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NEW PARADIGM IN CONTROL OF MACHINING SYSTEM'S DYNAMICS

The increasing demands for precision and efficiency in machining call for effective control strategies based on the identification of static and dynamic characteristics under operational conditions. The capability of a machining system is significantly determined by its static and dynamic stiffness. The aim of this paper is to introduce novel concepts and methods regarding identification and control of a machining system's dynamics. After discussing the limitations in current methods and technologies of machining systems' identification and control, the paper introduces a new paradigm for controlling the machining system dynamics based on design of controllable structural Joint Interface Modules, JIMs, whose interface characteristics can be tuned using embedded actuators. Results from the laboratory and industrial implementation demonstrate the effectiveness of the control strategy with a high degree of repeatability.

1. INTRODUCTION

Complex mechanical components machined with zero defects are an essential condition in precision manufacturing and it becomes a new challenge for the next generation intelligent machining systems. Improved precision and accuracy of machines, processes and components offers substantial benefits to a wide range of applications from ultra-precision to mass customization with higher quality and better reliability. Within this context, this paper identifies critical problems that limit the performance of the machining system and address them by advancing novel solutions.

The accuracy of a machining system is determined by the interaction between machine tool structure and cutting process and it is affected by variations introduced by various disturbance sources [1]. These sources can be classified in: positioning and kinematic errors,

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temperature errors, and static and dynamic errors. In this paper machining system capability is studied with respect to the static and dynamic behaviour. Under machining conditions, detrimental vibrations are susceptible to arise out of the interaction between the process and the machine tool structure dynamics. These vibrations may be divided into three basic types:

- Free or transient vibration;
- Forced vibration;
- Self-excited vibration.

Improving the geometrical and dimensional tolerances and surface quality of the machined components requires efficient methods for identification and control of the machining system vibration. This paper proposes a new approach for identification and control of the machining system's dynamics which is based on the ability to identify the condition of the system and to control the structural properties of the machine tool.

Increasing quality demands for the produced parts call for machining systems being able to respond to high variability in the processes they have to carry out, without compromising the accuracy of the part and/or the productivity of the process. This requires efficient control of the machining system in order to maintain its static and dynamic stability. It is known that as the force path is closing through the tool/workpiece interface, the stability of the system can be achieved either by controlling the process parameters or the machine tool structure. Therefore, in order to expand the stable ranges of the machining system without compromising productivity, control strategies should move from the traditional paradigm of control through the process and focus on ways of controlling the structure of the machine tool [2].

From the point of view of dynamics of the machining system there are two critical issues to be discussed: (i) the discrimination between forced and self-excited vibration and (ii) the prediction of stability limit. With respect to the first issue, the lack of qualitative criteria for evaluation of a machining system´s dynamical behaviour led to extensive use of FFT analysis for a relative comparison of signals recorded during machining [3], or simply the amplitude of measured signal [5]. Further computational analyses are then performed to detect whether or not chatter occurred. As these analyses are based on signal amplitude measurement at certain critical frequencies they are relative in their nature and consequently lead to subjective decision making Budak [4] uses visual observation of the chatter marks on the workpiece to validate the prediction of stability limits.

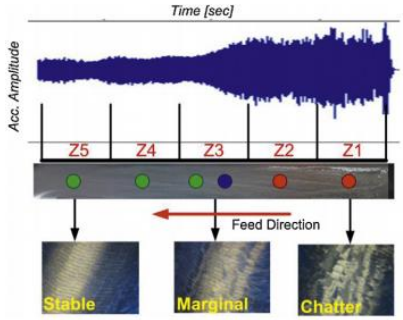


Fig. 1. Accelerometer data and surfaces of stable and chatter zones [5]

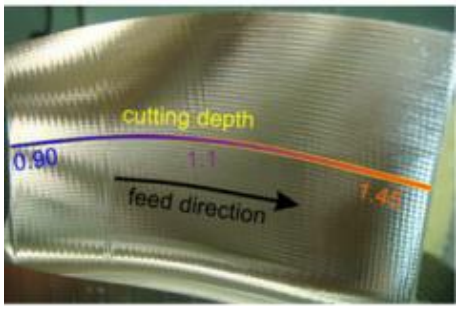


Fig. 2. Chatter marks in milling [4]

With respect to the second issue, the main techniques used today are related to a judicial use of "stability lobes" to find relatively stable operating regimes [7],[8].

2. IDENTIFICATION OF THE MACHINING SYSTEM'S DYNAMIC BEHAVIOUR

In the classical machining theory and practice, a large body of research has been dedicated for identifying the machining system's dynamics. Tlusty [6] had a major contribution in developing basic chatter theories. At a closer examination there are some phenomenological and technical shortcomings in the classical methodology. One critical issue is that the parameters describing dynamic behaviour of machining systems are extracted in the open loop system for the structure and in the closed loop for the process (see Fig. 3).

Fig. 3. Open-loop machining system (a), closed-loop machining system (b)

In the classical chatter theory based on the stability lobe prediction, it is required to measure the Frequency Response Function of the machine tool-workpiece system, cutting coefficients and the execution of stability law [9]. Accordingly, the traditional identification of machining system's dynamic parameters has invariably been approached in the following steps:

1. Identification of the dynamic properties of elastic structure of machine tools (open loop configuration). Commonly this step is done experimentally often using experimental modal analysis (EMA);
2. Identification of the characteristics of cutting process, i.e. the dynamic parameters describing the transfer function of the subsystem represented by cutting process dynamics;
3. Evaluation of stability lobe diagram of the machining system from step 1 and step 2.

Recently, Ito [10], studied essential features of chatter for establishing a unified chatter theory. In order to overcome the difficulty of measuring the frequency response function (FRF) through the impact test, Powalka [11] implemented an alternative method based on operational modal analysis (OMA) which allows to extract modal parameters from the

acceleration signal measured during actual machining. Zaghbani [12] proposed an algorithm of the OMA limited to the estimation of modal damping and natural frequency.

The approach presented in this paper has the purpose of identifying the operational dynamic parameters (ODPs) of a machining system i.e., the equivalent stiffness and damping of the system during operation in the closed-loop configuration. The approach introduces a probabilistic concept where parametric identification models are employed [26].

A parametric model is a special class of representation of a system, where the input in the model is driven by white noise processes and the model is described by rational system functions, including autoregressive (AR) (Burg, least square, Yule Walker, geometric lattice, instrumental variable), ARX (autoregressive with eXogeneous variables, iv4), moving average (MA), autoregressive-moving average (ARMA), Box Jenkins, Output Error models [13],[14]. The process output of this class of models has power spectral density (PSD) that is entirely described in terms of model parameters and the variance of the white noise process [15],[16]. Parametric models can be applied to any numbers of DOF structures, and in their recursive implementation can take into account the nonlinear nature of the system [17].

The response generated by a machining system is identified in a parametric ARMA model [18]. The model is a synthetic representation of the measured response. Using the model coefficients, the synthetic model is converted in a physical model for extracting operational dynamic parameters; damping ratios and natural frequencies (see Fig. 4).

