Determination of the Cutting Performance and Dynamic Behaviour of Universal Milling Machines by means of a Stability Model

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Abstract
Self-excited or chatter vibrations are well known among milling machine builders and users. Nowadays, chatter is one of the most important restrictions of the milling process. Self-excited vibrations prevent obtaining the required accuracy in workpiece, reducing the lifetime of the cutter and the mechanical components of the machine. Due to chatter vibrations, the dynamic stiffness of the structure of the milling machine limits the productivity in steel and cast iron roughing. Hence the stability of this kind of process is an indirect way to measure the dynamic stiffness of the design. This paper presents different stability models and proposes the most suitable for the determination of the cutting performance of universal milling machines. Finally the cutting performance and the dynamic behaviour of the milling machine is determined combining the measurements of the dynamic stiffness of the structure, the stability models, cutting tests and modal analysis.

1 Introduction

The self excited vibrations keep on being a classic problem that limits productivity of milling processes. Chatter vibrations appear in many different ways in the milling process. The prediction and suppression techniques vary depending on the process. For example, in high speed aluminium rough milling, the modes limiting the stability are associated to the tool and the toolholder, or to the spindle (the chatter frequency is roughly between 200 and 3000 Hz). However in steel face milling carried out by big milling machines the critical modes are related to whole machine tool structure (the chatter frequency is roughly between 15 and 150 Hz).

There are several architecture types applied in the design of the milling machine structure. However, when a wide workspace is required, big changes in the dynamic stiffness of the structure appear. Therefore the cutting performance varies significantly with the position of the cutting point in the case of face milling of steel. The presence, in some cases, of several modes with similar dynamic stiffness in the range where the instability grows adds additional difficulties in the determination of the critical modes.

Some attempts to determine and normalise the dynamic behaviour and cutting performance of milling machines can be found in the literature. These proposals did not take advantage of all the capabilities of the stability lobes to describe the dynamic behaviour of the machine.

For machine tool designers, it should be important to know how machine tool structure limits the cutting capabilities of a universal milling machine. The stability lobes combined with other test can be used to define the performance of the machine. Taking into account the relationship between the flexibility of the structure and the face milling operation of steel and cast iron, the first step should be the selection of the most suitable model for this kind of operation.

2 State of the art.

Pioneers like Taylor [1] were the first ones affronting “the most obscure and delicate of all problems facing the machinist”. Tobias and Fishwick [2], and Tlusty and Polacek [3], stated that the main reason for self-excited vibrations is the regeneration of the chip thickness and the mode coupling. Merrit [4] designed a closed-loop scheme, which made chatter easier to be understood. Nevertheless, almost all theoretical and experimental researches were only focused on continuous cutting processes.

The milling process simulation is more complex than the other machining processes due to the discontinuous nature of cutting process –there are usually several cutting edges machining at the same time- and due to nonlinearities of this type of process. In addition, the cutting forces vary in magnitude and direction in milling process, affecting the vibrations to the chip thickness. Some authors [5][6] replaced the time dependent cutting coefficient by a constant term to use solutions developed for turning operations.

Time domain simulations permit to analyze nonlinerarities of the process and specially the possibility that the tool may lose contact with the workpiece. During contact loss, the instantaneous chip thickness is zero, and hence the cutting force is zero. Another important nonlinearity is related to the process damping introduced by the cutting operation.

In the last two decades many research efforts have been carried out in time domain modeling approach [7][8][9]. In order to obtain stability areas, time domain techniques require simulations under a great quantity of cutting conditions which is time consuming.

Searching for an analytical solution, Minis and Yanushevsky [10] used Floquet’s theorem and Fourier series on a two-degree-of-freedom cutting model for the formulation of the milling stability. Altintas and Budak [11][12] developed a stability method, which led to an analytical determination of stability limits. It was based on a two dimensional single frequency analysis (or zero order approximation) and allowed calculating milling lobes diagram very fast. Later, the model was extended to three dimensional cases trying to predict the stability of face milling processes [13][14][15][16].

