary gains, the actuator uses more power, and more energy is used to suppress chatter vibrations than required. Therefore, the dK value must be chosen, such that the gain updation is sufficiently fast, so that the chatter vibrations are suppressed quickly, and at the same time, the actuator does not saturate. The efficacy of this proposed scheme is tested in real cutting processes for its dependence on the time required to update the gain and for its dependence on the levels of gain increments—as discussed in subsequent section. This proposed scheme is model-free and much simpler to implement than other previous adaptive gain updating schemes found in the literature.

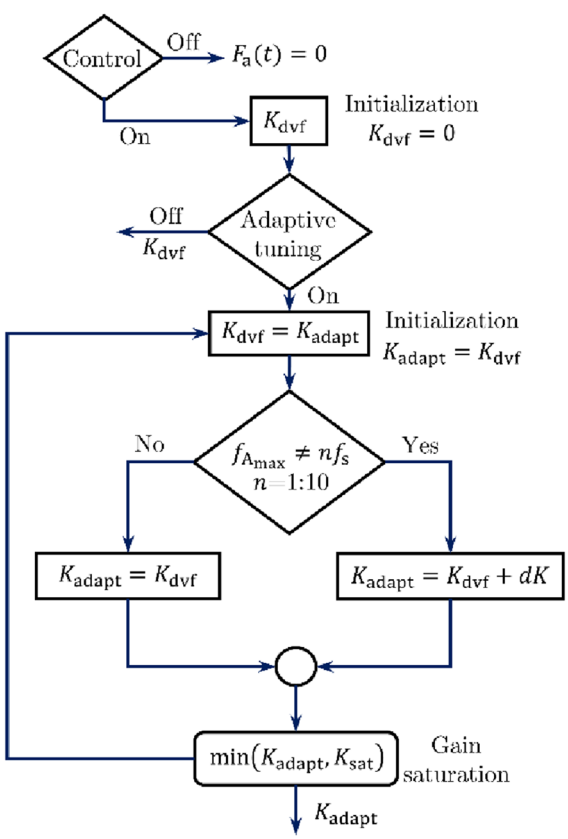

Fig. 6 Flowchart of the adaptive gain tuning method

Experiments with Active Damping

To demonstrate active damping in a milling process, as a guide to start experimentation, at first, cutting was conducted at parameter sets corresponding to model-predicted stability boundaries. These boundaries were generated for the case of the control being 'Off' using the classical analytical frequency domain method of stability prediction [3]. Measured dynamics (see Fig. 2 or Table 1) with a tangential cutting force coefficient of $K_t = 1865\mathrm{N/mm}^2$ and a normal cutting force coefficient of $K_n = 598.8\mathrm{N/mm}^2$ were used to generate the stability boundary for slot milling with a regular end mill with four teeth. Experiments were conducted on the setup shown in Fig. 1. The feed rate was set to 0.15 mm/tooth/rev, and slot milling was carried out with feed along

the y-direction. Experiments were conducted at different speeds and depths of cut. Spindle speeds (N) of interest were constrained by the recommended cutting speeds (v) for steel, which in the present case ranges from 50 m/min up to 125 m/min. And, since we cut using a 16 mm tool diameter (d), the corresponding spindle speed range of interest was 1000 rpm up to 2500 rpm ($v = \pi dN$).

The procedure for finding experimental stable/unstable points was as discussed in the section on 'Adaptive gain updating scheme'. For the present case, the corresponding spindle frequencies of interest in the case of forced stable vibration will be ~ 16.66 Hz (1000 rpm/60) to ~ 41.66 Hz (2500 rpm/60). And, since the forced response can have frequency content even at higher harmonics of this spindle frequency, we monitor the signals up to the 10th harmonic, i.e., up to ~ 416 Hz to classify if the cut is stable or not. Experimentally identified stability points for both cases are overlaid with theoretically generated stability lobes shown in Fig. 7. And as is evident, experiments match model predictions for the case of the control being 'Off'.