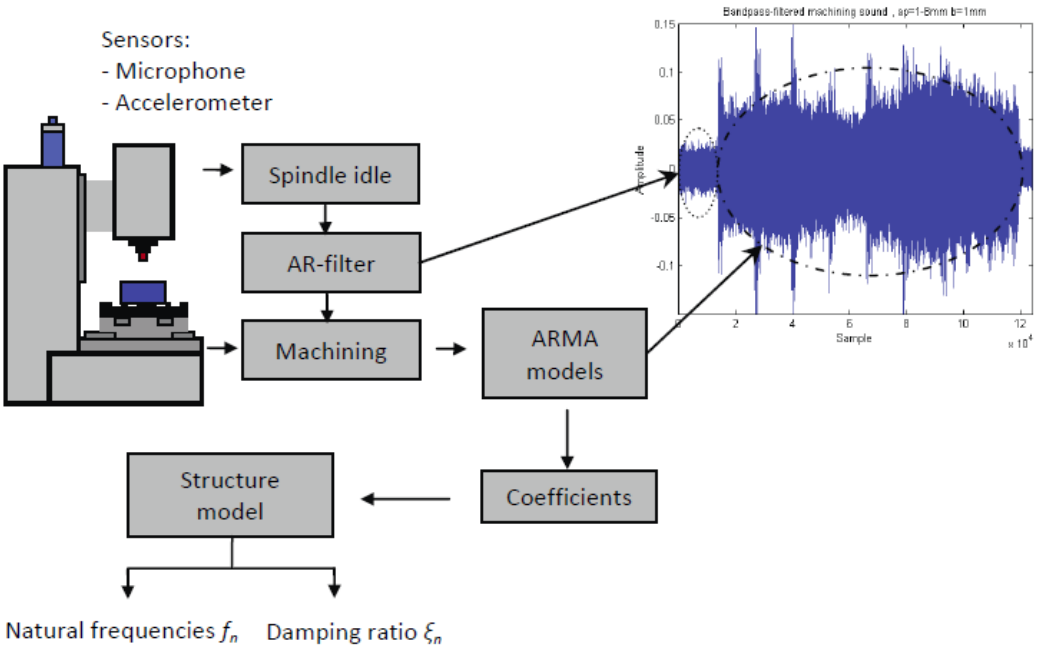


Fig. 4. ARMA model identification [19]

The modelling of a stationary time series as the output of a dynamic system whose input is white noise n(t), can be carried out in several ways. One way is to use the

parsimonious parameterization which is employing ARMA(p,q) representation were p is the order of the autoregressive part and q is the order of the moving average part. The input excitation in an ARMA process is not observable but is assumed to be random and broadband compared with the measured output sequence for the reasons explained above. The model for an ARMA process can be expressed as [20]

$$Y(z) = H(z)U(z) \tag{1}$$

where *Y(z), U(z)* and *H(z)* are the z-transforms (the z-transform is the discrete-time counterpart to the Laplace transform for continuous-time systems) of the output sequence, input sequence and the system impulse response (transfer function), respectively, and

$$H(z) = \frac{b_0 + b_1 z^{-1} + b_2 z^{-2} + \ldots + b_q z^{-q}}{1 - a_1 - a_1 z^{-1} - a_2 z^{-2} - \ldots - a_p z^{-p}} \tag{2}$$

where the b_i and a_i are coefficients of the polynomials of the MA part and AR part, respectively. As mentioned earlier, the ARMA model consists of two parts, an AR part and an MA part. Using Eqs. (1) and (2) the ARMA model can be expressed

$$\sum_{i=0}^{p} a_i y(t-i) = \sum_{i=0}^{q} b_i y(t-i), a_0 = 1 \tag{3}$$

3. RECURSIVE PARAMETER IDENTIFICATION

The model-based identification method used in this paper is based on the recursive prediction-error method (RPEM) [21],[22]. As before, the model structure is based on a parametric process where the input to the model is driven by white noise processes and the model is described by a rational system function and represented by the recursive autoregressive moving average (RARMA) model structure [23]. The process output of this model has the power spectral density (PSD) that is entirely described in terms of model parameters and the variance of the white noise process. By definition, a non-conservative mechanical system with positive damping is said to be dynamically stable, whereas one with negative damping is considered unstable. This gives a robust criterion for discrimination between forced and self-excited vibrations which is not related to de vibration amplitude criteria. Assuming that the machining system excited by a random excitation *e(t)* can be represented by an *n* degree of freedom nonlinear equation of motion

$$M\ddot{x} + C\dot{x} + Kx + g(x, \dot{x}) = e(t) \tag{4}$$

where *M, C* and *K* represent (n x n) mass, damping and stiffness matrices respectively; *e(t)* is a vector of external excitation. Matrices *C* and *K* contain both structural and process damping and stiffness respectively. The expressions are (n x 1) vectors of displacement,

velocity and acceleration for the *n* degrees of freedom system, and is a nonlinear function. The system of equations (4) can be recast in

$$\dot{z}(t) = F(z) + f(t) \tag{5}$$

where

$$z = \begin{bmatrix} x(t) \\ \dot{x}(t) \end{bmatrix}; \quad f(t) = \begin{bmatrix} 0 \\ M^{-1}e(t) \end{bmatrix}$$

and

$$\mathrm{F(z)} = \begin{bmatrix} \dot{x} \\ -M^{-1}C\dot{x} - M^{-1}Kx - M^{-1}g(x,\dot{x}) \end{bmatrix}$$

Let $y_j(k\Delta T)$, k = 0, 1, 2 …n be the discrete samples of the measurement response of the displacement of the j-mass, where ΔT is the sampling interval. Then, the observations $y_j(k\Delta T)$ can be represented by an ARMA model according to Eq. (5). The measurement equation is then of the form

$$\mathrm{Y_i = H(k\Delta t, X_j; \Theta) + e_k} \tag{6}$$

The purpose of RARMA is to recursively identify the join parameters Θ from the response measurements in the time domain

$$\begin{aligned} \widehat{\Theta}_k &= \widehat{\Theta}_{k-1} + \mu_k R_k^{-1} \psi_{k;\Theta_{k-1}} \varepsilon_{k;\Theta_{k-1}} \\ \varepsilon_{k;\Theta} &= y_k - \widehat{y}_{k;\Theta} \\ R_k &= R_{k-1} + \mu_k \left[\psi_{k;\Theta_{k-1}} \psi^T_{k;\Theta_{k-1}} - R_{k-1} \right] \end{aligned} \tag{7}$$

where ψ_k is the gradient of *y*. Thus, model parameter estimation refers to the recursive determination, for a given model structure, parameter vector Θ is a_2 … a_p, b_1, b_2, … b_q and the residual variance $\sigma_e^2(t)$ at every sample time instant k = 1, 2, … n. The AR characteristic equation of (7) can be written [25].

$$\sum_{i=0}^{p} a_i y(t-i) = \prod_{j=1}^{n} (\mu - \mu_j)(\mu - \mu_j^*) \tag{8}$$

where μ_j* is the complex conjugate of μ_j. From Eq. (8) the operational damping, ξ_j and frequency ω_j are recursively calculated at each time instant t as following

$$(\xi_{\mathrm{mod}})_j = \frac{\ln(\mu_j \mu_j^*)}{\sqrt{\ln(\mu_j \mu_j^*)^2 - 4\left[\tan^{-1}\left(\dfrac{\mu_j - \mu_j^*}{\mu_j + \mu_j^*}\right)\right]^2}} \tag{9}$$

$$(\omega_{\text{mod}})_j = -\frac{1}{2\Delta T}\sqrt{\ln(\mu_j \mu_j^*)^2 - 4\left[\tan^{-1}\left(\frac{\mu_j - \mu_j^*}{\mu_j + \mu_j^*}\right)\right]^2}$$

One of the major benefits of implementing the RARMA is the fast tracking of the instantaneous ODP under actual machining [24]. In industrial applications it is often the case that cutting conditions are changing due to variations in workpiece geometry, cutting parameters, clamping device position, relative cutting position, and machine tool kinematics.

A turning experiment demonstrates the recursive identification of ODPs. The longitudinal turning operation was carried out in a conventional turning machine with cutting conditions specified in Fig. 5. To acquire data for machining analysis, the sound from the machining process was recorded. A long slender workpiece of steel was used. The suitable order of the models was chosen after a number of trials (based on AIC), both on stable and unstable processes, in order to find the optimum order for each of the time series.