These frequency domain models suppose that the vibration has only a dominant harmonic (single frequency model or zero order approximation) and this simplification allows a semi analytical solution to the problem similar to the case of the continuous orthogonal cut. These models offer good results with great immersions of the mill near slotting cut, but as the cut becomes more interrupted the results differ from the reality, mainly due to the presence of additional lobes (flip bifurcation) that sum to the traditional ones (Hopf bifurcation) limiting the stability [17].

The stability of these interrupted cuts has been determined by means of alternative methods. Davies et al., [17] used a discrete map model for highly interrupted milling processes, where the time in the cut is infinitesimal and the cutting process is modelled as an impact. Bayly et al., [18] obtained similar results using temporal finite elements. Later, Insperger and Stépán developed the technique of semi-discretization [19][20].

The same problem can be approached within the frequency domain considering that vibrations are composed by multiple harmonics of the tooth passing frequency from a generic chatter frequency. These multifrequency models [21][22][23] produce additional lobes that become more important as the milling becomes more intermittent.

Face milling of steel is usually carried out using big mills, high number of inserts and high radial immersion. Therefore, the milling process is far to be interrupted. Considering these aspects and the differences between methods (see table 1), a tridimensional single frequency method can provide fast and accurate results in this kind of operation.

Attempts to establish standard tests in order to evaluate the cutting capacity of milling machines has repeated cyclically in the last 40 years. According to Tlusty [24], the first attempt dates of 1968 when a cooperative work supported by USAF was made between the Universities of Cincinnatti, Birmingham, Aachen and Munich.

METHOD	***ADVANTAGES***	***DRAWBACKS***
SINGLE FREQUENCY MODEL / ZERO ORDER APPROXIMATION	• Fast prediction of Stability Lobes. • Direct use of FRF-s.	• Imposible to model non-linearities. • Bad predictions in highly interrupted milling processes.
MULTIFREQUENCY MODEL	• Good predictions in highly interrupted milling processes. • Direct use of FRF-s.	• Imposible to model non-linearities. • The speed of the simulation depends on number of harmonics.
SEMI DISCRETIZATION	• Nonlinear complex force formulations can be considered. • Robust method.	• Modal parameters needed. • Slower than single frequency model.
TIME DOMAIN INTEGRATION	• Modelling of non-linearities. • Detailed simulations (surface finishing, roughness,…).	• Slow predictions. • Modal parameters needed.

Table 1: Compararison between main stability models

The next important step was in 1971 when Tlusty and Koegnisberger [25] proposed several tests to verify machine tools. The CIRP also supported a cooperative work in this sense between 1965 and 1975. In this work participated illustrious investigators like J. Boothroyd, T. Hoshi, F. Koegnisberger, J. Lombard, J. Peters, B. Stone, J Tlusty and M. Weck.

Some companies and organizations that have defined their own cutting standard tests. For instance, tests defined by a set of Swedish companies, ENIMS tests defined in Soviet Union and finally tests defined by German lathe builders (Vereinigte-Drehmaschinen-Fabriken) have been remarked in bibliography [24].

It is necessary to emphasize the work done by WZL of Aachen in this field. They published several works ([26], [27]) where some tests are proposed to determine the dynamic stiffness of the machine-tool structures and its influence on chatter free machining limit.

The most serious attempt to estandarize the definition of normalized cutting test to define the capacity of a machine tool has been the ANSI/ASME B5.54-1991 standard [28]. This proposal was led by professor Tlusty [24]. Two levels of verification are defined and great part of the variables that influence the stability of the machine are defined. This norm keeps on the investigation carried out by the group of Tlusty in the field of optimization of high speed machining of aluminium parts. Therefore, the proposal focuses in the field high speed machining and the aplication in great milling machines with structural chatter problems is not straightforward. Uriarte et al [29] tryed to cover this field propossing cutting test for this type of milling machines divided in different levels.

