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Alternative experimental methods for machine tool dynamics identification: A review

Alexander Iglesias [a,*], L. Taner Tunç [b], Orkun Özsahin [c], Oier Franco [d], Jokin Munoa [a,e], Erhan Budak [f]

[a] *Ideko, Dynamics & Control, Elgoibar, Basque Country, Spain*
[b] *Integrated Manufacturing Technologies Research and Application Centre, Sabanci University, 34956 Istanbul, Turkey*
[c] *Middle East Technical University, Ankara, Turkey*
[d] *Precision Controls Laboratory, University of Waterloo, Waterloo, ON, Canada*
[e] *Mechanical Engineering Department, University of Basque Country (UPV-EHU), Bilbao, Spain*
[f] *Manufacturing Research Laboratory, Sabanci University, 34956 Istanbul, Turkey*

A R T I C L E I N F O

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A B S T R A C T

An accurate machine dynamic characterization is essential to properly describe the dynamic response of the machine or predict its cutting stability. However, it has been demonstrated that current conventional dynamic characterization methods are often not reliable enough to be used as valuable input data. For this reason, alternative experimental methods to conventional dynamic characterization methods have been developed to increase the quality of the obtained data. These methods consider additional effects which influence the dynamic behavior of the machine and cannot be captured by standard methods. In this work, a review of the different machine tool dynamic identification methods is done, remarking the advantages and drawbacks of each method.

1. Introduction

The productivity and quality of the produced parts strongly rely on the dynamic response of the machining system, and its thorough knowledge helps foreseeing vibration related problems to improve machine design [1] and achieve an optimum machining performance [2].

Both machine and workholding design can be optimized from a dynamic perspective [1,3,4]. The critical modes limiting the machine behavior can be analyzed by means of a modal analysis and, thus, the weaker points can be identified. With this information, a proper machining system redesign can be carried out, with the possibility to compare and select the most appropriate component (guideways, bearings, supports…).

The dynamics involved in the cutting process can be related to structural modes of the machine tool, natural frequencies of the spindle or the tool, and the dynamics of the part and its workholding system. The structural modes of the machine tool usually lie on the low frequency range (<200 Hz), whereas the natural frequencies of the spindle or the tool can be located in the range from around 100 Hz to thousands of Hertz [5,6]. The part and workholder can present different frequency ranges depending on its size, material and

* Corresponding author.
E-mail addresses: aiglesias@ideko.es (A. Iglesias), ttunc@sabanciuniv.edu (L. Taner Tunç), ozsahin@metu.edu.tr (O. Özsahin), ofrancod@uwaterloo.ca (O. Franco), jmunoa@ideko.es (J. Munoa), ebudak@sabanciuniv.edu (E. Budak).

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shape.

One of the possible problems attributed to machine dynamics is the resonance produced when the harmonics of a periodic force, for instance milling force, match the natural frequencies of the machining system. This is usually not a problem at low frequency machine tool structural vibration modes, due to their relatively high damping. In practice, the final shape of a part is obtained combining roughing and finishing operations. In roughing, the magnitude of the dynamic force can be high, but the required surface quality is not demanding, and therefore the usual damping of structural modes is enough. The finishing operations are more demanding, but the low force levels involved help to ensure the specifications. In addition, small variations of the cutting speed can avoid resonances in an effective way [7]. However, some finishing processes can be seriously affected by the forced excitation due to the harmonics of cutting forces. This is the case in grinding [8], where strict tolerances are required, or thin wall machining [9], where the absence of joints involved in the structure results in low damped modes.

Chatter vibrations are a more severe problem to avoid during the cutting process. This self-excited vibration emerges when the dynamic compliance of the machining system is not enough to perform a certain operation [1]. The onset of chatter on machine tools is pernicious, since it prevents obtaining the required surface finish, decreases the life of tools and mechanical components and, thus, it is still one of the major limitations for productivity [5]. Regenerative effect [10] is the main mechanism originating chatter in cutting processes, whereas the possibility of mode coupling chatter [11] presence is a controversial issue. Although it has been described in certain robotic milling applications [11], it has been recently discarded for milling processes by other authors [12]. Regenerative effect can be predicted by means of the stability lobe chart [13], where each frequency mode ranging from around 20 Hz to thousands of Hertz, depending on the element causing the vibration, creates a lobe “family” limiting the process stability.

Lastly, the motion command of the moving axes constitutes a major source of excitation and can limit the achievable movements of the machine tool. Even though the generated jerk trajectory is limited by the CNC, where the jerk is the derivative of the acceleration and determines the acceleration capabilities of the drive, it can be similar to an impact inducing a transient vibration and exciting the low frequency range [14]:

$$\delta \cong 4000 \frac{J}{f_d^3} \tag{1}$$

where,

δ: vibration amplitude (μm)

J: Jerk (m/s^3)

f_d: lowest natural frequency (Hz).

Thus, if the machine presents high compliance in the low frequency range, the motion capabilities can be limited, and more conservative axes control tuning will be required in order to avoid vibrations and geometrical errors when accelerating and decelerating the moving axes.

A particular case of this type of vibrations occurs when the cutting tool enters or exits the workpiece to be machined, producing a similar effect to jerk. As the impact force increases, the marks on the workpiece will be more noticeable.

The dynamic characterization can also be of relevance to perform cutting process analysis such as the typical tool life characterization. Few works have addressed this issue. Kuljanic [15] tested and modeled the influence of the stiffness of the machining system on tool life, concluding that a lower stiffness could lead to higher tool life. Kayhan and Budak [16] drew the same conclusion, pointing out that although this is true for a stable process, chatter vibrations significantly reduce tool life. Ghorbani et al. [17] also verified the influence of vibrations on tool wear.

Current machine digitization trends have led to an increasing number of sensors and actuators integrated in the machine, which can be used for monitoring and maintenance purposes. Although currently most of the monitoring techniques do not consider machine dynamics, ongoing research works are pointing in this direction. Quinn et al. [18] used active magnetic bearings to excite a rotor shaft and assess the emergence of cracks considering the measured dynamic response. On the other hand, the correct assembly of a structure can be also assessed on a dynamic response similarity basis [19]. Lastly, Cao et al. [20] developed a method for monitoring sensor placement optimization based on dynamic response simulation.

For these reasons, the understanding and measurement of machining dynamics is crucial to predict machining behavior, implement mitigation measures and, ultimately, optimize machine tool design and cutting process. However, the standard theoretical and experimental dynamic parameter identification approaches do not consider system nonlinearities and are usually subject to a high level of uncertainty, making them valid only for particular experimental setups.

In this work, first the standard machine tool frequency response function estimation methods, which have been called into question [21], are briefly reviewed. Then, the main sources of errors in machine tool dynamic characterization, which are claimed to be a major source of inaccuracies in stability models [22], are reviewed. These sources of errors are related to the operational conditions of the machine tools. The variable dynamics inherent of the cutting process due to machine position change [23], tool change [23] or material removal process [24] are the most obvious reasons for dynamic response deviation with respect to the idle state FRF. The rotation of tool spindle axis can be subject to gyroscopic effects [25] or centrifugal forces induced effects [26] that alter the frequency response. Also, the contact of the tool with the workpiece [27], nonlinear friction effects [28] and control settings [29] may vary damping and stiffness properties of the system.

Finally, new alternative experimental methods used to avoid these error sources are described. The rotating tool excitation, inverse methodology for dynamic parameter extraction from cutting tests, operational modal analysis (OMA), real cutting force excitation method and digital image correlation are the five main groups of alternative methods for dynamic parameter extraction collected in this review. The strengths and weaknesses of these methods are pointed out and finally, an overall conclusion is drawn.

2. Conventional experimental dynamic characterization methods

The excitation $F(j\omega)$ and the response $p(j\omega)$ of the machine tool are related through the frequency response function (FRF) $[\Phi(j\omega)]$ as:

$$\{p(j\omega)\} = [\Phi(j\omega)]\{F(j\omega)\}. \tag{2}$$

In general, FRF can be defined in three directions, therefore a matrix of 9 terms is needed:

$$[\Phi(j\omega)] = \begin{bmatrix} \Phi_{xx}(j\omega) & \Phi_{xy}(j\omega) & \Phi_{xz}(j\omega) \\ \Phi_{yx}(j\omega) & \Phi_{yy}(j\omega) & \Phi_{yz}(j\omega) \\ \Phi_{zx}(j\omega) & \Phi_{zy}(j\omega) & \Phi_{zz}(j\omega) \end{bmatrix}. \tag{3}$$

If the FRF at the cutting point is sought, then, in general, the dynamic response on both tool and workpiece sides must be considered along with their respective potential cross terms [23]. The dynamic response will therefore yield:

$$[\Phi(j\omega)] = [\Phi_{tt}(j\omega)] + [\Phi_{ww}(j\omega)] - 2[\Phi_{tw}(j\omega)] \tag{4}$$

where $[\Phi_{tt}(j\omega)]$ is the drive point FRF at the tool, $[\Phi_{ww}(j\omega)]$ is the drive point at the workpiece and $[\Phi_{tw}(j\omega)]$ is the cross FRF between tool and workpiece.