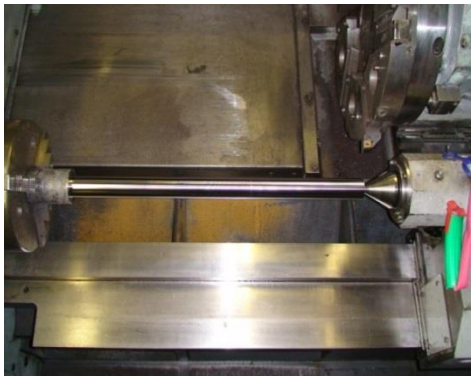


Fig. 5. Experimental set up for identification of chatter in turning

Cutting conditions

Material	Steel grade SS 2244, 0.35-0.40% C, HBS 262.
Workpiece	Length: 585mm Diameter: 45mm
Insert	Sandvik CNMG 120412
Feed	0.25 mm/rev
Cutting speed	150, 155, 160 m/min

Table 1. Model parameters in stable and unstable machining

AR parameters			MA parameters	
Stable machining				
a_1	a_2	a_3	b_1	b_2
-0.5029 ±0.0107	0.2472 ±0.0794	-0.5619 ±0.0645	-0.7519 ±0.0127	0.09649 ±0.0121
Unstable machining				
a_1	a_2	a_3	b_1	b_2
-0.3947 ±0.0273	0.4593 ±0.0181	-0.9344 ±0.0262	-0.5464 ±0.0720	0.4476 ±0.0716

The ARMA(3,2) model parameters, estimated from the appropriate models along with their standard deviations are depicted for stable and unstable machining, respectively,

in Table 1. As can be noticed, the RARMA-ODP algorithm identifies two dominant operational frequencies and related damping ratios [19].

Figure 6 illustrates the results from on-line modelling of the first and second identified modes during the machining of the slender bar. As the cutting tool approaches the middle section of the bar, the system's stiffness and damping reach very low values. The machining system becomes unstable. As the tool approaches the chuck, the system is gradually recovering its stable conditions.

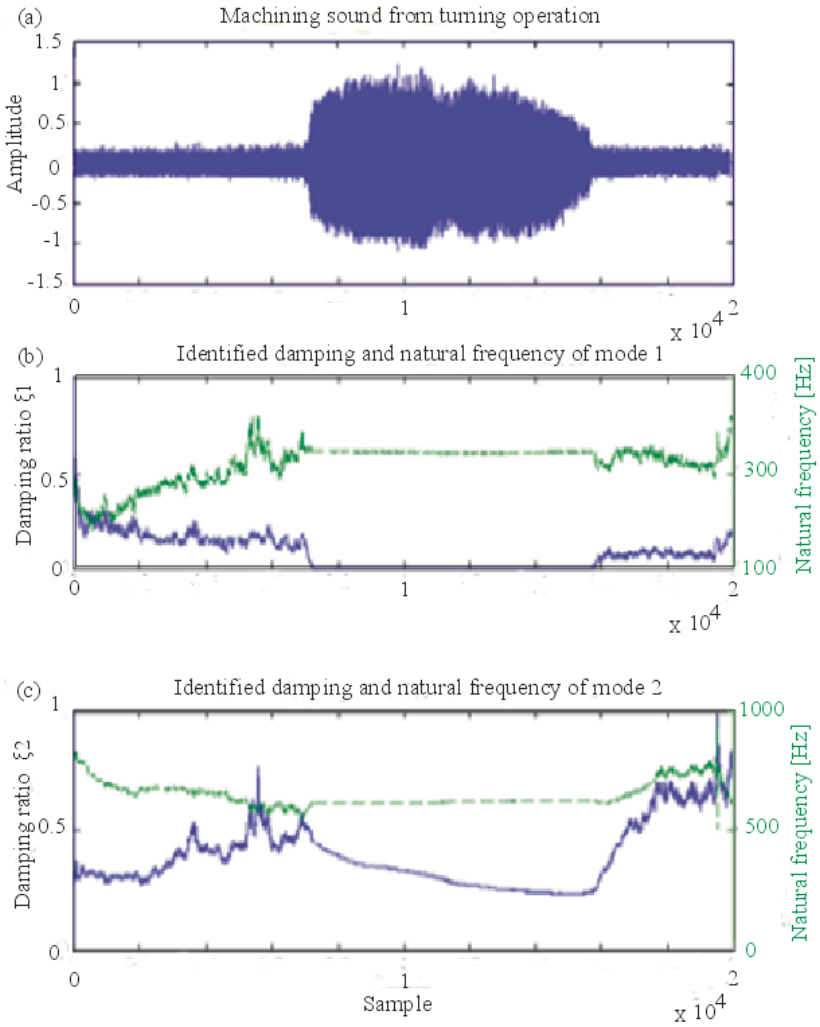


Fig. 6. In-process identification and modelling of damping and natural frequency in turning [19]. Sound pressure level measured by a microphone (a), Identified first natural frequency and damping ratio (b), Identified second natural frequency and damping ratio (c)

The sound pressure level is acquired by a microphone Fig. 6a. When chatter is completely developed the chatter frequency increases at f_1 = 290 Hz for the first mode Fig. 6b. With increased chatter intensity a significant drop in the damping ratio is apparent, see Table 1 and Fig. 6b. In Figure 6c frequency f_2 for the second mode is monitored.

4. CONTROL OF THE MACHINING SYSTEM'S DYNAMICS

Economical sustainability in manufacturing requires continuous monitoring and control of performance to fulfil all the more stiffer competitiveness and stricter customer requirements. Non-conformity and variations in part features is one of the principal challenges the manufacturing industry.

Machine tools elastic structures interacting with the machining process will deflect under static and dynamic loads respectively. Such deflections, regardless of their static or dynamic origin will eventually affect the dimension, form and surface of the machined parts. Traditional strategy for controlling the dynamic behaviour of a machining system is based upon tuning process parameters, i.e., depth of cut, rotational speed, and feed, to match with the inherent static and dynamic characteristics of the machine tool structure interacting with the process which often results in sub-optimal machining operations. Budak [5] discusses the significant role of the spindle speed combination on the stability limits. When proper spindle speed combination was selected, the total depth of cut for the operation could be increased by 25%. The effects of cutting conditions and tool geometry on process stability in turning and milling were investigated by Tunc [27]. The previously developed models by the authors are used in simulations to demonstrate conditions for enhanced process damping, and thus chatter stability. Brecher [28] gives examples of recent research activities in the field of active damping systems for machine tools while Altintas [9] presents a review of methods for adaptive control of chatter using active dampers. Jemielniak [29] developed methods for controlling chatter through spindle speed variation. Similar strategy has been implemented by Smith [31] for chatter suppression.

The present paper advances a new paradigm for controlling the machining system static and dynamic behaviour based on two fundamental principles:

1. The basic criterion for structural designing of machine tools is the rigidity criterion, i.e. maximizing the structural stiffness of machine tool components. This results in an over-dimensioned structure with respect to the strength.

It is well known that high precision and accuracy of machine tools requires high rigidity of structural components. The increasing precision requires also machine tools to withstand excitation forces that otherwise will have detrimental effects on the accuracy of the machined components. Therefore, static stiffness is together with kinematic accuracy one of the most important criteria for structural designing of machine tools. Traditionally, the enhanced of rigidity was achieved by increasing the mass of the structural machine elements. Flexible machine tool with lighter mass and lower inertia are capable for higher speed rates but meantime are therefore prone to vibration.

