VIRTUAL MACHINING SYSTEM ENGINE FOR SIMULATION OF THE PROCESS MACHINE INTERACTION

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The aim of this paper is to introduce a novel methodology, based on a finite element (FE) computation engine for simulation of process machine interaction occurring in machining systems. FE modelling of the milling process has the purpose of being accountable for a thorough validation of the parametric identification approach, and of providing a good physical insight into the phenomena investigated. The system considered here has a lower number of degree-of-freedoms which permits a thorough analysis. However, when taking into account the system's nonlinear and time-varying nature, it is apparent that the results are far from being trivial. Therefore, the analysis of the milling process, taking into account nonlinearities restricting the growth of response amplitudes in the case of chatter-type instability, provides some intrinsic information of the basic features on the system that might be of both fundamental interest and practical use.

Keywords
FEM, milling, simulation, machine tool, cutting process

1. Introduction

In intermittent machining operations, the time-varying and discontinuous nature of the cutting process represents a challenge in terms of cutting parameter selection, control and optimization [Brecher, 2009]. It is difficult to theoretically analyse the dynamic stability of machining systems due to the lack of an explicit scheme for representing both physical processes and structural systems in an integrated manner under a common test framework. Many of the experimental methods used to analyse, control and optimize machining systems are based on off-line procedures or test environments that do not replicate the actual machining operation [Tlusty, 1983]. For instance, modal parameters of the machine tool elastic structure can be extracted by using common experimental modal-analysis techniques, but such techniques are formulated and are valid for stationary mechanical systems (with constant parameters) [Altintas, 2004]. The knowledge of modal parameters is not enough for process monitoring and control of real machining operations. It is necessary to estimate the overall damping of the machining system in order to compute the stability charts.

The aim of this paper is to introduce a novel numerical method, based on a finite element (FE) computation engine for simulation and validation of the real-time identification schemes applied in machining. In section 2, a framework for dynamic stability analysis is introduced and in section 3 a virtual machining system engine (VMSE) based on an integrated FE computation is presented. In section 4 the engine is used to conduct virtual stability analysis of PMI. Finally, discussions and conclusions are outlined in section 5.

2. A framework for dynamic stability analysis

The stability of machining system is highly operational dependent. For instance, damping can change during machining due to variations in cutting conditions, workpiece geometry and contact condition of machine tool structure subsystems (e.g. ball bearing in spindle units). As the total system damping approaches zero level, the system is on the point to lose its stability. Therefore the static stiffness and damping of the system are important characteristics for controlling stability. As mentioned earlier, the main reason for the problem in studying dynamic stability of machining systems is the lack of an explicit scheme for representing both physical processes and structural systems in an integrated manner under a common test framework. To deal with this problem, a framework for dynamic stability analysis has been formulated. The framework consists of two parts; an operational machining system and a VMSE respectively (see Fig. 1).

2.1 Operational machining system

The operational machining system is used to track machining system operational dynamic parameters (ODP, i.e. operational damping ratio ξ_{op} and operational frequency f_{op}) to account for the interaction between the cutting process and the machine tool elastic structure, the PMI, during real cutting operations [Archenti, 2009]. Operational damping ratio, ξ_{op}, is the overall damping of the machine tool structure (modal damping) and the dynamic cutting process (process damping). The computation of ODP is implemented on-line through a recursive scheme to monitor early changes in PMI damping in order to control chatter. The parameters in the recursive autoregressive moving-average (RARMA) model are updated in real-time.

The key concept of the identification procedure in this paper is to find a feature of the measured random response that can be used to discriminate between stable and unstable PMI. Random signals cannot be characterized by simple, well-defined mathematical equation and their future values cannot be predicted. Probability and statistics have to be used to analyze their behaviour [Ljung, 2006], [Mohanty, 1985], [Dimentberg, 1983].

The discrimination between stable and unstable PMI is treated in the framework for recursive model-based identification by [Archenti, 2011]:

- formulation of a qualitative/semi-qualitative mathematical model of the machining system response for subsequent quantitative analysis;
- estimating the system's ODP;
- implementing a proper control design for chatter suppression.

The term 'qualitative' implies that the model is based on a statistical analysis of measured system response [Tarantola, 2005]. Although the model-based identification approach presented here leads to the estimation of key dynamic parameters (hereby, the term 'semi-qualitative'), they are solely valid within a certain confidence interval.