Therefore, these 9 FRFs characterize the dynamic response of the machining system (see Fig. 1) and can be used to determine possible limitations related to chatter vibrations and forced vibrations created by the cutting force. Thus, it is important that these FRFs are obtained as close as possible to the cutting point in both tool and workpiece sides, especially in those cases where local modes can affect the dynamic response. This characterization is typically conducted through hammer or shaker tests. It has to be noted that the dynamics may vary depending on the machine position, the tool/head configuration and/or the workpiece material removal state.

Apart from the need to obtain FRFs at the cutting point to calculate cutting forces, the FRFs at the drive systems of the moving axes are also of interest for studying the effect of inertial forces on these axes. These FRFs can be obtained by using the drive axes to excite the structure and the response of internal signals of the machine to catch the response. This measurement can be done either in open or closed loop.

Point FRFs are enough to analyze the main dynamic behavior and can be used in frequency domain stability models [2,30]. However, it is usual to apply curve fitting algorithms to extract modal parameters from FRFs [31,32] which creates the possibility to work in modal space [33]. Modal parameters can also be used in time domain stability simulations [34], although recently direct use of FRFs in time domain stability analysis has also been proposed [35]. This fitting also permits the study of the machine movements considering the effect of control parameters of the drives on the main modes [36,37].

The point FRFs are not enough to assess the improvement of the dynamic behavior of the machine tool. A complete experimental modal analysis (EMA) is required to extract the shape of the critical modes, detect the weak points and start an optimization loop based on the results of EMA [3,38]. This review will be focused only on the measurement and estimation of point FRFs.

From a theoretical approach, a finite element model of the machine [39,40] can be useful to estimate the point FRF at a desired location. In this case, damping has to be arbitrarily determined and/or experimentally correlated [40] but, in return, the model can be reduced and coupled to the drive model which includes the control loop, by means of a mechatronic simulation [36,37].

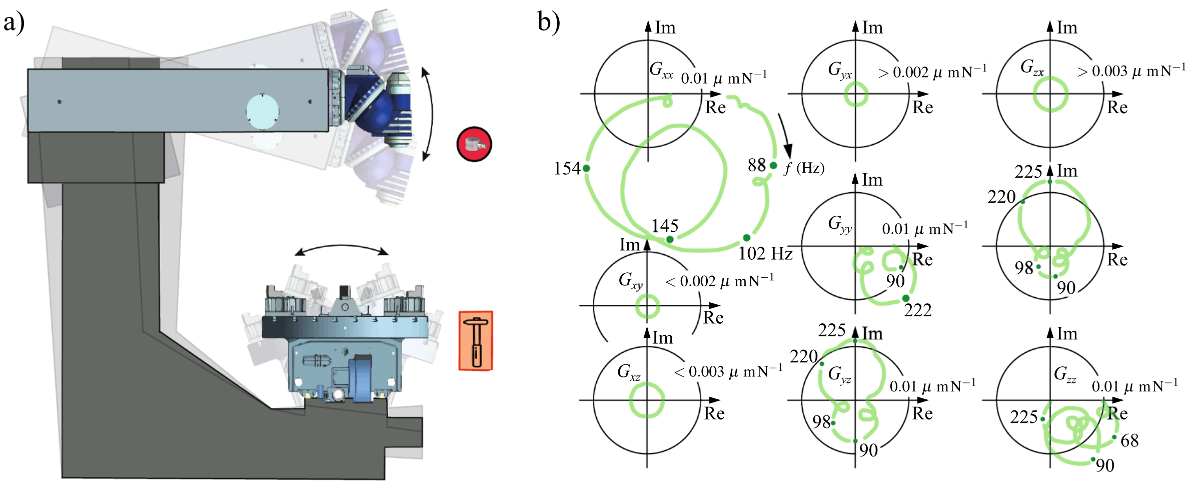

Fig. 1. Dynamics of a milling center: a) Cross FRF in a milling center with modal displacement from tool and workpiece side under the same vibration mode. b) Dynamic response characterization of a machine tool through 9 FRFs [3].

2.1. Impact hammer excitation

Nowadays, impact hammer testing is the usual method for dynamic parameter identification when simple FRFs are sought due to its simplicity, especially when analyzing environments with difficult access. There is no need for any previous setup to perform the measurement.

The hammer test (also called impact or tap test) consists of exciting the machine tool by means of an instrumented hammer and measuring the response by means of an accelerometer, velocimeter or displacement sensor.

Depending on the nature of the excitation, a different range of frequencies can be excited. The weight and hardness of the hammer tip, its material and the impact velocity affect the impulse shape and therefore the excited frequency range. The main problems related to hammer testing are related to poor signal to noise ratios, signal truncation and overload problems [41]. Moreover, the lack of repeatability of the measurements of the different averages (both in magnitude and direction) can also become a source of errors [31]. When an irregular surface is excited, the excitation direction may not be the same for every measurement. In addition, the typical double impact issues can come into picture when flexible structures like slender tools are excited. Kim and Schmitz [42] evaluated the reasons for uncertainties in FRF measurements by impact hammer testing. They reported statistical variations, calibration coefficients and misalignment of the impact force as some of the major contributors to the uncertainty in the FRF measurements.

2.2. Shaker excitation

In general, hammer testing is not well suited for nonlinear systems [43] and for that reason, when nonlinear systems are analyzed, the shaker is a more preferred option, although its time-consuming set-up makes it usually not worthwhile.

The nonlinearities can have different sources in a machine tool as well as in a general mechanical system. These nonlinearities can have their origin in the Coulomb friction at mechanical interfaces [44], rolling elements of translation carriages [45] or backlash in mechanical joints [46]. In a nonlinear system, the superposition, the homogeneity and the Maxwell reciprocity principles which are fulfilled in linear systems do not apply, which makes the perturbation magnitude and shape determinant in the system's final response [47]. Therefore, the dynamic response becomes excitation dependent. From a theoretical point of view, the nonlinear systems can lead to bifurcations [48].

Extensive research has been carried out to identify and characterize nonlinear systems [44], from both experimental and theoretical points of view. From the experimental point of view, the analysis of the system nonlinearity requires the use of specific shaker excitation methods for dynamic characterization.

Shaker excitation can be classified in two main groups, random or deterministic excitation [49]. While random signals can only be defined by their statistical properties, deterministic signals can be expressed by a mathematical relation (see Fig. 2). The choice of the input type to excite the system depends on its linearity. If the system is linear, theoretically speaking, all input types would result in the same FRF. However, if the system is nonlinear, a random excitation of the system will result in its linearized representation. On the other hand, with a deterministic excitation, the nonlinearity of the system with respect to the input force or the excitation type can be characterized [50].

One of the most frequently used methods is the frequency sweep with different controlled force and displacement levels ("slow swept sine"). This technique is usually very costly and time consuming. The "periodic chirp", in which a sine excitation continuously varies from a minimum to a maximum frequency in a certain period T, is the fastest option. Nevertheless, this second excitation type may not be enough to observe the real behavior at each single frequency.