2. The overwhelming effect of the machine tool's structural joints on the overall machine stiffness and damping.

The negative effect of contact deformations is obvious and is due to increased compliance in structures that is caused by the structural joints [30]. Rivin [33] has shown that contact deformations in the connections, such as keys and splines, and bearings are responsible for about 40% of the effective compliance of the power transmission systems. Contact deformations in spindle units of machine tools are responsible for 30–40% of the

total deformations at the spindle end. Contact deformations in carriages, cantilever tables, etc., constitute up to 80–90% of the total, and those in moving rams about 40–70% [32].

Several research papers were dedicated to identifying joint characteristics. Sakai [34] proposes a method for evaluating the dynamic characteristics of a linear rolling bearing subjected to a radial force and a bending moment in order to evaluate the damping of a rail-carriage system. Ito [30] has thoroughly examined the effect of joints on overall deflections in machine tools demonstrated that the static stiffness of the jointed structure is duly deteriorated compared to that of an equivalent structure; the smoother the joint surface and the less the flatness deviation of the joint surface, the larger is the joint stiffness. Recognizing the critical contribution of the joints to machining system capability, Jedrzejewski [35] developed diagnostic measurement methods for regulation of sliding joints. A reproducible regulation the joint was achieved by measuring the time needed for the stabilized linear velocity of a sliding part, such as a milling table.

Though these studies and many other investigations performed in the past, there are no any attempts to design machine tools with controllable joints. Although structural joints are an excellent source of damping, the modern design practices fail to exploit this potential in developing machine tools with enhanced dynamic stiffness. This is mainly due to lack of:

- Controllability of joint characteristics;
- Suitable functional materials which can compensate for loss of stiffness while increasing damping capacity.

In addition to this, lack of theoretical models also prevents accurate prediction of the behaviour of joints under dynamic loading at the design stage. Most of the theories of elasticity available at present deal only with the problem of the monolithic elastic body, i.e., elastic body with infinitely rigid joints.

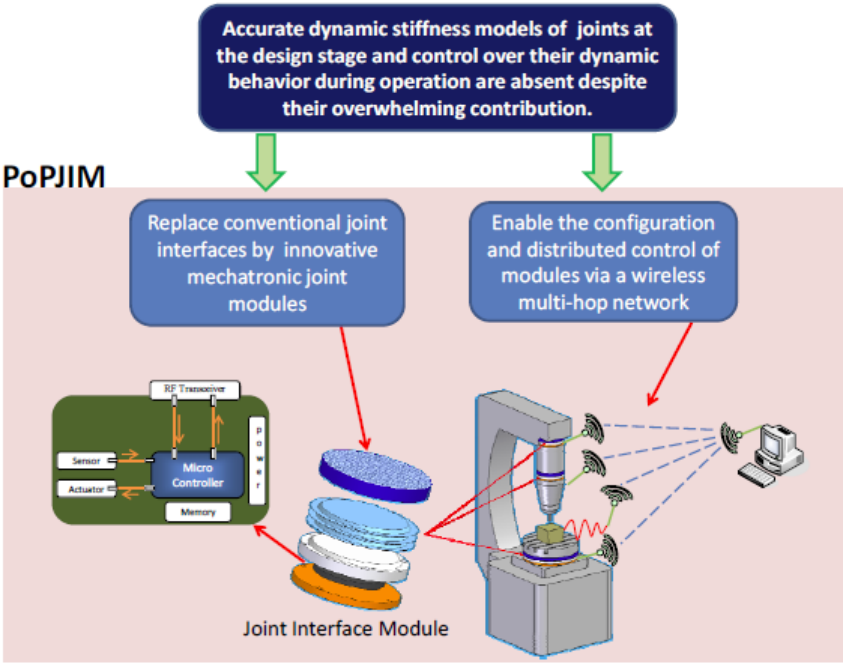


Fig. 7. Join interface module for controlling the static and dynamic behaviour

This paper discussed a new paradigm for controlling the machining system which is based on the capability to control the structural properties of the machine tool and as a result, controlling the outcome of the machining process. Novelty of the concept presented in this paper is to control the machining process dynamics by adjusting dynamic stiffness of the machine tool at structural joints. In other words, instead of changing the process parameters, dynamic stiffness of the machine tool is tuned to maintain the process stability. With this approach, the process efficiency is not sacrificed as the process parameters remain unchanged while the overall dynamic stiffness of the machine tool structure follows the demands of voluntary or involuntary changing machining conditions. The overall concept for controlling the machining system is illustrated in Fig. 7.

Carefully designed damping and stiffness characteristics i.e. the dynamic stiffness in the joints allows the dynamic behaviour of the machine tool to be predictable with a higher degree of accuracy. The mechatronic design of joints includes an integrated control system which enables it to be self-adaptive for optimising the dynamic stiffness within its design range during a machining operation. A demand beyond the design range of a specific joint is met by reconfiguring the machine tool with one designed for the required level of performance.

4.1. CONTROL STRATEGY

Control of the machining system is implemented in two steps:

1. Real-time Monitoring and Identification of the machining system dynamic condition;
2. Control of the prestress in joint to tune the stiffness and damping according to the system condition.

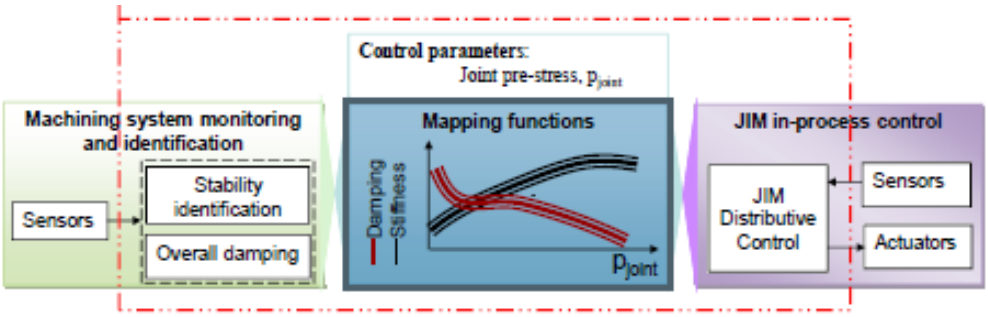


Fig. 8. Control architecture of the Joint Interface Modules (JIM)

Signals from acceleration and acoustic sound sensors are used to identify and model the system dynamic condition. The relationships between pre-stress and the stiffness and damping in joints are then used to tune the parameters of the joint at required level by the control algorithm using piezoelectric actuators. The control architecture (see Fig. 8) consists of primary and secondary stages of control. In the primary stage of control, a master

controller acquires the incoming vibration signals and estimates the joint parameters like operational damping, eigenfrequency and optimal pre-stress. Whereas in the secondary stage of control, a local controller acquires the incoming optimal pre-stress from the global master controller and provides the appropriate actuating signals to the sensing and actuation systems. Thus, the actual pre-stress measured by the sensing and actuation systems tracks the optimal pre-stress in the presence of external disturbances.

4.2. CONTROL THE LATHE – CASE STUDY

The joint interface module was incorporated in the tailstock of a lathe as shown in Figure 9. The joint interface module consists of three components, creating two controllable interfaces; the upper component (2) in which the quill mechanism is residing, the intermediate plate with a determined design (3) and the lower component (4) which is mounted on the machine body. Two stacks of piezoelectric actuators were mounted between clamping vices (5) on the sides of the tailstock (1) and as voltage is supplied, the piezoelectric elements are expanding, thus pulling together the positioning components which are bolted on the upper and lower vices of the JIM. Consequently, the two components of the JIM are pulled together, increasing the load on the two interfaces created by components 2, 3 and 4. In this case, 6 bolts are guaranteeing a minimum pre-load on the interface which amounts to approximately 120 KN.