Experiments were subsequently conducted with the controller being 'On' and with the gain set at a value of $K_{\mathrm{dvf}} = 2000\mathrm{Ns/m}$, i.e., the saturation value, for all experiments, i.e., adaptive gain updating was disabled. The mechatronic delay in the closed-loop system was found to be negligible, and hence, no delay compensation was necessary. Experiments were done to investigate the capability of the active vibration control system to detect and suppress chatter in real time. And like for the case of the control being 'Off', in this case, too several experiments were conducted at different combinations of speeds and depths of cut. Though model predictions with the control being 'On' were possible, since the focus of this paper is to demonstrate model-free active damping, predicted boundaries for the control being 'On' are not shown in Fig. 7. However, it is evident from Fig. 7 that the minimum stability limit with Control 'On' case is significantly higher when compared with the Control 'Off' case. The cutting experiments with the control being 'On' were performed only up to a depth of cut of 0.65 mm. The forced vibration response of the workpiece-flexure system was very high for cutting at these depths of cut, and to ensure that there is no damage to any of the elements of the machine, experiments at higher depths of cut were avoided. However, even cutting at a depth of cut of 0.65 mm, the improvement from the case of the control being 'Off' is a significant 300%—suggesting that active control of vibration can not only stabilize cutting, but it can also improve the productivity potential.

For a representative case of cutting at a depth of cut of 0.25 mm at a speed of 1560 rpm, response levels for the control being 'Off' and 'On', and the correspondingly obtained surfaces are shown in Fig. 7. Also shown in Fig. 7 are the frequency spectra of the response signals.

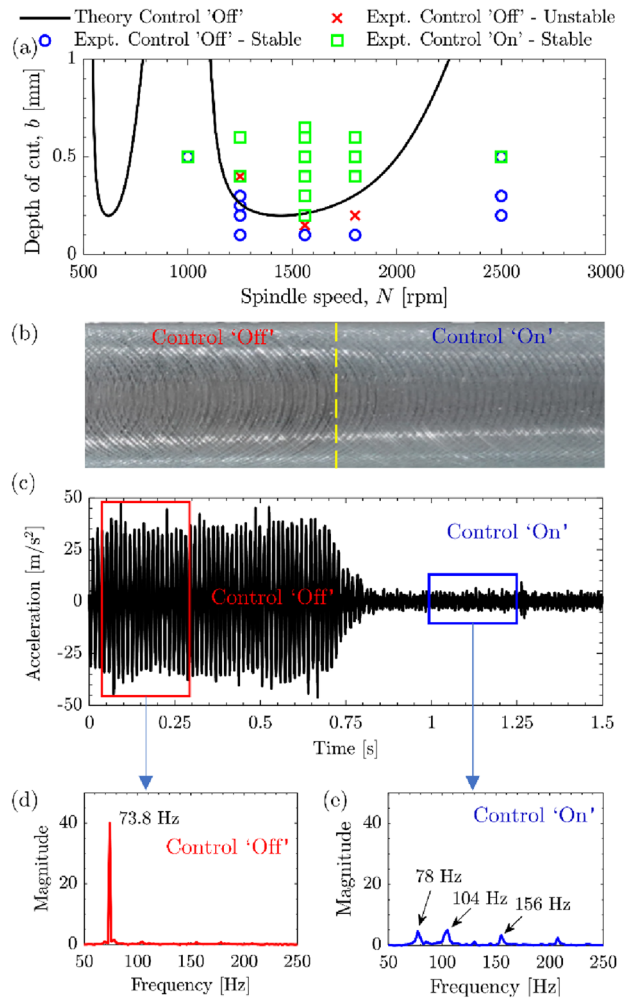

Fig. 7 **a** Stability behavior with Control 'Off' and 'On', **b** surface roughness after cutting, **c** measured acceleration signal, **d** FFT of the acceleration signal, and **e** FFT of the acceleration signal. Note: for **b**–**e** experiments were performed at $N = 1560$ rpm and $b = 0.25$ mm, and for Control 'On' case, the gain was set to $K_{\mathrm{dvf}} = 2000$ Ns/m

It is evident from Fig. 7 that when the active control is 'Off', chatter marks are imprinted on the cut surface of the workpiece, and chatter vibrations are also evident from the acceleration signal and corresponding frequency spectra. From Fig. 7d, it is evident that chatter occurs in the vicinity of the natural frequency of the flexure, i.e., at 73.8 Hz. The Control 'On' case stabilizes the cut as is amply evident from the cut surface as well as from the corresponding accelerometer signal and from the frequency spectra showing peaks at only the harmonics (3rd, 4th, and 6th) of spindle frequency (1560 rpm/60 = 26 Hz), see Fig. 7e.