The RARMA parameter estimation as well as the computation of ODP are performed recursively at each time t (t = 1,2,…,n). The RARMA method provides a robust tool for discriminating between forced and self-excited oscillations by tracking the time-varying dynamics to be implemented in schemes for real-time identification and control. The real-time identification is based on the recursive prediction error method (RPEM) [Ljung, 2006], [Söderström, 2001] of regression model parameters and on the extraction of ODP from characteristic polynomials of a discrete model [Archenti, 2011].

2.2 Virtual machining system engine

One limitation with recursive identification is that the decision about the model structure and the model order has to be made a priori to start the identification. To improve the accuracy of RARMA, a VMSE has been developed.

The VMSE, based on FE computation, represents the machining system (including both the cutting process dynamics as well as machine tool structure dynamics) and generates a response of known damping ratios and frequencies which are estimated by an offline RARMA algorithm. By comparing the theoretical (true) values from the VMSE to those produced by the RARMA the model structure and its order can be optimized.

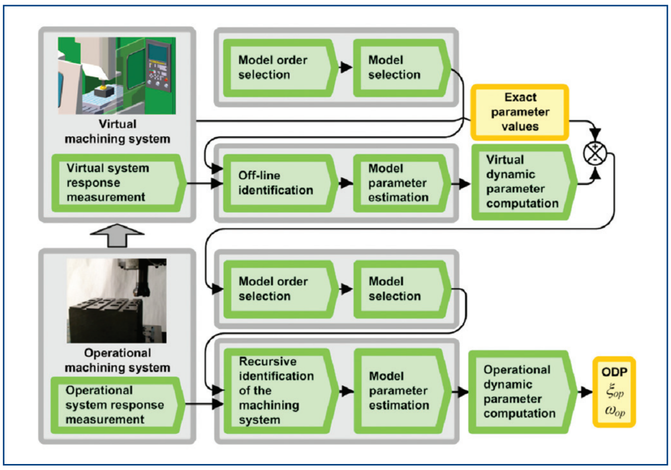

***Figure 1.** Computational framework for dynamic stability analysis*

3. Virtual machining system engine

The term ´engine´ implies that the introduced virtual machining system is capable of emulating the physical interaction that occurs in a real machining system under normal operation. The two subsystems, machine tool elastic structure and cutting process dynamics, are represented by physical relations and their mutual interaction is described by physical processes.

The FE-modeling of the intermittent cutting process has the purpose of being accountable for [Archenti, 2011]:

- Thorough modal parameter estimation (mass, stiffness, damping) of the elastic structure.
- Parametric identification of the dynamic response resulting from the interaction between the elastic structure and the cutting process.
- Use the above results to optimize the model structure in recursive identification of operational machining systems as illustrated in Fig. 1.
- To provide a good physical insight into the phenomena investigated.

3.1 Architecture

The relative motion between the tool and workpiece represents the sum of all individual relative displacements in the machine tool elastic structure and those generated during cutting process. Thus, if the purpose is to study the stability of the machining system and not to identify individual sources of excitation then it is sufficient to project some (those lying in the frequency interval of interest) individual motions at interface tool – workpiece. In a more general case the dynamic coupling between tool and workpiece can be represented by a number of degrees of freedom. The modal parameters for each DOF which represents the input to the VMSE are identified from real machining systems by the methods described in [Archenti, 2011].

In our study we restrict it to 3 DOF which in any case do not influence the accuracy of the study. In other cases, when the analysis of the system aims for the identification of vibration sources, an adequate representation can be achieved by a full 3D CAD model.

3.2 Machine tool structure dynamics

The machine tool structural system is represented by two orthogonal bending modes and one torsion mode, see Fig. 2 (module 1). The X-Y coordinate system is fixed with respect to the machine tool structure, and its axes are aligned with the principal modes of oscillation. As will be described later in this section, another coordinate system, tempo-spatial x-y is used to represent the delay process and is in temporal and spatial motion with respect to the reference system.

Structural damping is introduced in the system as Rayleigh type of damping; the damping ratio, ξ is calculated at the natural angular frequency ω according to

$$\xi = \frac{1}{2}\left(\frac{\alpha_M}{\omega} + \beta_K \omega\right) \tag{1}$$

where α_M and β_K are the mass, stiffness and damping parameters, respectively.