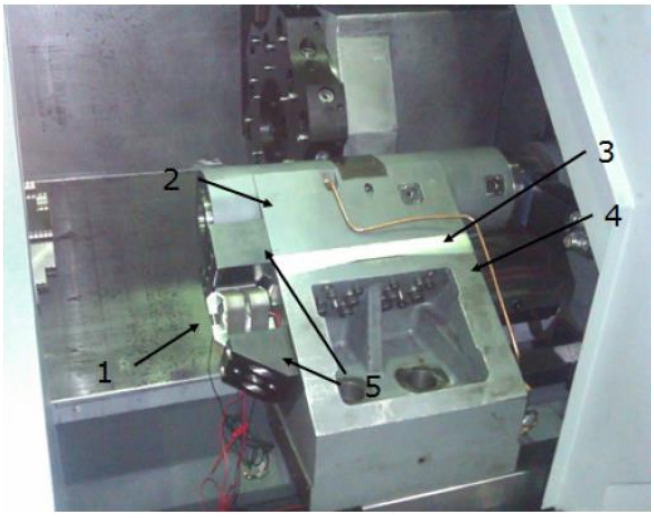


Fig. 9. The Joint Interface Module incorporated in the tailstock of the lathe

The machining tests were carried out in an AFM TAE-35 lathe. The machining experiments consisted of longitudinal turning processes, on a steel slender bar, clamped between the chuck and the tailstock of the lathe. Spindle speed was kept constant at n = 2000 RPM and feed at f = 0.2 mm/rev. The bars had a length of 350 mm and a starting

diameter of 32 mm. The system was tested between two extremes of load on the interface, as a result of supplying 0V to the actuators for the minimum pre-load configuration and 150 V for the maximum pre-load configuration. In the maximum pre-load, the actuators can deliver 17 KN of force [36]. This results to a range of pre-load on the interfaces between 120-137 KN.

During machining, at the high pre-load configuration, the first two passes (at diameters of 32 mm and 28 mm respectively) were stable, however the last pass, which would shape the bar to a final diameter of 24 mm, was unstable, with chatter starting to develop as the tool was moving towards the middle of the bar. For this reason the following discussion will only focus on the response of the system in the last pass.

In the beginning of the cut (region A, Figure 10), the process is stable, as close to the tailstock, the bending stiffness of the system is high while damping is low. As the tool approaches the middle of the bar and bending stiffness is decreasing, the process is becoming unstable (region B) and the chatter develops as the system lacks damping. Soon the process becomes definitely unstable as it reaches the middle of the bar where bending stiffness is minimized (region C). As the cutting tool travels towards the chuck in a higher bending stiffness area, chatter associated vibrations decrease (region D) and soon the process recovers again the stability (region E).

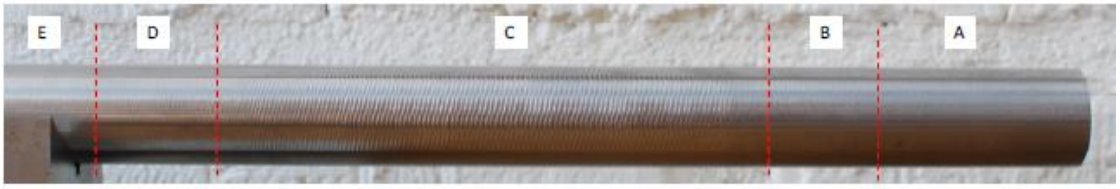


Fig. 10. Machined bar final diameter of 24 mm, high pre-load configuration. As the process was unstable, distinctive chatter marks appear on the surface

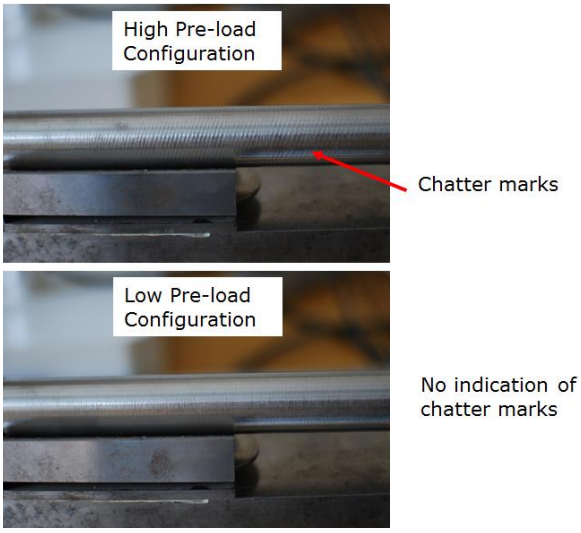


Fig. 11. Comparison of the surface of the bars machined in the high pre-load configuration (upper) and low pre-load configuration (lower)

In the case of the low pre-load configuration (when 0V was supplied to the actuators), at the same diameter, no chatter was manifesting during the process. Figure 11 illustrates a comparison of the bars after machining in both pre-load configurations. In a manner similar to the milling work holding joint, the reduction of the pressure on the interface led to an increase in damping. This damping increase consequently changed the way the system responds to machining excitations, in the way that a previously unstable process became stable. In the high pre-load configuration (upper) where damping is lower, the manifestation of chatter causes the distinctive chatter marks on the surface of the bar, which disappear in the low pre-load configuration (lower) as damping increases.

Figures 12a and 12b present a comparison of the sound signals acquired during machining in the time and frequency domains. The red curve presents the signal in the high pre-load configuration, where chatter is developing when the tool is approaching the middle of the bar (peak in the signal) and as it moves away from it, the phenomenon slowly fades with the process moving back into stability. The blue curve presents the signal in the low pre-load configuration, where the process was stable throughout the whole length of cut as a result of higher damping. The chatter frequency of 560 Hz coincides with the natural frequency of the bar which is around 550 Hz.

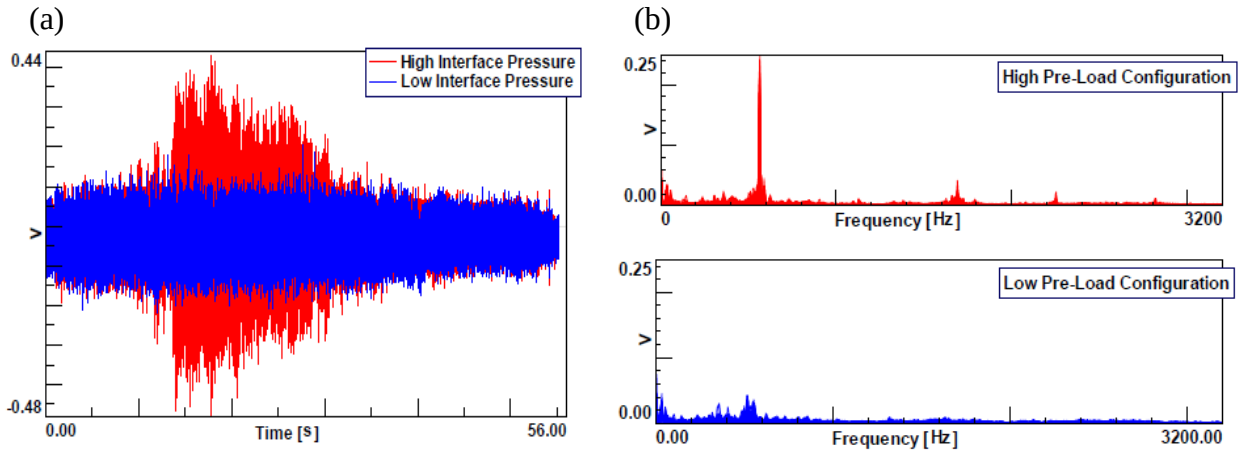
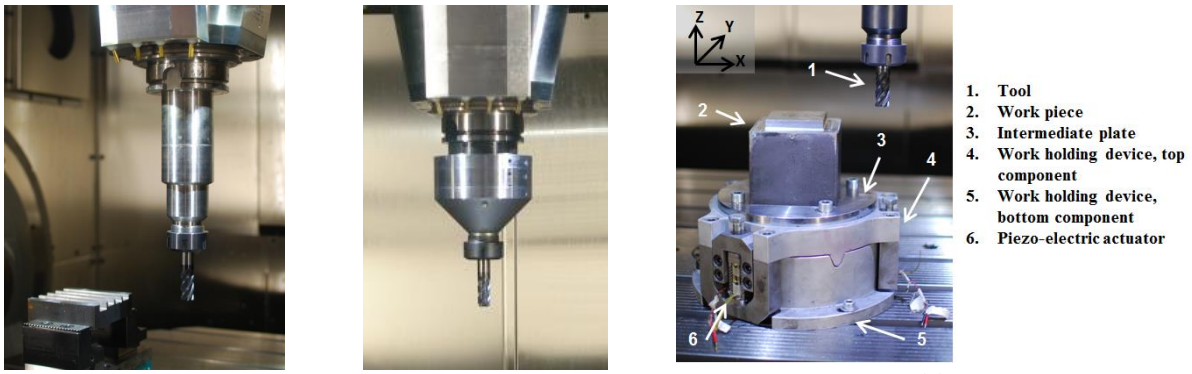