The excitation through shaker allows using different references, even simultaneously, to shake the structure in a proper way and decouple similar modes in shape and frequency. Several Multi-Degree of Freedom (MDOF) fitting methods have been developed, such

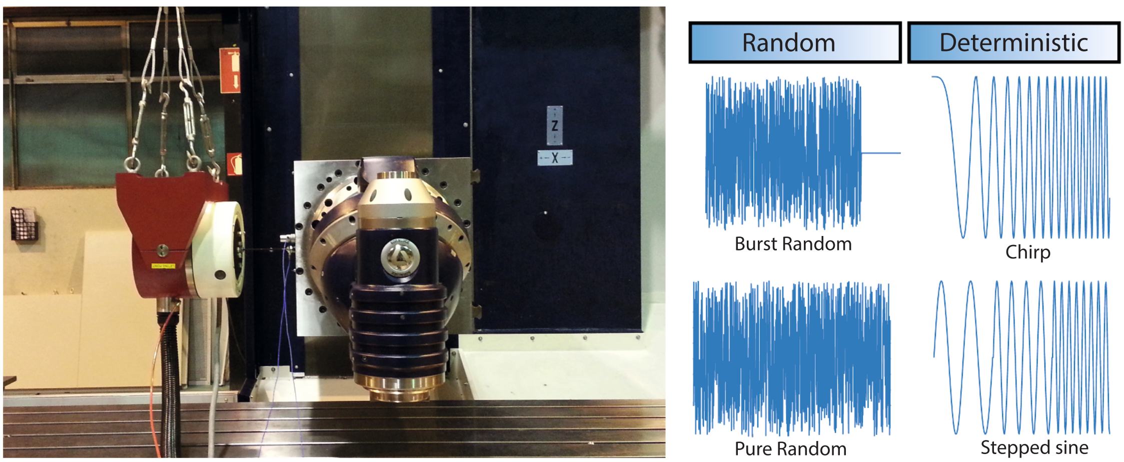

Fig. 2. Shaker excitation setup and most common excitation signals.

as the Non-Linear Least-Squares (NLLS) or the Rational Fraction Polynomial (RFP) [31] or the more recent Least-Squares Complex Frequency-domain method, which improves the identification of closely spaced modes [32].

The dynamic characterization through shaker excitation requires a complex setup and it is therefore very time consuming. For that reason, the process can be further simplified by adding an inertial actuator to the machine tool to perform the dynamic characterization as an automated process [51,52]. Recently, leveraging current trends in cyber-physical system development for manufacturing equipment, intelligent functions to obtain the dynamic response of the machine tool by means of integrated inertial actuators and sensors have been implemented [53].

Rasper et al. [22] encountered big differences in the dynamics of a machine measured with an impact hammer compared to the results obtained by means of a shaker excitation. Certain variability in the mobility due to the changes in the angular position of the spindle was also observed. The effect was explained measuring the mobility on an idle machine tool, where dynamic stiffening effects cannot be detected. Therefore, rotating spindle measurement methods development is regarded as indispensable.

2.3. Machine drives excitation

Machine tool feed drives are used for positioning the machine through their sensors and actuators [54], and their positioning accuracy and dynamics will determine the quality of the machined part and productivity of the machine.

Existing vibratory modes which interact with the machine servo loops constitute a major limitation in the way of increasing the control gains. CNC systems provide frequency response (Bode) plots of the feed drive, where the axis is self-excited and the response is measured via the velocity or position feedback sensors.

During the Bode plot measurement, it is important to impose a position ramp at a constant speed to minimize the static friction effect, and hence, locate the machine axis in the linear viscous regime. Moreover, the amplitude of the excitation force must be low enough to prevent the axis to return to the low-speed regime. An offset reference value of 300 mm/min is shown in the literature [14]. This speed is considered high enough to always keep the system under dynamic friction conditions during the excitation test.

Regarding the excitation signal type, different approaches are followed by CNC manufacturers. On the one hand, Pseudo Random Binary Sequence (PRBS) signals are employed to excite the machine all over the frequency band. The PRBS signal can be constructed as follows:

$$u(t) = A\hat{\text{A}} \cdot sign(w(t/Nc)), \tag{5}$$

where A is the amplitude, $w(t)$ is the white noise stochastic process and Nc is the clock period. The clock period can be modified in such a way that the power of the input signal is focused on lower frequencies.

On the other hand, excitation commands such as frequency sweeps are also employed in the industry. Linear or logarithmic formats are available for identification purposes, where the latter exerts higher energy at low frequencies.

3. Sources of inaccuracies in dynamic characterization

The experimental dynamic characterization of machine tools presents certain degree of uncertainty [4]. The acquisition and signal treatment procedure [41] can result in some inaccuracies including:

- Acquisition errors: electrical noise, mass loading effect by the sensors [34,55], interaction between the stinger and the excited structure…
- Signal processing errors: aliasing, leakage, windowing, discretization…
- Fitting errors: difficulties with close spaced modes, ….

Apart from these factors, there are other aspects increasing the uncertainty when determining the dynamic compliance of a machine tool [56]. Many effects rising under operational conditions have been reported as possible sources of error for dynamic identification [28]. The machine tool is not a static passive structure, since it is controlled through electric drives, it can move, rotate or cut material and the possible side effects of these events are not considered when performing a classic hammer or shaker test at the idle

Table 1
Machine tool factors affecting dynamic behavior.

Factor	Effect on dynamics	Reference
Variable dynamics	Changes in frequency, damping and receptance	[57-78]
Gyroscopic effect	Increase of natural frequency Reduction of damping	[79-87]
Bearing stiffness	Increase of natural frequency Increase of stiffness	[88-94]
Control parameters	Dynamic stiffness change	[95-99]
Non-linear dynamic properties of the interfaces	Non-linear damping Change of stiffness	[100–112]
Tool and workpiece contact	Increase of stiffness Increase of damping	[113–123]

state of the machine. These effects rise under operational conditions and cause deviations in the machine tool dynamics identification (see Table 1).

All aspects related to the six effects collected in Table 1 are described more in detail in the next six subsections:

3.1. Variable dynamics

In many operations, the critical (dominant) dynamics are not constant during the cutting process and this introduces an important difficulty in the dynamic characterization. These variations can be tool dependent, cutting position dependent, or material removal dependent.

- Tool dependency:

In several machining systems, such as high speed aluminum rough milling, the modes limiting the cutting process are associated with the tool, the toolholder or the spindle (the chatter frequency lies roughly between 300 Hz and 6000 Hz). Therefore, in order to characterize the response at the tool tip, the measurements have to be repeated for each combination of tool/toolholder/spindle.

In order to overcome this drawback, Schmitz and Donalson [57] proposed a receptance coupling (RCSA) technique to predict the dynamic response at the tool tip. RCSA has been from them extended to particular cases such as boring bars [58]. The technique allows coupling of analytical and/or experimental FRFs of individual components, bringing the response of the final assembly together. By means of the RCSA, considerable time can be saved on the prediction of the FRFs at the tool tip for different tools, combining the theoretical response of the different tools and the experimental results of the toolholder/machine assembly. This technique permits to handle tool dependency efficiently, even in the case of complex damped boring bars for turning [59] and milling. [60]. Postel et al. [61] extended the RCSA approach by identifying speed dependent spindle dynamics, increasing the accuracy of the stability prediction.

- Position dependency:

In steel roughing and other difficult to cut materials, where spindle speed is relatively low, the critical modes are usually related to the whole machine tool structure (the critical frequencies are roughly between 15 and 200 Hz).

In multi-axis machines with movable carriages and extensible elements such as a ram, the dynamic response of the system is strongly dependent on the machine position and has to be considered when performing the dynamic characterization [23]. Therefore, in some trajectories the dynamic properties are continuously changing, hindering the measurement process [4]. Some authors have studied the effect of this continuous variation in the stability predictions [4,62].

This issue can be either addressed theoretically, modelling the axis position dependent dynamics by means of a simplified finite element model or experimentally. Brecher et al. [63] theoretically modelled a multi-axis machine through a Dual Craig-Bampton method. Law et al. also used reduced order models to analyze the position dependency of machine tools [64] and parallel kinematic machines [65]. Deng et al. [66] predicted the machining position and feed direction-dependent tool point FRFs based on the modal theory, matrix transformation, BP neural network and PSO algorithm.

Finkeldey et al. [67] used a mixed approach in which some positions were experimentally characterized and the remaining different workspace pose dynamics were predicted through machine learning methods.

Other works have automated the experimental position dependency characterization through a machine dynamics calibration system, by means of an integrated active damper on the machine [68].

- Material removal dependency:

In thin wall machining, the dynamic properties change as the machining position varies and as the material is removed. In these operations, the characterization of the dynamic behavior of the workpiece can become a key factor for the effectiveness of the machining process. This is the case of many parts of the aerospace sector, including structural and engine parts, or pipe threading in the oil and gas sector.

Generally, FEM models are used to obtain the dynamic flexibility and to predict the modal parameters and vibration modes, considering the variations due to material removal [69-73].

The analysis and modelling of the dynamic behavior of thin-walled components is a hot research topic. Structural modification techniques are the most efficient approaches for determining the dynamics of flexible parts in every machining step. They consist of calculating the dynamic properties of a certain modified structure with changes in mass, damping and/or stiffness based on the original model. The Matrix Inversion Method [74] and Matrix Perturbation Method [75] have been proposed for this purpose.