3.3 Cutting zone physics

To represent the cutting process, five dynamic force types are generated at each cutting edge:

1. A cutting force component proportional to the axial depth of cut and feed load in tangential F_{ti} and radial F_{ri} directions, projected on the X-Y coordinate system (Fig. 2).

$$F_{ti} = K_t a_p h_i(\varphi_i) \tag{2}$$

$$F_{ri} = K_r a_p h_i(\varphi_i) \tag{3}$$

where K_t, K_r are the tangential and radial cutting coefficients, respectively, ap is the depth of cut and $h_i(\varphi_i)$ is the chip thickness variation with respect to the radial engagement angle φ_i corresponding to the ith tooth. These force components are then projected in the X-Y reference coordinate system;

2. Damping forces in x direction F_{Dx} and y direction F_{Dy}, respectively, representing the process contribution to the overall machining system damping (Figure 2)

$$F_{Dx} = \frac{K_{Dx} 2\cdot\pi}{\omega}\cdot\frac{dx}{dt} \tag{4}$$

$$F_{Dy} = \frac{K_{Dy} 2\cdot\pi}{\omega}\cdot\frac{dy}{dt} \tag{5}$$

where K_{Dx} and K_{Dy} are damping coefficients in X and Y direction, respectively, ω is the angular rotational speed, and dx/dt and dy/dt are time derivatives of space coordinates;

3. A regenerative force component as a function of chip thickness variation. For radial immersion angles less than $2\pi/z$ where z is the number of teeth simultaneously in engagement, the regenerative effect is modulated once every revolution, while otherwise between consecutive teeth;
4. A random, normal-distributed $E(x, \sigma^2)$ component is added to account for variations due to changes in contact area at the chip-tool interface, tool wear, non-homogeneity of work material and deflection of the workpiece/fixture. These force components are in effect only during tool engagement arc of cut;
5. An acceleration force, important for higher cutting speeds, is added as a body load to the model.

These dynamic forces are only in effect between the entry and exit angles.

The maximum cutting force is estimated from the stress distribution, calculated from a thermo-mechanical module 2 in (Fig. 2) run in parallel with the main computational algorithm (module 1 in Fig. 2). In stress calculation, the chip-thickness at the first full contact with the cutting edge is considered. Therefore, for the force calculation,

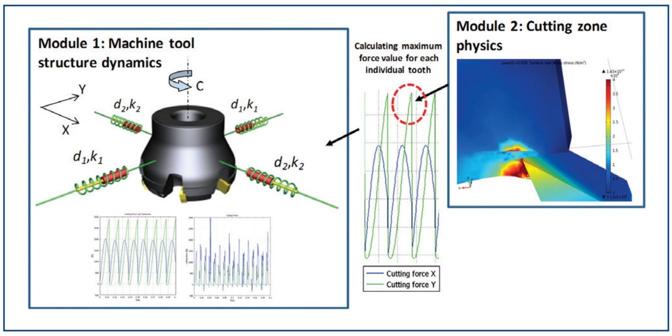

Figure 2. *The machine tool structural system is represented by two orthogonal bending modes (in X and Y directions) and one torsion mode C in module 1 of the VMSE. The cutting zone physics module is generating the maximum force values based on stress. [Archenti, 2011]*

the contact length and the roundness of the cutting edge are considered. This makes it possible to analyse the machining system with respect to tool geometry.

3.4 Regenerative effects

For representing the regenerative effect that is characteristic for unstable machining, two different computational techniques have been employed. For regenerative effect between two consecutive teeth an "extrusion" technique was implemented. The oscillations along x and y, respectively, calculated on the cutting edge of *i*-1 tooth are projected to the *i* tooth and subtracted from the corresponding oscillation. For simulation of regenerative effect at every revolution the arbitrary Lagrangian-Eulerian (ALE) method has been used.