Fig. 12. Comparison of microphone signals acquired during machining with high preload (red) and low pre-load (blue) – time domain: The amplitude of the signal increases sharply as the process becomes unstable (a), The distinctive chatter frequency lies around 560 Hz (b)

4.3. CONTROL THE MILLING MACHINE - CASE STUDY

Three kinds of joint interface modules were incorporated in a HERMLE C50 milling machine; A passive tool holder (a), an active tool holder (b) and an active workholding device (c) as shown in Fig. 13. In all three prototypes, the internal interface geometry was designed in order to maintain static stiffness on the XY plane. The tool holding JIMs were designed on an HSK 100 mounting interface which allow them to be used in all spindles with such a coupling, while the workholding device can be mounted on the machine table using the T-slots.



(a) (b) (c)

Fig. 12. JIM prototypes for controlling the milling machine; (a) Passive JIM tool holder, (b) Active JIM tool holder, (c) Active JIM workholding device

(a) Passive JIM tool holder

On the passive tool holder (a), the friction damping mechanism on the internal interface was replaced with a viscoleastic damping mechanism via the application of viscoelastic polymer layers. These polymers transform strain energy from the vibrations occurring during the machining process into heat. The dimensions of this JIM are replicating an off the shelf tool holder in order to assess its performance in improving the response of the machining system to vibrations from the cutting process.

The tests consisted of slotting operations on a steel workiece at increasing axial depths of cut, with a 16 mm diameter solid carbide tool. Spindle speed was set at 3100 RPM, feed at 0.05 mm/rev and axial depth of cut increasing from 1 mm. The stability limit with the use of the reference tool holder was identified at 1.5 mm depth of cut while with the use of the JIM tool holder, at 4mm depth of cut the process was still stable as shown in Fig. 14 and Fig. 15. The chatter frequency when using the reference tool holder was identified at 880 Hz, originating from the first bending mode of the tool holder about the HSK coupling.

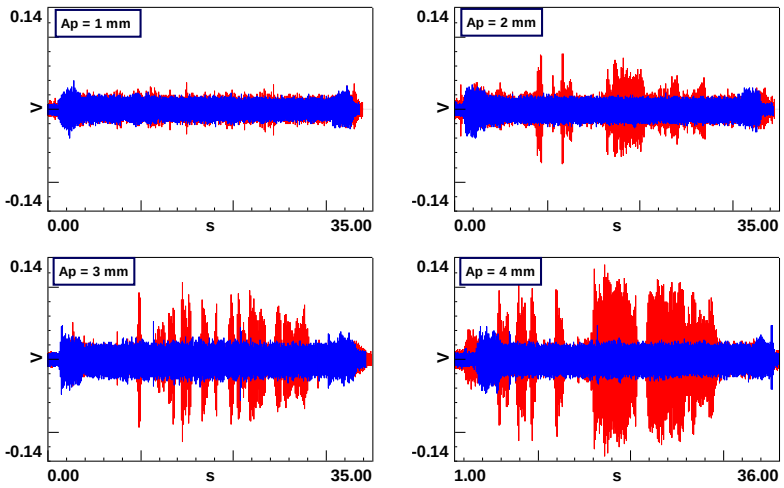


Fig. 13. Time domain microphone signal from the cutting processes. Red: Reference tool holder, Blue: JIM tool holder. Using the JIM tool holder allowed for machining in stable conditions with increased depth of cut as compared to the reference tool holder

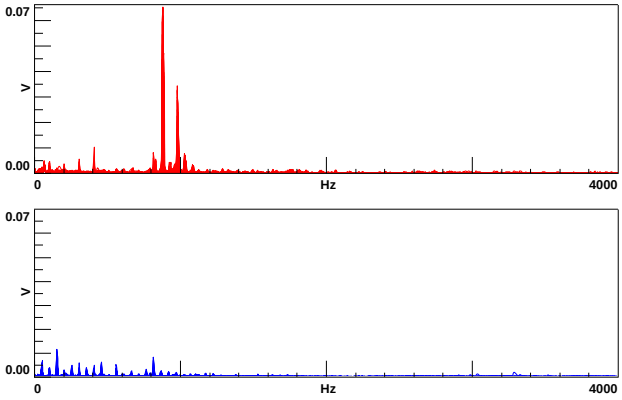


Fig. 14. Frequency domain microphone signal from the cutting processes. Red: Reference tool holder, Blue: JIM tool holder. The Chatter frequency resides at approximately 880 Hz

(b) Active JIM tool holder

The active tool holder (c) exploits the same principle described earlier for the lathe tailstock. The load on an internal interface can be adjusted by pneumatic means through a configuration of pistons inside the tool holder. By supplying air pressure through the coolant channel of the spindle, the load on the interface is increased, resulting in higher stiffness but lower damping due to the restriction of microslip on the targeted interface.

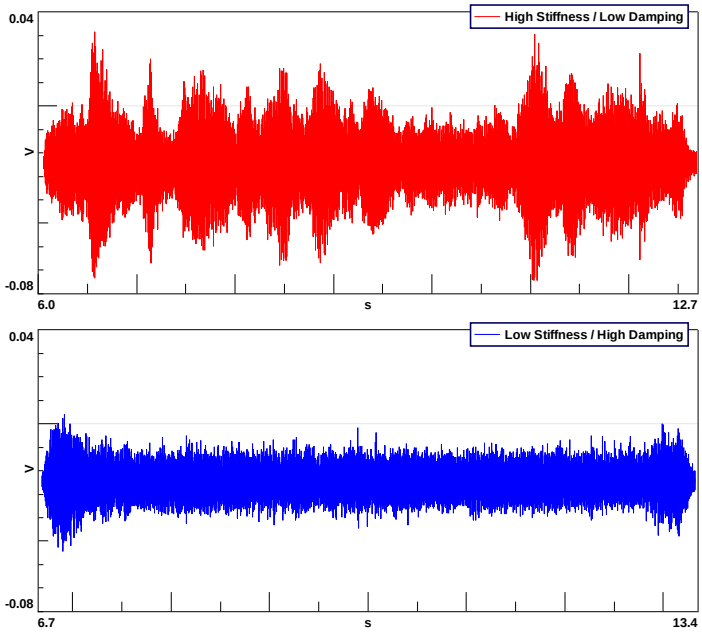


Fig. 15. Time domain microphone signal from the machining process with the JIM preset at the two air pressure configurations. Top: high stiffness/low damping, Bottom: low stiffness/high damping

Releasing the pressure in the piston chambers stiffness is lowered and the system gains damping. The air pressure in the tool holder is regulated by an actuated valve and the load

range on the internal interface is between and 7.5 KN (maintained by a bolt) and 12.5 KN when 6 bar air pressure is supplied. A microphone records the sound during the process and if it becomes unstable, the control system signals the valve to release the pressure, resulting in damping increase. The same 16 mm solid carbide tool was used and the same steel work material mentioned earlier. Cutting speed was 5000 RPM, feed was 0.01 mm/tooth and the depth of cut 2.5 mm.