Biermann et al. [76] presented a general approach to simulate workpiece vibrations during 5 axes milling of turbine blades simulating dynamic parameters of the thin wall by a FEM model, whereas Budak et al. [24] included the effect of workpiece dynamics on chatter stability analysis in 5-axis milling of turbine blades. They took the continuously varying structural dynamics of the blade type thin wall geometries into account by implementing a structural modification algorithm based on the FE mesh of the final shape and the stock to be removed. Yang et al. [77] also researched the characteristics of the in-process material removal of large-scale thin-walled workpieces with curved surfaces.

The damping estimation of the thin-walled parts is more difficult due to the effects of the clamping on the different modes. Methods to estimate the damping have been proposed in the literature [78], but their reliability has not been confirmed.

3.2. Gyroscopic effects

The gyroscopic effect causes a splitting of the natural frequency into two backward and forward frequencies in modes related to rotating axles that are critical in high speed machining of light alloys (see Fig. 3). It has to be noted that gyroscopic effects are only significant at very high rotating speeds, being negligible at conventional speeds. The higher the speed and the inertia the higher the gyroscopic effect [79]. In fact, the inertia of the axis is usually reduced when the spindle speed is increased, and this reduces the impact of gyroscopic effects in many processes.

Although there is consensus about the bigger effect of damping on the backward frequency [80], there are several researchers who reported a decrease in dynamic stiffness due to the gyroscopic effect due to the reduction of the damping of the forward mode [25,81]. In fact, Movahhedy and Mosaddegh [82] described gyroscopic effect as a negative damping effect.

Although this effect is reported in rotor dynamics and machine tool spindle models, both modes cannot be identified experimentally in machine tools. This is due to the damping values of the spindle assembly. Cao et al. [83] showed that since the damping of the spindle assembly is usually between 2% and 5%, backward and forward modes combined into one peak and cannot be experimentally distinguished. On the other hand, Bediz et al. [84] measured the dynamics of a miniature ultra-high-speed spindle at different speeds (up to 170,000 rpm) using a custom-made impact excitation system and identified the backward and forward modes experimentally, showing the speed dependency experimentally. Uriarte et al. [85] confirmed experimentally the effect of gyroscopic effects in a magnetically levitated spindle through ramp up and cost down tests.

Second effect of the gyroscopic moments on the spindle shaft dynamics is that the cross FRFs of the system between the orthogonal directions increases with the increasing spindle speed [83,84,86]. Tian and Hutton [25] determined that the gyroscopic effects of the rotating spindle increase the magnitude of the real parts of the eigenvalues of the system. Therefore, the critical axial depth of cut is reduced and the instability regions are extended, especially at high speeds.

Cao et al. [83] claimed that the gyroscopic effects have much less effect than centrifugal forces on bearing stiffness. Gagnol et al. [87] found significant differences in the stability lobe charts of a static and a dynamic FRF and attributed it to the gyroscopic effect and spin softening effect. Recently, Ma et al. [60] included the gyroscopic effects in the prediction of the point FRFs of rotating damped tools by means of receptance coupling.

3.3. Bearing stiffness

Spindle bearing preload changes with centrifugal forces, cutting forces and thermal deformations. When spindle bearing preload is increased, the natural frequency and the dynamic stiffness of the system also increases [88].

In standard bearing assemblies, centrifugal forces produce an enlargement of the normal force and a decrease of the contact angle at the outer race. In order to ensure equilibrium, the normal force at the inner ring will become smaller and the contact angle will increase (see Fig. 4). The enlargement of the difference between contact angles at the inner and outer race will reduce the bearing stiffness.

During the cutting process, centrifugal forces and cutting forces push the balls of the bearings against the race, changing bearing's stiffness and damping values. According to [89], system stiffness increases with bearing preload and therefore, given that centrifugal forces press bearing balls against the outer race reducing its preload, high rotational speeds result in a bearing stiffness decrease. This was also confirmed by [90,91]. The latest modeled rigid and constant preload spindles. They verified that the spindle speed decreases the bearing stiffness, reducing the natural frequencies.

Postel et al. [92] analyzed the effect of variation in axial and radial preload and spindle speed on a five-axis milling machine spindle, concluding that the axial preload changes give raise to the bigger changes in dynamic response.

Abele and Fiedler [26] pointed at the temperature increase at the bearings as another cause for bearing preload and, hence,

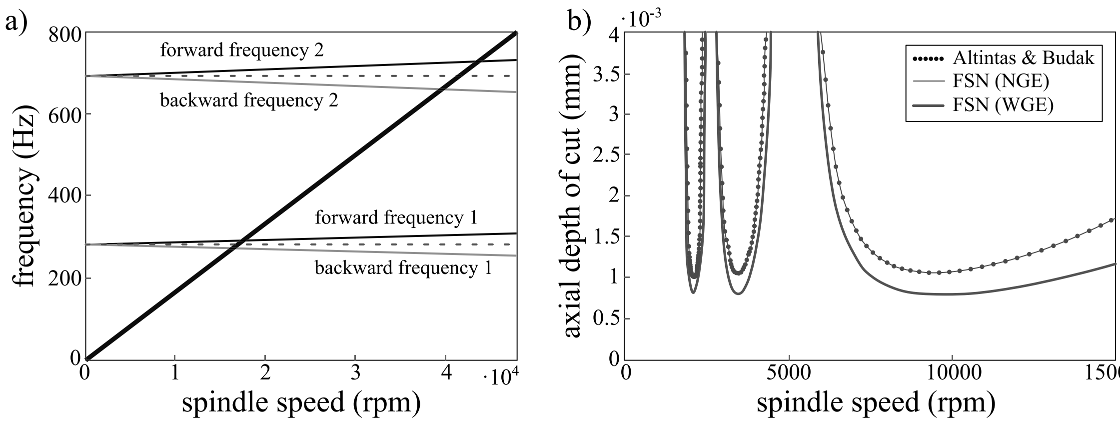

Fig. 3. a) Gyroscopic effect in spindle's natural frequency: forward and backward frequencies and b) gyroscopic effect on stability [25].

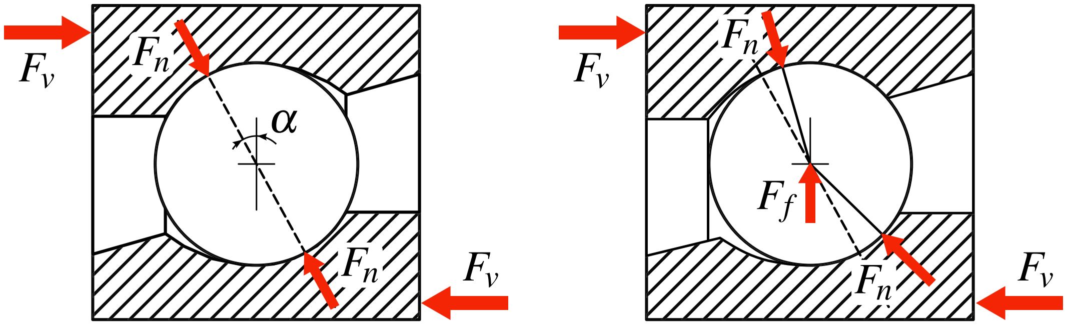

Fig. 4. Bearing contact forces distribution change due to centrifugal forces [26].

dynamic stiffness change. The temperature difference between the inner and the outer race also affects the bearing stiffness. In most bearing assemblies, rotating speed gives rise to a differential temperature between the inner and the outer race of the bearing, being the latter cooler. Although this normally implies an increase of the preload and, in turn, an increase of the bearing stiffness, a proper face-to-face bearing configuration can cancel this effect and preserve bearing preload regardless the temperature increase [93].

Hongqi and Yung [94] developed and validated a dynamic thermo-mechanical model for high speed spindles considering the bearing preload change under thermal expansions.

All these works have proved the presence of important changes in the dynamics of the spindle for high-speed machining applications.

3.4. Control parameters

Altintas et al. [95] pointed out the necessity of coupling the control loops to the structural dynamics in order to simulate the overall dynamics of the machine tool structures. Wiesauer et al. [96] showed clear changes in the dynamics of a ball screw drive of machining center when the motor is braked versus the free ball screw with the control active in the idle state.