In general, it is necessary to remark that although several attempts are found in the bibliography, there is not any standard that allows to define the capacity of cut of big milling machine in whole workspace. That is a real demand of many manufacturers and customers.

Taking into account the relationship between steel face milling operation and structural chatter, the tridimensional single frequency model will be adopted to use in the characterization of the milling machine cutting capabilities [30].

3 Three dimensional single frequency stability model.

Cutting forces can excite vibration modes in any direction, and consequently a superficial undulation in the workpiece is produced and the chip thickness will vary.

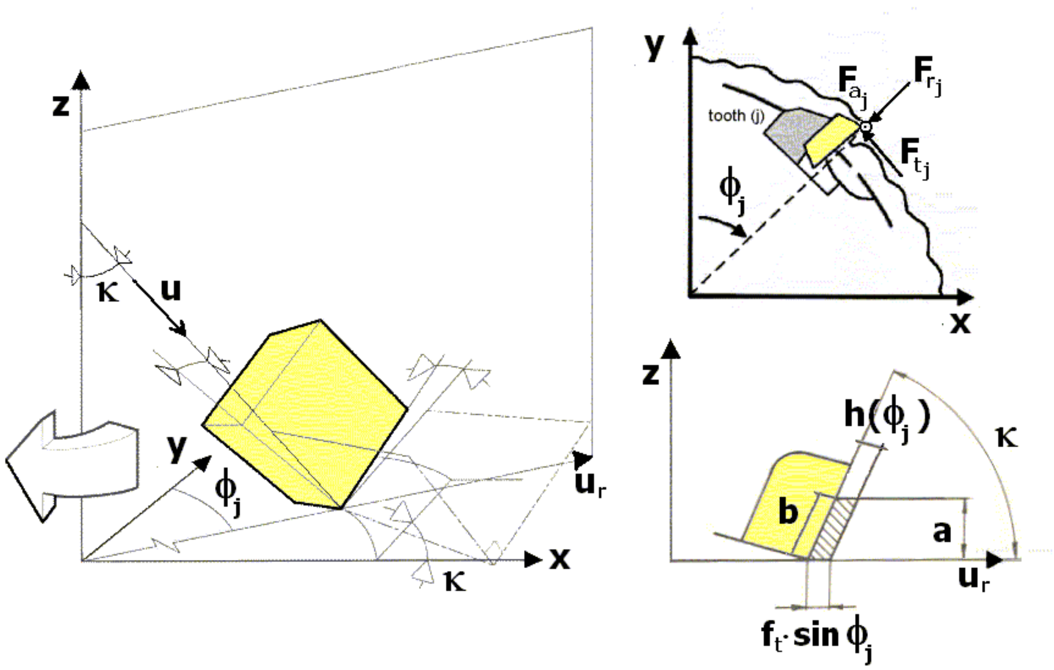

Figure 1: Geometry of the face milling.

3.1 Force model.

Although several attempts to study the stability with nonlinear force models can be found [30], in this work a linear force models will be used. This model differences two type of forces: cutting forces proportional to the chip area (h·b) and edge forces related to the ploughing effect and proportional to the length of edge (S) ([11]). If the cutting coefficients are considered constant, they do not describe the effect of the feed rate (f_t) or other cutting process parameters.

$$\begin{Bmatrix} Ft_j \\ Fr_j \\ Fa_j \end{Bmatrix} = \left\{ K_{tc} \cdot \begin{Bmatrix} 1 \\ K_{rc} \\ K_{ac} \end{Bmatrix} \cdot h(\phi_j) \cdot b + \cdot \begin{Bmatrix} K_{te} \\ K_{re} \\ K_{ae} \end{Bmatrix} \cdot S \right\} \cdot g(\phi_j) \tag{1}$$

g(ϕ_j) is a unit step function that determines if the *j*th tooth is cutting or not. The radial immersion angle can be defined for the teeth number *j* as, $\phi_j=\phi+(j-1)\,\phi_p$.