Figure 16 presents the response of the system on a straight slot with the sytem preset at the two air pressure configurations. In the high stiffness / low damping configuration (high air pressure) the process is unstable, while at the low stiffness / high damping configuration (low air pressure) the process remains stable.

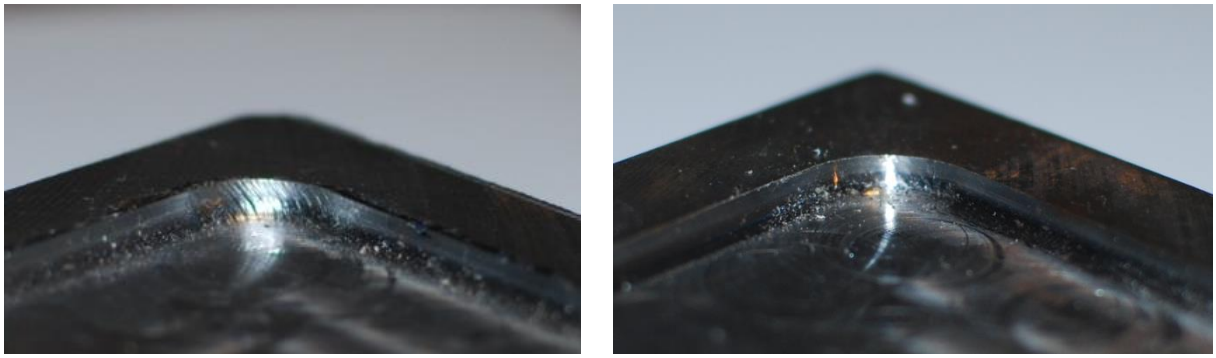


Fig. 16. The corner of the machined pocket with the tool holder at high stiffness/low damping configuration (left) and low stiffness/high damping configuration (right)

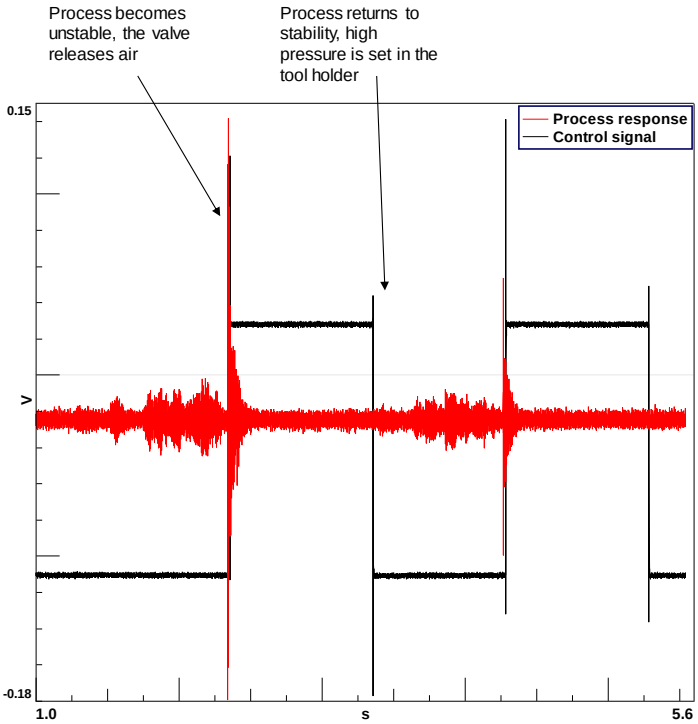


Fig. 17. Time domain signal of the process response and the control system voltage supplied to activate the valve

Figure 17 exhibits the resulting surfaces from machining a pocket, where a common problem of instability appears at the corners of the pocket. When the system is set to the low stiffness / high damping configuration, instability does not appear and the chatter marks are avoided.

Figure 18 presents the response of the system when the control system is active during a process where a pocket is machined. When the process becomes unstable, the control system activates the valve to release the pressure from the system. In this manner, the damping in the system increases and the process returns to stability. Such a method can prove beneficial regarding the dynamic stability of the system, in cases of depths of cut differentiating during the process, or when the stiffness of the workpiece is changing resulting to chatter.

(c) Active JIM workholding device

The workholding JIM was used to cross examine the effects of pre-load on the internal interface and the different treatments of the interface for damping enhancement. The machining conditions are the same as the ones used in the case of the passive JIM tool holder and the reference tool holder menstions earlier was used. The workholding JIM was tested in two configurations of pre load (3 KN and 13 KN) and three interface contact configurations:

1. Untreated metal to metal contact
2. Coating of the one side of the interface (damping mechanism remains friction) with high damping nanomaterial
3. Application of the viscoelastic polymer (friction damping is replaced by viscoelastic damping).

In this set of experiments, the stability limits for the different configurations are presented in Table 2

Table 2. Stability limits for the workholding JIM setup

Interface contact treatment	Low Preload	High Preload
Untreated	1.5 mm	1.5 mm
Nanocoating	2.5 mm	2 mm
Viscoelastic	2.5 mm	3 mm

Even though the chatter problem was originating from the tool holder as mentioned earlier, the integration of the JIM close to the process facilitated the increase of the stability limit when the damping in the internal interface was enhanced by using high damping materials. In the case of the coating, where the damping mechanism remains of friction, lower pre-load on the interface allowed for an increase of microslip and therefore damping, resulting in an increase of the stability limit. In the case of viscoelastic treatment, higher pre-load results in higher stiffness but is not detrimental to damping since the friction damping mechanism is replaced by viscoelastic damping. Hence, dynamic stiffness is increasing resulting in an increase in the stability limit.

5. CONCLUSIONS

Dynamics of machining systems is a field of major importance in manufacturing research and of outmost industrial relevance. Static and dynamic capability of the machining system is determined by the interaction between machine tool and cutting process. In this respect, the control of machining system stability can be achieved by changing dynamic parameters of the machine tool, the process or both. The traditional way to control machining system dynamic behaviour is by controlling the process parameters, i.e., depth of cut, rotational speed, speed rate as well cutting tools' micro-geometry and material. By this, static stiffness in the direction of cutting force and overall damping of machining system are improved.

Although being aware about the significant contribution of joints' stiffness and damping to the overall capability of the machining systems, the classical theory of the machining system lacks a unified concept for consciously designing structural interfaces with controllable characteristics. Apparently, there is a tendency today both among scholars and manufacturers to develop and implement complex schemes for on-line monitoring and control of machining systems. The simple explanation is the existing knowledge gap between machine tool manufacturers and machine tool users regarding static and dynamic capability. Due to unlimited combinations of tools' geometries and materials, workpieces' shapes, dimensions, and materials, fixtures and toolholders it is impossible to predict the behaviour of a machine tool and by this the accuracy of resulted parts. The consequence is that the machine tool users are forced to add advanced sensor systems for monitoring and controlling the machining system. In an industrial environment these solutions are costly and complex and not reliable due to adverse conditions. The concept presented has proven the fundamental relationship between the conditions of machine tool joint interfaces and the process static and dynamic stability.