The tuning parameters in the velocity control loop play an important role in machine tool mode damping or amplification. It has been demonstrated that *Kp* (speed loop proportional gain) and *Ti* (speed loop integral time) affect the frequency response function of the system. According to [29], a too low *Ti* parameter could decrease dynamic stiffness, whereas a too high value of *Kp* does not provide additional damping. However, Wiesauer et al. [96] observed that a lower gain value results in a lower amplitude response of the system. The decrease of position control proportional gain *Kv* in a linear motor has also led to a slighter decrease in dynamic response [36]. On the contrary, another research by Grau et al. [97] in a linear drive showed the opposite trend, with a lower frequency response for higher *Kp* and *Kv* values.

Beudaert et al. [98] used the root locus technique to analyze the interaction of the servo controller with the machine dynamics. The damping ratio of the critical mode was increased through the control tuning and thus, the stability limit was increased.

Other electronic features such as the master–slave commanded coupling preload force in a rack and pinion drive of a machine tool can slightly alter its dynamics, as it was demonstrated by Franco et al. [99] (see Fig. 5).

As a conclusion, the effect of the control loops and the movements of the machine in the low frequency modes related to the

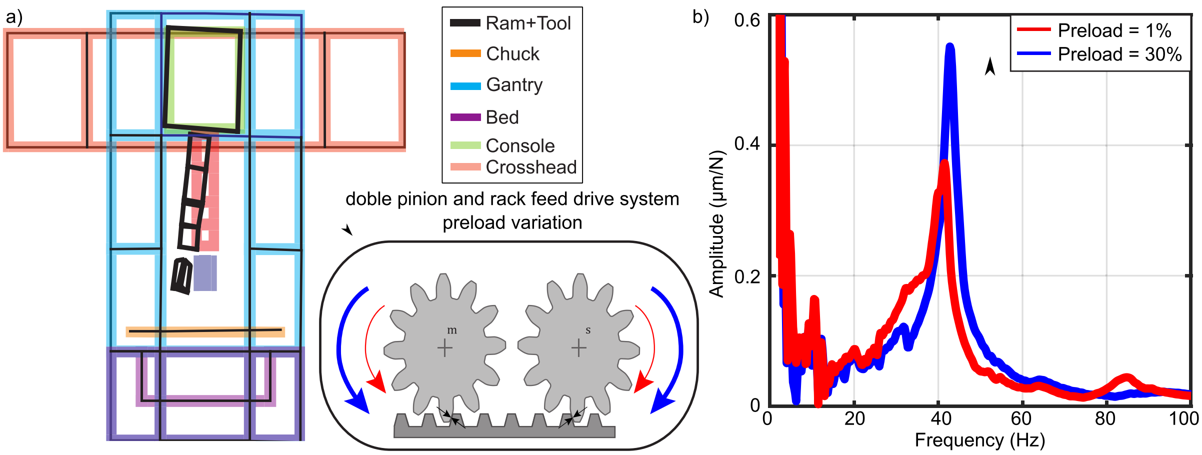

Fig. 5. a) Experimental rack and pinion related mode shape and b) effect on dynamic stiffness.

machine tool structure has been demonstrated and should be considered in the dynamic characterization.

3.5. Non-linear dynamic properties of the interfaces

The interfaces are key elements in the definition of the dynamic properties of machine tools. Zhang et al. [100] concluded that approximately 60 % of the total dynamic stiffness and 90% of the machine total damping is originated at the joints. On the one hand, the stiffness of the joints defines in a great extent the overall stiffness and the dynamic behavior of the whole machine tool system. Much effort has been done in bolted joints and other type of fasteners modeling, but many difficulties have been found due to non-smooth, nonlinear and time variant behavior of these joints. The stiffness of the joints can be identified based on experimental measurements of the complete assembly [40,101]. Ibrahim and Pettit [102] offered a review of current problems and uncertainties regarding joint dynamic modeling.

On the other hand, the damping of the system is mostly defined by joints and interfaces [7]. In general terms, the overall damping value of one mode increases as the number of joints involved in the shape of considered mode is increased.

Bianchi et al. [103] analyzed the friction influence on the dynamic compliance of the machine tool at the cutting point. Later, Rebelein and Zaeh [104] developed a virtual machine tool model that considers multiple damping sources. This model was extended by adding feed drive servo controls and machine movements [28,105]. They concluded that linear dissipation sources and non-linear friction forces have the highest influence on the machine tool vibration characteristics. A damping variation of up to 35% from standstill to moving responses was claimed.

In 2020, Sato et al. [106], after analyzing the effect of the friction force at a ball screw driven table, claimed that non-linear friction affects the machine dynamic characteristics. Similarly, Oshita et al. [107] showed that the damping and stiffness properties of the guideways are modified in the radial and vertical directions when the axis is moving at different feedrates. Wiesauer et al. [96] measured an increase of damping due to carriage friction on a machine tool dynamic response. Tunç and Gonul [108] indirectly observed the effect of friction when comparing the dynamic response of a robot in static and quasi-static state. The dynamics varied significantly, and the machining stability predictions matched better with experiments for the quasi-static case.

Contact surface dynamics may also affect the tool point FRFs, such as the spindle-holder and holder-tool interfaces. Spindle-holder interface dynamics mainly affects the spindle and holder dominant modes and holder-tool interface dynamics affects the tool elastic modes [109]. Since contact dynamics are directly related to contacting surfaces, wear at the spindle-holder interface also affects the contact parameters and stiffness decreases as wear increases [110]. Although it is difficult to include the effect of wear in contact models, models using distributed springs can be used for that purpose. In distributed spring models, the influence of losing some of the contact areas due to wear can be simulated by removing the corresponding spring elements [111]. In addition, Namazi et al. [111] showed that increased bending loads change the contact stiffness between the tool holder and spindle taper, which need to be included in the estimated contact stiffness.

In addition, contact parameters at the spindle-holder interface are directly affected by the centrifugal forces. As an example of this, radial stiffness of the HSK tool interface decreases with increasing spindle speed due to the loss of contact at the tapered face in the high-speed range [112]. Similar results are also reported for the 7/24 tapered interface [112]. Centrifugal forces also have an effect on the drawbar mechanism by the modification of the applied axial preload and thus, the contact stiffness. Chen and Hwang [90] reported that the centrifugal forces cause an increase in the drawbar force, hence, an increase in the contact stiffness and the dynamic stiffness of the spindle unit.

As a conclusion, it is possible to affirm the dynamics and specially damping of machine tool is defined by joints and interfaces with an important non-linear behavior. Therefore, it is interesting to measure the machine tool dynamics close to operational conditions.

3.6. Effect of the contact between tool and workpiece

There are authors that claim that the contact between tool and workpiece produced during the cutting process can affect the stiffness and the damping of the system. Jensen and Shin [27], for instance, claimed that the natural frequency of the structure varies significantly when the machine is in contact with the workpiece with respect to when it is not due to this additional stiffness produced by the cutting process. Zaghbani and Songmene [113] also reported considerable deviations in the FRF obtained through tap testing of a tool in contact and without contact with a workpiece. One of the reasons that may explain this effect is the own stiffness of the cutting process, which can alter the intrinsic stiffness of the idle system to some extent.

Additionally, the so-called process damping phenomenon is another effect that could add additional damping to the machining system when the cutting process takes place. There is no consensus related to the analytical description of the process damping effect and there are different theories to explain the involved mechanisms [114,115]. One of the early theories was based on the dynamic cutting force coefficients (DCFC) [116]. Collaborative studies of several CIRP laboratories were summarized by Tlusty [117]. The difficulty of the test setups and inconsistency in the collected data were also discussed together with the possible causes. In one of the modern studies, Altintas et al. [118] used a servo driven oscillatory system to measure the dynamic cutting force coefficients.

The most extended theories to explain process damping is built on top of the tool-workpiece indentation, where it is claimed that rubbing of the tool against the workpiece in dynamic cutting provides additional damping to the machining system. When there are many oscillations in a single tool revolution, the effective clearance angle becomes negative in the slope-down motion of the tool. This means that the flank face of the tool rubs and indents against the undulations left on the workpiece surface as a result of dynamic cutting. Due to such an indentation, contact force arises against the vibration direction, which acts as extra damping effect to the cutting process at low cutting speeds. Since the modelling of this effect is a complex task, it has become a major research subject. In the noted early studies, Sisson and Kegg [114], Wu [115], Peters et al. [116] and Tlusty [117] showed that the indentation forces arising due to flank-wave indentation against tool vibration contribute to the dynamics of the cutting process by increasing the overall damping acting on the system. Sisson and Kegg demonstrated that edge hone on the tool, cutting speed and clearance angle are the critical process parameters which majorly affect process damping. They studied the effect of process parameters based on experiments and time domain simulations by modeling the indentation forces as a function of the tool flank - workpiece interface volume and a material constant provided by the previous study of Wu [115,119]. However, no method was proposed for identification or modeling of the process damping and determination of stability limits under its effect. Eynian and Altintas [118,120] proposed a general model to predict process damping effect in generalized turning operations, where they included the cutting edge – workpiece indentation to calculate dynamic cutting forces with indentation effect. Budak and Tunc [121] proposed a novel and analytical approach to identify the amount of process damping acting on the cutting system, based on the inverse solution of the stability equation. The identified process damping coefficient was later related to the indentation effect to develop a generalized model. In a later study, Tunç and Budak [122] extended this approach to general milling processes. They considered two types of tool flank geometries such as linear and elliptical flank. The effect of process parameters and tool geometry on process damping was discussed by Tunç and Budak [123] through simulations and experimental results.