The ALE method is an intermediate between the Lagrangian and Eulerian methods, and it combines the best features of both – it allows moving boundaries without the need for the remeshing to follow the material deformation. A deformed mesh can be useful if the boundaries of the computational domain are moving in time. In this case a new mesh needs not to be generated for each configuration of the boundaries; instead the software simply perturbs the mesh nodes so they conform to the moved boundaries. At the first revolution the mesh movements follows the structural deformation, while at the next revolutions the deformation are subtracted from form those corresponding to the deformed mesh. The chip-thickness variation is in effect while it has a positive value (reproducing the nonlinear effect).

4. Virtual stability analysis

The VMSE has been used to simulate milling operations with various cutting condition. As a preliminary step in the FE model building, the

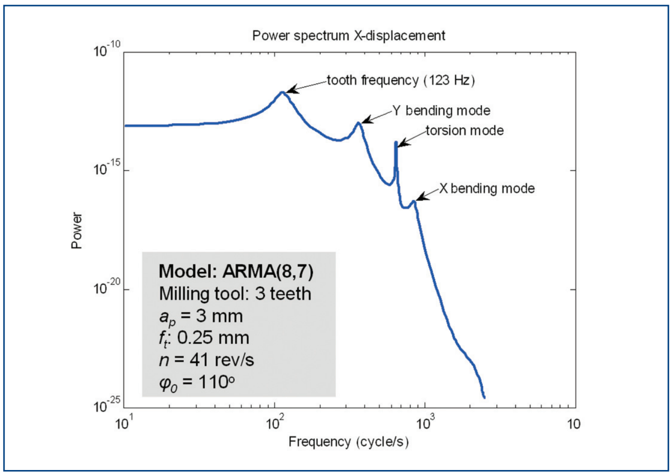

Figure 3. *Power spectrum of the response of a stable PMI identified by an ARMA(8,7) model*

natural frequencies of the structure have been computed as follows: a bending mode Y direction (452 Hz); a torsion mode C in X-Y plane (639 Hz); and a bending mode X direction (980 Hz).

Though the FE simulation and model-based identification are performed in the time domain, many of the results are presented in the frequency domain as power spectra. In stable machining for a three-tooth milling cutter engaged in arc of cut, 25°<φ_0<135°, the dominant frequency is the tool passing frequency. The power spectrum of the response of a stable PMI identified by an ARMA (8, 7) model and the identified structural modes are shown in Fig. 3.

At very low engagement arcs of cut, the cutting force resembles an impulse with energy distributed over a broad frequency range. Figure 4 illustrates the power spectrum of the simulated response of the PMI when tooth engages over a short arc of cut. The impulse-like cutting force excites a broadband frequency range. The torsion mode at 639 Hz dominates the power spectrum of the response in X and Y respectively.

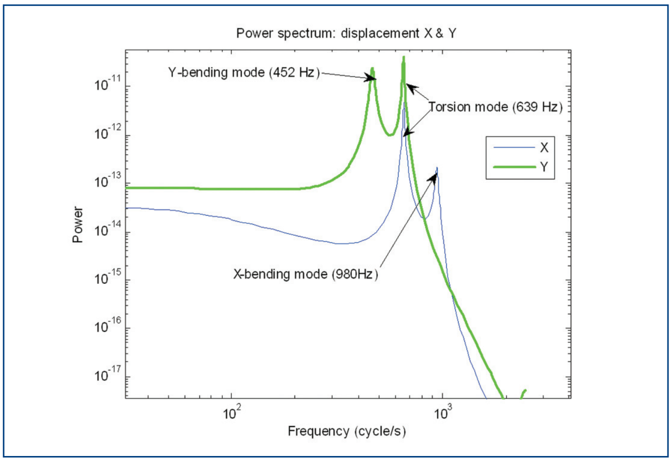

Figure 4. *Power spectrum, identified with ARMA(6,5) model, of the simulated response of the PMI when tooth engages over a short arc of cut*

Tab. 1 presents the identification results for the same cutting parameters as above.

Simulated		Parametric identification		e_r	
f [Hz]	ξ [%]	f [Hz]	ξ [%]	f [%]	ξ [%]
452	2	465	2.74	2.9	37
639	2	639	2.18	–	9
980	2	991	2.15	1.1	7.5

Table 1. *Identification of the system response for short engagement arc of cut (φ_0 = 20°). Frequency f, damping ratio ξ and the corresponding errors e_r*

4.1 Resonance and regenerative vibrations

At the tooth passing frequencies close to the natural frequencies of the elastic structure, resonance occurs.