Though active damping of chatter on a real machine using a direct velocity feedback control scheme at a fixed control gain was shown to be effective, it may not always stabilize processes with nonlinearities and/or those prone to interruptions or perturbations [29]. Such processes may benefit

from an adaptive gain updating method in which the gain is adjusted as per the level of disturbance detected.

For the adaptive gain updating method to be effective, it is important to characterize the scheme's dependence on the increment in gain (dK) made in every step and the time required to make that increment. And for both these, paramount is evaluation of the real-time FFT of the vibration signal, for which a sample of data points is needed. For accurately measuring all the harmonics of the excitation frequency, 750 data points were considered in a sample, and because of which the FFT calculation takes 0.15 s. This is called the DAQ time in this paper. Since it is known that chatter occurs almost instantaneously, taking only a few milliseconds to develop fully, it can damage elements of the machine tool. Therefore, two levels of the DAQ time are considered here: 50% and 100% of the DAQ time. And, for each of these levels, the value of dK is also varied to see the combined effect of both the DAQ time and the dK value. Furthermore, since signals are inevitably corrupted by noise, a noise floor of 0.8 m/s^2 is kept, so that the gain updation loop gets activated only when the vibration signal is above this level.

Experiments were conducted for two levels of the DAQ time and for two values of dK. The dK values were chosen as 50 Ns/m and 500 Ns/m, which are less than the saturation value of control gain, $K_{sat} = 2000$ Ns/m. In total, four experiments were conducted. An unstable point when the controller is 'Off' in Fig. 7a, i.e., $N = 1800$ rpm and $b = 0.25$ mm was chosen for all the four experiments, and the adaptive gain for each experiment was recorded. The acceleration response obtained for the cutting experiments with adaptive gain tuning is overlaid with the response corresponding to the Control 'Off' case, as shown in Fig. 8.

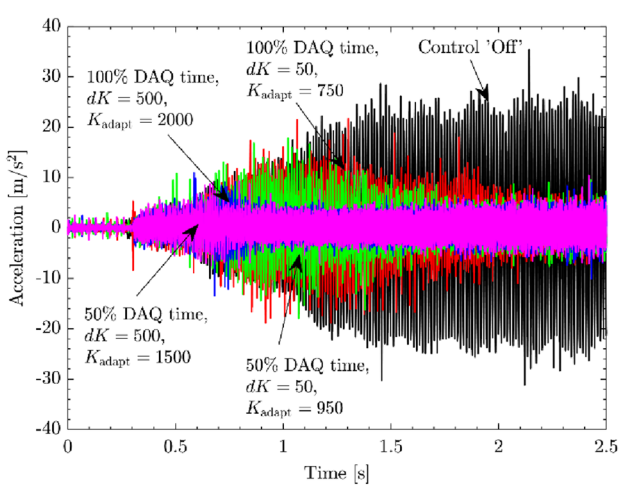

Fig. 8 Measured acceleration with adaptive gain updating for different levels of DAQ time (50% and 100% of the DAQ time) and different dK values ($dK = 50, 500$)

It is evident from Fig. 8 that the adaptive gain tuning method is effective in detecting and suppressing chatter vibrations during real cutting. It is observed that the chatter vibrations get suppressed fastest in the case of $dK = 500$ and 50% DAQ time and slowest in the case of $dK = 50$ and 100% DAQ time. It is evident that for higher dK values, the control gain increases faster; therefore, more actuator force is applied in a short period of time. Also, for the low value of the DAQ time, the number of sample points taken to evaluate the FFT plot is small, and hence, the rate of increment of the dK value is fast.