Finally, it is important to remark that the dominant tendency assumes that the dynamics of the machine tool should be identified when the machine is not cutting, and the additional stiffness and damping is considered by means of the cutting coefficients that can be calibrated experimentally.

4. Alternative experimental methods for dynamic characterization

In the literature, most of the dynamic characterization studies have been performed based on impact-hammer or shaker tests. However, the obtained results are conditioned by the effects described in section 3. In this regard, several alternative experimental methods have been developed to perform dynamic characterization under operational conditions. In this section, these alternative methods to the classic hammer or shaker test are described.

4.1. New excitation devices for rotating tools

Different gadgets and methods have been proposed in the last years to excite a rotating spindle. As a first attempt, the rotational effects on the machine tool dynamics could be captured by means of impact hammer testing performed directly on the rotating spindle. In the literature, this approach was implemented by several researchers [56,124,125], which may be also interesting for tool point FRF identification in micro milling operations. Bediz et al. [126] developed a custom-made impact excitation setup and measured the speed dependent dynamics of an ultra-speed miniature spindle [84]. For a similar purpose, in a recent study, Wiederkehr et al. [127] proposed a practical approach relying on shooting of micro-spheres at the cutting tool tip. They modeled the impact force once a micro-sphere is thrown on the cutting tool tip. Bachrathy et al. [128] used a similar method to obtain the natural frequencies of thin wall geometries during machining instead.

The automatic hammer testing is an interesting alternative, although it usually requires a complex set-up which makes the procedure unfeasible. For this reason, non-contact methods have been more deeply studied.

With the purpose of considering the tool and workpiece contact, Weck [3] proposed the use of electrohydraulic or electromagnetic relative exciters to analyze this effect. These shakers can provide a static preload to the system, which can simulate the tool and workpiece contact, before applying the dynamic load.

Sims et al. [129] described the use of piezoelectric sensors and actuators for tool excitation and response measurement. They recommended this method for small milling tools, which may be difficult to test with a modal hammer. Kono and Umezu [130] went

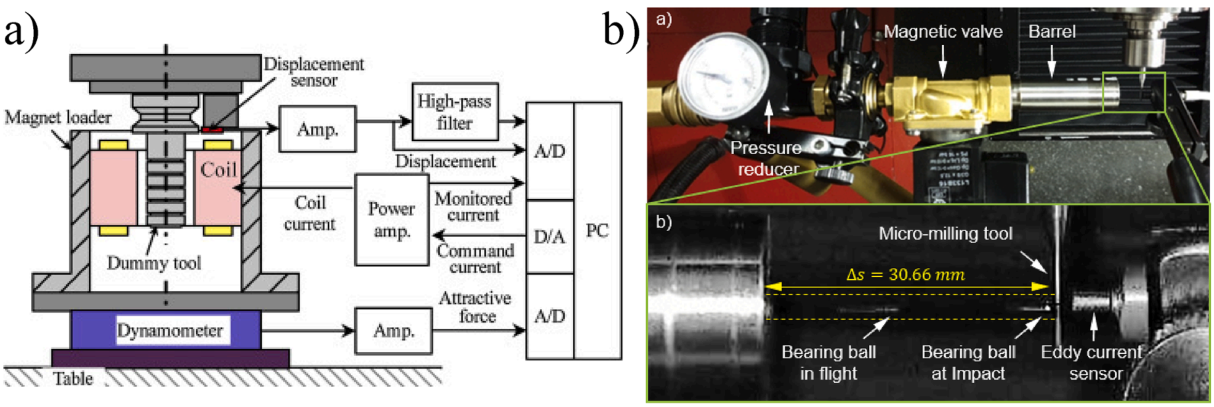

Fig. 6. FRF measurement systems for rotating spindles (a) non-contact FRF measurement system as proposed in [133] (b) ball-shooting system for micro-milling tools as proposed in [127].

one step further exciting a thin wall with a piezoelectric actuator and estimating the displacement response without attaching any sensor to the workpiece.

Active magnetic bearings (AMB) have also been used in order to identify the spindle tool system's frequency response function (FRF). Abele et al. [131] used an AMB to identify the spindle tool system's FRF. This method allows making non-contact measurements when the spindle is running. By means of the substructure receptance coupling techniques the theoretical dynamics of the tool can be added to the spindle's and the response at the tool tip can be estimated for different tools. Rantatalo et al. [132] also used AMB to excite the structure and non-contact displacement sensors to measure the response. All these authors reported differences between 0 rpm and rotating spindle FRFs.

Matsubara et al. [133] (see Fig. 6) and Tlalolini et al. [134] used a non-contact excitation device for FRF analysis in rotating spindles. The former analyzed the changes in FRFs and stability lobes due to speed and force changes, concluding that the last induces larger variations.

The non-contact magnetic shakers have a big potential to measure rotating tool FRFs and automatize the calculation of stability diagrams for high-speed applications.

4.2. Inverse methodology for dynamic parameter extraction from cutting tests

A direct method to avoid all sources of inaccuracies described in section 3 is to identify the machine tool dynamic parameters directly from real cutting tests. The main idea is to perform a set of cutting tests to identify the stability limit for different spindle speeds. Then, an inverse mathematical procedure is used to extract the main dynamic parameters based on the experimentally identified stability limit and corresponding chatter frequency.

The first step in these procedures is the identification of the experimental chatter stability limit. Quintana et al. [135] performed cutting tests on a roof shape workpiece at different speeds, printing the real stability lobes in the workpiece itself. Ismail and Soliman [136] proposed a similar approach but using a speed-ramp instead. They used the *R*-value chatter indicator to define stability borders and compared them with cutting tests successfully, although they observed mismatches when very high speeds were used. Grossi et al. [137,138] also used a speed-ramp test, but unlike Ismail and Soliman, they kept the feed per tooth constant (see Fig. 7). They observed that the dynamic stiffness decreases and the natural frequency increases at higher spindle speeds.

Similarly, Postel et al. [61] identified the spindle dynamics by the combination of inverse stability and receptance coupling approaches and showed spindle dynamics are also affected by the feed rate.

Kruth et al. [139] considered the structure having a dominant mode in a single direction. They developed an inverse analytical formulation based in zeroth order solution. They assumed a single mode direction and predicted modal parameters using experimental data. However, this method cannot take account two modes with similar frequencies but with perpendicular directions. Suzuki et al. [140] studied a spindle which has symmetric modes in two orthogonal directions. They used an iterative numerical method to adjust dynamic parameters to fit experimental data based on zeroth order solution. Kilic et al. [141] developed an analytical formulation for the case of two equal orthogonal modes.