In the example in Fig. 5a the spindle speed is 326 rev/sec so that the tooth passing frequency is close to the 980 Hz bending mode. The damping is only insignificantly changed, thus correctly representing a stable system. Meantime, the amplitude increases by two orders of magnitude. The time domain vibration in X direction can be seen in Fig. 5b. The vibration pattern for the stable machining operation that undergoes resonant vibrations has quite evenly distributed amplitude.

The response of an unstable PMI when the tool is in full immersion (φ_0 = 180°) is shown in Fig. 6a. The regenerative effect is produced by the chip thickness variation between consecutive teeth and between tool revolutions. The estimated damping ratio identified in an ARMA (4, 3) model drops to a very low level (0.07 %), correctly identifying an unstable process. From Fig. 6b, the nonlinear effect due to the loss of contact between tool and work is apparent.

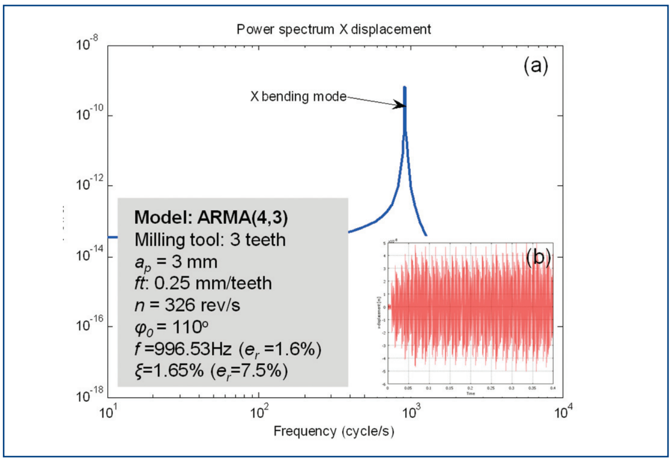

Figure 5. *(a) Power spectrum of a ARMA (4, 3) at tooth passing frequency close to 980 Hz. (b) time domain stable vibration in X direction*

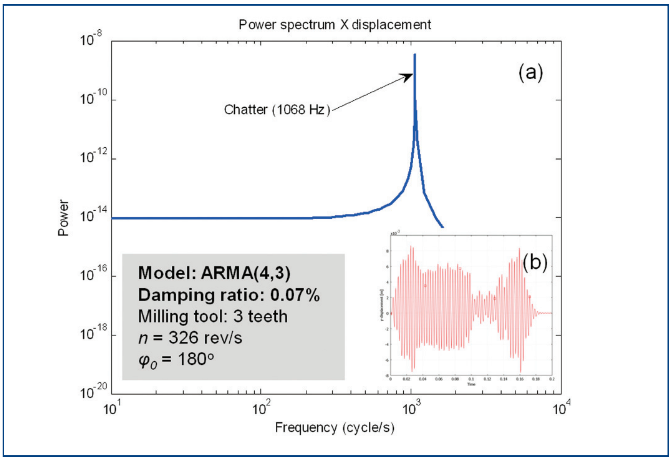

Figure 6. *(a) Parametric identification: power spectrum for unstable machining system, (b) time domain chatter as a regenerative phenomenon between consecutive teeth, vibration in X direction*

4.2 Structural and process damping

One of the main reasons for developing an FE model for milling was to investigate the accuracy of real-time parametric identification. In the following example the structure have been computed as follows: a bending mode in Y direction (327 Hz), a torsion mode C in X-Y plane (411 Hz), and a bending mode in X direction (671 Hz).

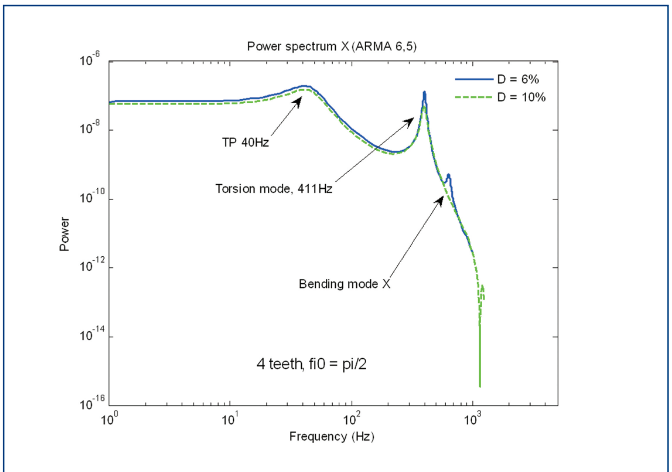

Figure 7. *Parametric identification: power spectrum of a ARMA(6,5) model of the displacement signal in X direction*