3.2 Dynamic chip thickness.

It is necessary to define the vibration *(h_d)* in the direction of the chip thickness depending on the position of the flute *j*. The chip thickness consists of static part (h_s), and a dynamic component (h_d). Considering the influence of the lead angle κ, and tooth passing period *T*, the chip load can be written:

$$h_j(\phi_j) = (h_s + h_d)\cdot g(\phi_j)$$
$$h_s(\phi_j) = f_t \cdot sin(\phi_j)\cdot sin\kappa$$
$$h_d(\phi_j) = \{u\}^T \cdot \{\Delta r\} = \{sin(\phi_j)\cdot sin\kappa \quad \cos(\phi_j)\cdot sin\kappa \quad -\cos\kappa\}\cdot\{\Delta x \quad \Delta y \quad \Delta z\}^T \tag{2}$$

Considering:

Δx=x(t)-x(t-T), Δy=y(t)-y(t-T), Δz=z(t)-z(t-T)

3.3 Dynamic milling forces.

Considering equation (2) and neglecting the static chip thickness, the tangential (F_{tj}), radial (F_{rj}) and axial (F_{aj}) forces can be expressed as,

$$\begin{Bmatrix} Ft_j \\ Fr_j \\ Fa_j \end{Bmatrix} = K_T \cdot b \cdot\cdot \begin{Bmatrix} 1 \\ K_R \\ K_A \end{Bmatrix} \cdot \{u\}^T \cdot \{\Delta r\}\cdot g(\phi_j) \tag{3}$$

The elemental forces are projected to cartesian axes using the transformation matrix:

$$\begin{Bmatrix} Fx_j \\ Fy_j \\ Fz_j \end{Bmatrix} = \begin{bmatrix} -\cos(\phi_j) & -sin(\phi_j) & 0 \\ sin(\phi_j) & -\cos(\phi_j) & 0 \\ 0 & 0 & 1 \end{bmatrix} \cdot \begin{Bmatrix} Ft_j \\ Fr_j \\ Fa_j \end{Bmatrix} = [Q_1]\cdot \begin{Bmatrix} Ft_j \\ Fr_j \\ Fa_j \end{Bmatrix} = \tag{4}$$

$$\{F_j\} = \cdot g(\phi_j)\cdot [Q_1]\cdot K_T \cdot b \cdot \{k\}\cdot \{u\}^T \cdot \{\Delta r\} \tag{5}$$

$$\{F\} = K_T \cdot b \cdot \left[\sum_{j=1}^{Z}\left(g(\phi_j)\cdot [Q_1]\cdot\{k\}\cdot\{u\}^T\right)\right]\cdot\{\Delta r\} = K_T \cdot b \cdot [A(t)]\cdot\{\Delta r\} \tag{6}$$

Rearranging the expressions, new cutting force equation is obtained. Different directional coefficients of the matrix [A] depend on the position of the mill, the lead angle and the relation between the cutting coefficients. The stability analysis will be explained roughly and more details can be obtained in the bibliography [11][12]. Both, milling forces and [A(t)] matrix are periodic at tooth passing frequency T=2π/Ω. Thus [A(t)] can be expanded into a discrete Fourier series.