Özsahin et al. [142] developed a similar inverse approach based on stability tests. They simulated and validated experimentally

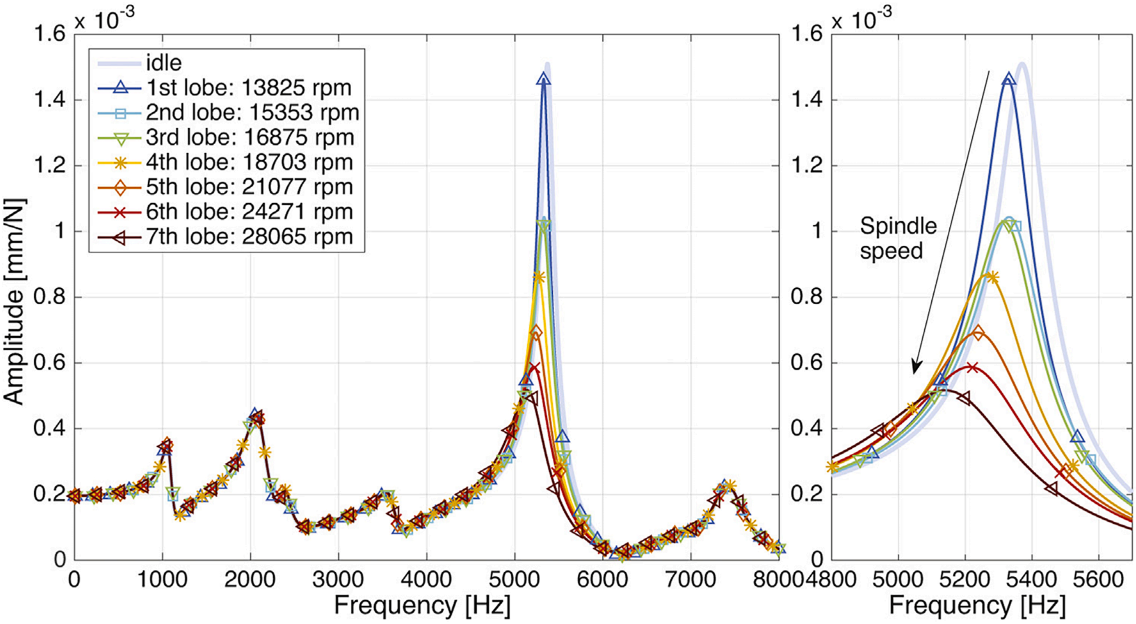

Fig. 7. Speed-varying FRF according to [138].

that tool point FRF is affected by spindle speed and cutting forces simultaneously. They discovered that when the stability of the cutting operation is determined by the tool modes, the tool point FRF is not affected by rotating speeds. However, when spindle and holder modes are involved, deviations due to operational conditions become crucial for a proper stability prediction. Similarly, Eynian [143] developed two closed form methods to obtain the dynamic parameters from experimental measurements, avoiding the problems related to iterations of previous methods, thus, allowing a faster calculation.

In the inverse stability analysis, expression of the chatter frequency and axial depth of cut is equated into experimentally identified ones to obtain the unknown natural frequency and damping ratios. As an alternative approach to the mathematical computation of the experimental results into the analytic stability equations, Postel et al. [144,145] employed neural networks based on chatter results and identified the unknown natural frequencies and the cutting force coefficients.

This inverse methodology can even address deviations related to dynamic behavior which are not considered by standard theoretical machining stability prediction methods yet, such as the damping increase due to the so called process damping effect. Budak and Tunç [121] proposed an analytical inverse stability solution where their major assumption was that the dynamic contact between the cutting tool flank face and the workpiece only alters the damping value. From this perspective, they used the ratio between experimentally identified and the theoretically expected stability limits. Later, they extended their approach to milling processes [122]. In their proposed approach, they used the ratio between the experimentally identified stability limits and the theoretical stability limits.

Although the inverse methodology provides a very accurate dynamic characterization, the limitations of the method are significant. Previous extensive cutting tests must be conducted and the method is only valid for very simple dynamics, because in most cases relies on a certain mathematical model with multiple simplifications and assumptions.

4.3. Operational modal analysis

In the operational modal analysis (OMA) the dynamic response of the machine tool is obtained under operating conditions, thus every possible non-linear effect which is not considered under the standard idle hammer test is included in the measurement. The main drawback is that the OMA is an output-only analysis method; the natural frequencies, mode shapes and damping ratios can be obtained from an output signal without knowing the input signal. It is not possible to extract residues, modal mass and modal stiffness, since they depend on the unknown input force.

There are two main different mathematical approaches to deal with the input data from the OMA analysis. On the one hand, the frequency domain decomposition (FDD) methods consist in calculating the spectral density matrix $G_{yy}(\omega)$ for each measurement and performing a Singular Value Decomposition approximating the spectral density matrix as:

$$G_{yy}(\omega) = [\phi]^H[\Pi][\phi] \tag{6}$$

where $[\Pi]$ is the singular value matrix, which contains the natural frequency values and $[\phi]$ is the singular vectors unitary matrix, which contains the mode shapes.

There are two derivations of the FDD method. In the enhanced frequency domain decomposition (EFDD) method, a single degree of freedom (SDOF) power spectral density function is identified around a natural frequency and then it is transformed back to the time domain through the Inverse Discrete Fourier Transform (IDFT). On the other hand, in the frequency domain decomposition (CFDD) method, the curve-fitting of the SDOF function around a peak is performed directly in the frequency domain, resulting generally in a more accurate estimation of the dynamic parameters. Both EFDD and CFDD can estimate damping ratios and can also handle possible harmonics in the input signal. Effective harmonic detection and removal of harmonics is one of the major challenges for an accurate operational modal analysis and different novel methods have been proposed for that purpose [146–148].

FDD methods rely on frequency domain, suited for stationary excitation, whereas stochastic subspace identification (SSI) is a time domain algorithm suited for both stationary and non-stationary excitations [149]. Thus, SSI is more powerful compared to FDD. It is non-iterative, since it identifies the “states” of the system before identifying the system itself.

Burney et al. [150] determined the dynamic parameters of a machine tool under working condition using a time-series-technique. The method was based on the analysis of the displacement signal between the tool and the workpiece, and studied the dynamics of the machine tool under different cutting conditions. Zaghbani and Songmene [113] developed and demonstrated a complete methodology to apply OMA under real working conditions in a milling process. Two different methods were applied: autoregressive moving average method (ARMA) and least square complex exponential (LSCE) method. They demonstrated experimentally that the dynamic parameters obtained from OMA were more accurate than those obtained through impact testing at 0 rpm. According to the experimental results, the stability limit calculated using the data obtained through the OMA analysis is lower and more accurate than the stability limit obtained using the static FRF. Schedlinski applied a classical operational modal analysis on a laser cutting machine [151] and a milling and a turning machines [152]. Some authors have even performed an OMA on a machine tool with microphones as sensors [153]. Cai et al. [154] also used OMA for structural dynamics identification in machine tools. They observed that the dynamics of machine tools presented in operation varied significantly from those results obtained by the traditional experimental modal analysis. Powałka and Jemielniak [155] applied OMA to milling of flexible parts, concluding that the stability diagrams constructed from OMA testing at different spindle speeds were in good agreement with impact testing.

Li et al. [156] and Mao et al. [157] performed an evolution of the OMA analysis and used the active excitation modal analysis (AEMA) instead. This method consists of exciting the machine tool with the drive system. Li et al. [156] performed a pure OMA post-processing of the obtained signals and came up with noticeable damping decrease with respect to a standard hammer test. Mao et al.

[157] scaled the mode shapes using the dynamic modification technique and synthetized an alternative FRF. They found differences in frequency with respect to the conventionally obtained FRF.

Berthold et al. [158] developed an OMA analysis in which the speed and the width of cut are respectively sinusoidally and linearly modified in order to avoid the harmonic excitation and achieve a random excitation of the machine tool.

4.4. Real cutting force excitation modal analysis

Unlike operational modal analysis, where the excitation forces are not measured, the real cutting force excitation modal analysis captures the response in real conditions and, in addition, takes into account the values of the input force, since this is measured. The use of the cutting force as an excitation input has some difficulties.

First of all, a sensor capable to directly measure forces has to be included in the force flow. Piezoelectric cells, dynamometers or toolholders can be used to measure this force [159]. However, the high price, limited workspace and fragility of these sensors have limited the industrial use of these sensors. Recently, high bandwidth toolholders based on strain gages have been proposed to measure cutting forces [160]. The use of strain gages may imply the reduction of the stiffness of the sensory element to assure the sensitivity of the gages [161]. To avoid this, the use of indirect measurement methods is proposed [162]. The estimation of the force based on internal sensors has been widely reported in the literature [163,164]. Albrecht et al. [165] overcame these drawbacks by using a capacitance displacement sensor and a Kalman filter to estimate the cutting force until 1 kHz in milling.

Additionally, it is not easy to measure the vibration generated by the cutting force exactly at the cutting point while the cutting process is taking place. In the case of low frequency modes, the dynamic properties are related to the flexibility of the machine tool structure and therefore it can be considered that there are not changes in the dynamics when using different tools [23]. Therefore, the response can be measured in the closest non-rotating points without any appreciable precision reduction [166]. Nevertheless, this is an important difficulty to apply cutting force measurements-based methods to obtain the dynamics associated to cutting tools and toolholders. The development of tools and toolholders with embedded accelerometers can solve this issue in the near future [167].

Finally, the cutting process introduces forces in three directions that are not independent and present strong harmonics related to the spindle speed and tooth passing frequency that are jeopardizing the synthesis of the FRFs.