Two processes subject to intermittent cutting force were generated, with a structural damping of 6 % and 10 %, respectively. Fig. 7 shows the power spectra in X direction while Tab. 2 presents a comparison between simulated dynamic parameters and those identified in ARMA models. The tool passing frequency (TP) and torsion mode are correct identified. The estimation of the natural frequencies appears to be unbiased and have relatively little uncertainty. The variability in damping estimates is greater than that for frequency estimates.

	Simulated		Parametric identification		e_r	
Bending mode	f [Hz]	ξ [%]	f [Hz]	ξ [%]	f [%]	ξ [%]
Y	325.40	6	331.13	5.35	1.8	12.1
	323.20	10	326.84	9.89	1.1	1.1
X	655.60	6	649.20	5.51	1.0	18.1
	648.30	10	638.75	9.80	1.5	1.9

Table 2. *Virtual PMI's response in X and Y direction. Comparison between simulated and identified modal parameters. Frequency f, damping ratio ξ and the corresponding errors e_r*

When added to the structural damping, the process damping contributes by additionally enhancing the overall damping. In a simulation case, a process damping of 8 % is added to the original structural damping of 2 % to obtain a total damping of 10 % (see Fig. 8).

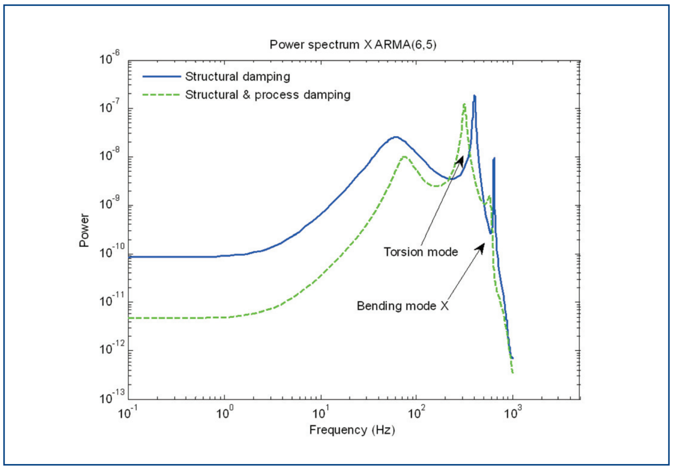

Figure 8. *Parametric identification: power spectrum of ARMA(6,5) identification of the structural and total damping*

The ARMA (6,5) identification process estimates the original damping at 1.7% (15 % error), while the total damping is estimated at 8.31 % (16.9 % error). These results are presented in Fig. 8 as power spectra of estimated models.

5. Discussion and conclusions

Dynamic instability cannot be solely characterized by a quantitative criterion in terms of the amplitude levels of the measured response signal; the qualitative pattern in the system's response itself determines the occurrence of instability.

The FE simulation results demonstrated, as illustrated for instance by Fig. 5, that in case of impulse-like cutting force excitation, the broad frequency spectrum can excite the natural modes and thus leading to resonance. The system is still stable (as revealed by the damping ratio value) but the amplitudes are considerably increasing. The same remark is valid for tool passing frequencies close to one of system natural frequencies. The behaviour is representing a stable system with the amplitude of vibration naturally increasing due to resonance phenomenon. This demonstrates that the damping is a robust criterion for discrimination between forced and self-excited vibration.

The main contributions of the performed research work are:

1. a model-based identification framework using a single step estimation of dynamic parameters characterizing the PMI as presented in Fig. 1;
2. a virtual representation of the PMI in a unique FE machining system engine for the purpose of generating simulated data to be used for validation of ODP.

Acknowledgement
This work is funded by VINNOVA (The Swedish Governmental Agency for Innovation Systems) and KTH DMMS (Design and Management of Manufacturing Systems) and has been supported by XPRES (Initiative for excellence in production research).

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