By controlling a machining system (MS) capability the entire production system can be improved by decreasing lead times through better allocation of resources and smoother flow of material through the production line. In a modern manufacturing system, it is important to efficiently and accurately assess and control the MS capability. Controlling MS capability means to react to unexpected conditions such as machine breakdown or a bad component part, and taking actions to rectify the situation. The performance of production systems and ultimately the environmental impact of the material and energy utilization impact on environment is determined by the capability of each machining system units that compose a production system.

REFERENCES

[1] NICOLESCU C.M., 1996, *On-line Identification and Control of Dynamic Characteristics of Slender Workpieces in Turning*, Journal of Materials Processing Technology, 58/4, 374–378.

[2] FRANGOUDIS C., 2014, *Controlling the Dynamic Characteristics of Machining Systems through Consciously Designed Joint Interfaces*, Licentiate Thesis, KTH Royal Institute of Technology. Department of Production Engineering, Stockholm.

[3] KURATA Y., MERDOL S.D., ALTINTAS Y., SUZUKI N., SHAMOTO E., 2010, *Chatter Stability in Turning and Milling with Process Identified Process Damping*, Journal of Advanced Mechanical Design, Systems and Manufacturing, 4/6.
[4] BUDAK E., TUNC L.T., ALAN S., ÖZGÜVEN H.N., 2012, *Prediction of Workpiece Dynamics and its Effects on Chatter Stability in Milling*, CIRP Annals - Manufacturing Technology, 61/1, 339−342.
[5] BUDAK E., COMAK A., OZTURK E., 2013, *Stability and High Performance Machining Conditions in Simultaneous Milling*, CIRP Annals - Manufacturing Technology, 62/1, 403−406.
[6] TLUSTY J., 1978, *Analysis of the State of Research in Cutting Dynamics*, CIRP Annals, 27/2.
[7] TLUSTY J., ZATON W., ISMAIL F., 1983, *Stability Lobes in Milling*, Annals of the CIRP, 1, 32/1, 309−313.
[8] WECK M., VERHAAG E., GATHIER M., 1975, *Adaptive Control for Face Milling Operations with Strategies for Avoiding Chatter Vibrations and Automatic Cut Distribution*, Annals of the CIRP, 24/1, 405−409.
[9] ALTINTAS Y., WECK M., 2004, *Chatter Stability of Metal Cutting and Grinding, CIRP Annals - Manufacturing Technology*, 53/2, 619−642.
[10] ITO Y., 2013, *Position Paper – Fundamental Issues in Self-Excited Chatter in Turning*, Journal of Machine Engineering, 13/3, 7−25.
[11] POWAŁKA B., JEMIELNIAK K., 2015, *Stability Analysis in Milling of Flexible Parts based on Operational Modal Analysis*, CIRP Journal of Manufacturing Science and Technology, 9, 125−135.
[12] ZAGHBANI I., SONGMENE Y., 2009, *Estimation of Machine-Tool Dynamic Parameters during Machining Operation through Operational Modal Analysis*, International Journal of Machine Tools & Manufacture, 49, 947–957.
[13] AKAY M., 1994, *The Autoregressive Moving Average (ARMA) Method*, Biomedical Signal Processing, 252−280.
[14] POZNYAK A.S., 2009, *Parametric Identification*, Advanced Mathematical Tools for Automatic Control Engineers, Stochastic Techniques, 357−416.
[15] CADZOW A., 1987, *Spectral Analysis*, Handbook of Digital Signal Processing, 701−740.
[16] NEUMAN D., 1992, *Fault Diagnosis of Machine-Tools by Estimation of Signal Spectra*, Fault Detection, Supervision and Safety for Technical Processes, 147−152.
[17] CHEN H.F., 2009, *Recursive System Identification*, Acta Mathematica Scientia, 29/3, May, 650−672.
[18] LE T.P., PAULTRE P., 2013, *Modal Identification Based on the Time–Frequency Domain Decomposition of Unknown-Input Dynamic Tests*, International Journal of Mechanical Sciences, 71, 41–50.
[19] ARCHENTI A., 2008, *Model-Based Investigation of Machining Systems Characteristics*, Licentiate Thesis, KTH Royal Institute of Technology.
[20] PRIESTLEY M.B., 1988, *Current Developments in Time Series Modelling*, Journal of Econometrics, 37/1, 67−86.
[21] LJUNG L., 1981, *Analysis of a General Recursive Prediction Error Identification Algorithm*, Automatica, 17/1.
[22] STOICA P., SŐDERSTRŐM T., 1981, *Analysis of an output error identification Algorithm*, Automatica, 17/6, 861−863.
[23] ARCHENTI A., DAGHINI L., NICOLESCU C.M., 2010, *Recursive Estimation of Machine Tool Structure Dynamic Properties*, CIRP International Conference on High Performance Cutting, / [ed] Tojiro Aoyama, Yoshimi Takeuchi, Gifu, 365−370.
[24] ARCHENTI A., NICOLESCU C.M., 2010, *Recursive Estimation of Operational Dynamic Parameters in Milling Using Acoustic Signal*, International Conference on Process Machine Interactions / [ed] Y. Altintas, Vancouver.
[25] ARCHENTI A., 2011, *A Computational Framework for Control of Machining System Capability: From Formulation to Implementation*, Doktorsavhandling, KTH, Skolan för industriell teknik och management (ITM), Industriell produktion, Maskin- och processteknologi.
[26] NICOLESCU C.M., ARCHENTI A., 2013, *Dynamic Parameter Identification in Nonlinear Machining Systems*, Journal of Machine Engineering, 13/3, 91−116.
[27] TUNC L.T., BUDAK E., 2012, *Effect of Cutting Conditions and Tool Geometry on Process Damping in Machining*, International Journal of Machine Tools and Manufacture, 57, 10−19.
[28] BRECHER C., BAUMLER S., BROCKMANN B., 2013, *Avoiding Chatter by means of Active Damping Systems for Machine Tools*, Journal of Machine Engineering, 13/3, 117−128.
[29] JEMIELNIAK K., WIDOTA A., 1984, *Suppression of Self Excited Vibration by the Spindle Speed Variation Method*, Int. J. Machine Tools and Manufacture, 23/3, 207−214.
[30] ITO Y., 2008, *Modular Design for Machine Tools*, McGraw-Hill.
[31] SMITH S., TLUSTY J., 1992, *Stabilizing Chatter by Automatic Spindle Speed Regulation*, Annals of the CIRP, 41/1, 433−436.
[32] RIVIN E.I., 1999, *Stiffness and Damping in Mechanical Design*, CRC Press.
[33] RIVIN E.I., 2001, *Interrelation of Stiffness and Damping in Machine Tool Dynamics*, in Transactions of the North American Manufacturing Research Institution of SME, 137−143.

[34] SAKAI Y., TSUTSUMI M., 2013, *Dynamic Characteristics of Linear Rolling Bearings for Machine Tools*, Journal of Machine Engineering, 13/3, 129−142.
[35] JEDRZEJEWSKI J., KWASNY W., J. MILEJSKI D., SZAFARCZYK M., 1985, *Selected Diagnostic Methods for Machine Tools Acceptance Tests*, CIRP Annals - Manufacturing Technology, 34/1, 343−346.
[36] KRAS A., BOURGAIN F., CLAEYSSEN F., 2014, *Amplified Piezo Actuator APA with Viscoelastic Material for Machine Tool Semi-Active Damping System*, Journal of Machine Engineering 14/3, 83–96.