To further check the robustness of the proposed adaptive control technique, experiments were conducted at the representative speed of 1800 rpm at three different levels of depths of cut: $b \in [0.25, 0.35, 0.45]$ mm. These depths of cuts are all unstable for the control being 'Off'—see Fig. 7a. For adaptive gain tuning, the initialized control gain was kept at 0 and the maximum value it can take was limited to 2000 Ns/m. As the 50% DAQ time and $dK = 500$ was observed to be best to mitigate chatter (see Fig. 8), these values are used herein for cutting at the three different depths of cut of interest. Measured acceleration responses with adaptive control technique corresponding to three levels of depths of cut are shown in Fig. 9.

As is evident from Fig. 9, the adaptive controller stabilizes the cut for all three levels of depths of cut. Since the depths of cut are different, the stabilized forced vibration amplitudes are also different, with the higher depth of cut having a larger forced response than the smaller depths of cuts. From Fig. 9, it is also clear that the control gain adapts in near real time to the level of unstable vibrations, which in turn depend on the depths of cut. Since the responses corresponding to 0.35 mm and 0.45 mm depths of cut are higher than the response corresponding to the 0.25 mm depth of cut, the control gain stabilizes at the maximum

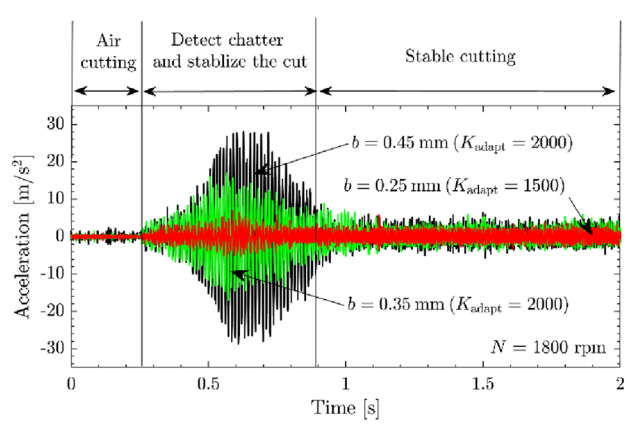

Fig. 9 Measured accelerations during real-time active damping of machine tool vibrations using the proposed adaptive model-free gain updating scheme

value, i.e., at K_{adapt} = 2000 Ns/m for cutting at depths of cut of 0.35 mm and 0.45 mm, respectively, whereas for the depth of cut of 0.25 mm, the gain stabilizes at K_{adapt} = 1500 Ns/m. Though the adaptive scheme is deliberately designed to supply only as much gain as necessary to damp the level of unstable vibrations detected during machining, it is possible that even with different levels of unstable vibrations detected, the controller may supply the same gain. This is not due to convergence, but due to the gain saturating at the user-defined upper limit.

Conclusions

This paper presented a novel and model-free adaptive gain tuning method for active vibration control of a milling process by detecting chatter in near real-time, and suitably suppressing it with the use of an actuator. For a fixed level of control gain, a ~ 300% improvement in productivity was demonstrated. With the adaptive gain tuning scheme, higher gain increments with shorter updating times were observed to result in the process being stabilized quicker.

Since our proposed method is model-free, it offers advantages over other model-based methods in situations wherein the dynamics of the machine tool change considerably and/or when there exist cutting process nonlinearities. Model-based schemes would require a re-design of the control strategy or use of other advanced and robust control strategies in such cases. Whereas our model-free approach works simply by adapting the control gain to the level of unstable vibrations detected during machining. And, by adapting the control gain in this manner, our model-free control is quicker to implement in 'real-time' as opposed to other model-free methods that updated filter parameters dynamically by requiring a real-time least square error minimization of signals, which is computationally inefficient and is hence slower to respond to instabilities. Furthermore, since the actuator supplies only as much force as is necessary to damp the unstable vibrations, the method is energy-efficient, i.e., it draws less current than a control strategy that operates with a fixed and potentially higher than necessary gain. These advantages make the proposed scheme suitable for industrial use.

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Declarations

Conflict of Interest On behalf of all authors, the corresponding author states that there is no conflict of interest.

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