Opitz and Weck [166] and Minis et al. [168] used randomly distributed channels for random excitation approach. They measured system compliance through random cutting force excitation by recording both force and response signals, they built theoretical stability lobes out of these measurements and finally they correlated them with actual cutting tests. Deviation with respect to the compliance measurements through impact hammer excitation was observed and parameters like preload, direction of preload, feed rate or damping were addressed as the main causes of this deviation.

Liu et al. [169] used the ARMAX module of Simulink® (Matlab®) to perform the experimental modal analysis. They measured the force by means of a 3-component dynamometric plate and the response by means of an accelerometer in real-time.

Later, Özsahin et al. [170] used two different approaches. First, they obtained the FRF for discrete frequency points at specific cutting speeds, measuring the force and response at the corresponding speed. Secondly, in order to avoid the harmonic content problem of the cutting forces, a specially designed workpiece with randomly distributed channels was cut. This way a random excitation of the system was achieved. Both approaches showed significant changes in system dynamic parameters in comparison with the standard hammer test.

Other authors developed similar techniques to obtain dynamic parameters from a random cutting force excitation [171–173]. They pre-machined a workpiece and performed cutting tests to apply a random excitation. Li et al. [171] reported a damping decrease with respect to hammer test results.

All these approaches based on random excitation neglect or suppress the excitation created by the spindle rotation and the tooth passing harmonics.

Instead of using a random excitation, Aguirre et al. [174] used the harmonic excitation of a milling process to develop a method to excite the structure by means of the cutting force harmonics. Thus, a kind of chirp excitation is achieved considering all the possible

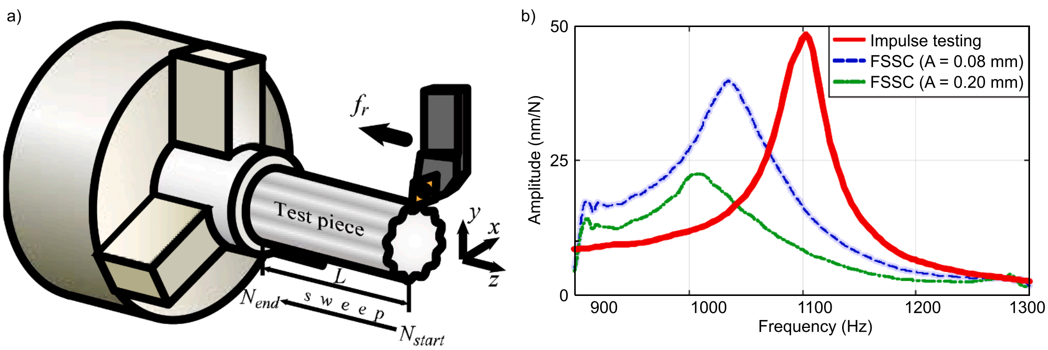

Fig. 8. a) Schematic diagram of Fast swept sine cutting (FSSC) test and b) experimental compliance comparison between impulse testing and FSSC test [176].

nonlinear effects of the moving and cutting machine tool. While Aguirre et al. excited a flexible fixture by means of this method and no changes with respect to the hammer test FRF were observed, Iglesias et al. [175] applied the same technique over a milling machine and observed a considerable variation with respect to hammer test. They combined the measurements of three different cutting tests to define a linearly independent system and obtain the different FRFs. Takasugi et al. [176] used a similar approach in a turning machine, applying a variable speed over a sinusoidal surface to create harmonic excitation and concluded that their swept sine cutting excitation resulted in FRFs with lower natural frequency and compliance (see Fig. 8). This method offers a great potential because it can obtain the real response of the machining process, although it is difficult to apply to systems with changeable dynamics with the machine position.

4.5. Digital image correlation

Digital Image Correlation (DIC) is a 3D full-field, optical technique which can be used to measure vibration on any kind of structure. High-resolution and high-speed cameras can be used to record the response of structures and extract dynamic parameters through an output-only methodology. This emerging technology can be adapted to conventional EMA and OMA techniques and presents a high potential due to its clear advantages with respect to conventional EMA and OMA methods: no need of complex cabling and sensor setup, measurement of a vast grid of points at a time, avoids mass loading and does not need a previous knowledge of the geometry to be tested. However, there are other drawbacks which will need to be addressed in the future to make it worthwhile and extend its use. These drawbacks are related to the measurement process itself (hidden parts of the structural components to analyze, number of cameras needed to catch the 3D movement, initial calibration, surface preparation, lighting needs…) and to the processing and postprocessing algorithms (edge detection, motion tracking method, resolution at high frequencies…).

Trebuna and Hagara [177] applied DIC for the experimental analysis of simple plates and compared it to a conventional EMA with laser vibrometers, with a good correlation between the two. Later, a similar approach was adopted to successfully analyze and compare with EMA the modal parameters of several cutting tools [178] and a small machine [179]. In a recent study, Law et al. [179] used output only modal analysis (OOMA) in order to extract the modal parameters of machine tools using visual vibrometry. They implemented mass change methods to extract the mode shapes to be able to identify the FRFs. Visual vibrometry technique was developed relying on a high frame rate camera and a smart-phone camera.

In order to leverage the benefits from conventional EMA analysis and digital image correlation, Bregar et al. [180] proposed a hybrid approach to reconstruct the full-field FRFs from the combination of the high-speed-camera measurements and the classic accelerometer measurements, using a dynamic substructuring approach, increasing the reliability and consistency of the result.

Motion amplification technique [181] has been developed around DIC technique. It consists of a video-processing method that detects subtle motion through high speed cameras and amplifies that motion to a level visible to the naked eye. The objective is to analyze vibrations of the system, similarly to the classic Operation Deflection Shape (ODS) technique but with the advantages of a full video recording of the specimen.

5. Conclusions

The dynamic behavior of a machining system limits the productivity of metal cutting processes. Therefore, an accurate machine dynamic characterization is essential to properly describe the dynamic response of the machine or predict the stable cutting conditions. However, it has been demonstrated that current conventional dynamic characterization methods are often not reliable enough to be used as valuable input data. These static methods present strong limitations since they do not consider the machine tool in its natural state, that is, traveling, rotating, cutting…, where non-linear effects play a major role in the dynamic response.

For this reason, alternative experimental methods have been developed to increase the quality of the obtained data by characterizing the machine in its operating conditions. In this paper, a review of the alternative methods developed in the literature with the aim of dynamic characterization has been carried out. These methods have been posed to overcome the limitations found with conventional methods, about which important inaccuracies have been reported.

The new mechatronic devices and non-contact shakers proposed to excite rotating tools are appropriate when the dynamics to consider are related to the tool, toolholder and spindle. The non-contact magnetic shakers have a big potential to measure rotating tool FRFs and automatize the calculation of stability diagrams for high-speed applications.

The inverse identification can be used instead when the general dynamic response is needed. The obtained results can be very accurate for reproducing the real behavior of the machine, since they are based in direct stability measurements, but the extensive experimental testing campaign required and the fact that it is only valid for simple and dominant dynamics systems limit its applicability. Further development of these techniques is needed to identify complex and non-linear dynamics.

The operational modal analysis is a simpler approach, where dynamics can be studied while the machine is operating. However, the dynamic information provided by this method is limited due to the absence of measurement of the excitation force. This method is especially useful for mechanical systems where the excitation forces cannot be measured.

In the case of machine tools, the operational modal analysis is not that interesting because the real cutting force excitation method can be measured in operational conditions thanks to dynamometric tables and toolholders. Therefore, the excitation and the response can be measured while cutting and different FRFs can be calculated using different techniques. Although it is very sensitive to machine position, it has an enormous potential in the industry 4.0 age. However, it has limitations for the measurement of tool dynamics due to the difficulties for measuring the vibration close to the tool tip during cutting.

Lastly, the emergent digital image correlation technique could become a breakthrough in the near future, as the vision technology evolves. The technique requires a complex set-up and lighting, and does not have enough maturity for its industrial application in medium–high frequency ranges and moving machine elements.

Although the alternative experimental methods are usually improving the accuracy of the dynamic data, their particularity makes them only suitable for specific situations, they require a more complex setup and they are normally more time consuming. The current trends in machine digitalization and intensive data gathering will be soon leveraged to overcome the current limitations of alternative experimental methods. Instead of needing an expert who determines the machine dynamics by means of a rudimentary manual method, the autonomous intelligent machine will be capable of performing this task more accurately based on the alternative methods described in this paper.

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Declaration of Competing Interest

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

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