The number of harmonics (*m*) to be considered to an accurate solution depends on relationship between the immersion and the number of cutting edges. Two solutions can be obtained: Single Frequency Solution and Multi Frequency Solution, [11][21].

$$[A(t)] = \sum_{m=-\infty}^{\infty}[A_m]\cdot e^{i\cdot m\cdot\Omega\cdot t} \text{ where } [A_m] = \frac{1}{T}\int_0^T [A(t)]\cdot e^{-i\cdot m\cdot\Omega\cdot t}dt \tag{7}$$

3.4 Single-frequency solution.

The milling dynamic force is approximated to the average component of the Fourier series expansion [A_o]. It can be demonstrated that the matrix [A_o] is time-invariant, but depends on radial immersion and cutting coefficients.

$$[A_o] = \frac{1}{T}\int_0^T [A(t)] \cdot dt = \frac{Z}{2\pi}[\alpha(\phi)] \tag{8}$$

Thus, the interesting point is that now the average directional cutting coefficient matrix [α] is time independent. The single frequency chatter model assumes that the regenerative vibrations are dominated by a single chatter frequency ω_c. Therefore {Δr} can be represented as:

$$\{\Delta r(t)\} = \{r(t)\} - \{r(t-T)\} = (1 - e^{i \cdot \omega_c \cdot T}) \cdot \{r(t)\} \tag{9}$$

The vibration vector {Δ(t)} and the cutting forces {F(t)} are related by the frequency response matrix [Φ(iωc)], which represents the dynamics of the system. Using this matrix, the characteristic equation of the closed loop dynamic cutting forces is reduced to

$$\{F(t)\} = \frac{Z}{2\pi} \cdot b \cdot K_T \cdot \left[1 - e^{-i\omega_c T}\right] \cdot [\alpha(\phi)] \cdot [\Phi(i\omega_c)] \cdot \{F(t)\} \tag{10}$$

The stability analysis drives to an eigenvalue problem where the eigenvalues ($\Lambda = \Lambda_R + \Lambda_I \cdot \mathbf{i}$) can be related with the maximum stable depth of cut and different cutting speeds.

$$\det\{[I] + \Lambda.[\alpha][\Phi]\} = 0 \tag{11}$$

$$b_{lim} = -\frac{2\pi\Lambda_R}{Z \cdot Kt} \cdot \left(1 + \left(\frac{\Lambda_I}{\Lambda_R}\right)\right) \quad -> \quad a_{lim} = -\frac{2\pi\Lambda_R}{Z \cdot Kt} \cdot \left(1 + \left(\frac{\Lambda_I}{\Lambda_R}\right)\right) \cdot sin\kappa \tag{12}$$

$$n = \frac{60}{Z \cdot T} = \frac{60 \cdot \omega_c}{Z \cdot \left\{\pi - 2 \cdot arctan\left(\frac{\Lambda_I}{\Lambda_R}\right) + 2\pi \cdot k\right\}} \quad k = 0,1,2,... \tag{13}$$

Different values for the integer value *k* yield different spindle speed. Therefore taking into account the relationship between the limiting depth of cut and spindle speeds, the procedure for computing the stability lobes diagram can be fulfilled.

3.5 Dynamics of the process.

The relatioship between the dynamic displacement and the dynamic force is defined by means of frequency response matrix defined between piece and tool [14][26]:

$$[\Phi(i\omega_c)] = \begin{bmatrix} \Phi_{xx}(i\omega_c) & \Phi_{xy}(i\omega_c) & \Phi_{xz}(i\omega_c) \\ \Phi_{yx}(i\omega_c) & \Phi_{yy}(i\omega_c) & \Phi_{yz}(i\omega_c) \\ \Phi_{zx}(i\omega_c) & \Phi_{zy}(i\omega_c) & \Phi_{zz}(i\omega_c) \end{bmatrix} \tag{14}$$

To consider the combined flexibility of the part and the tool is quite simple. It is possible to show that if the FRF are defined in the same coordinate system, the global response matrix is obtained adding the reponse functions of both sides [11]:

$$[\Phi] = [\Phi_h] + [\Phi_p] \tag{15}$$

The described stability model is developed using XYZ process coordinate system. In this system X coordinate describes the feed direction and the tool axis defines the Z coordinate (see Figure 1). However the experimental FRFs that conpose the frequency response matrix are defined according to machine tool fixed coordinate system ($X_0Y_0Z_0$).

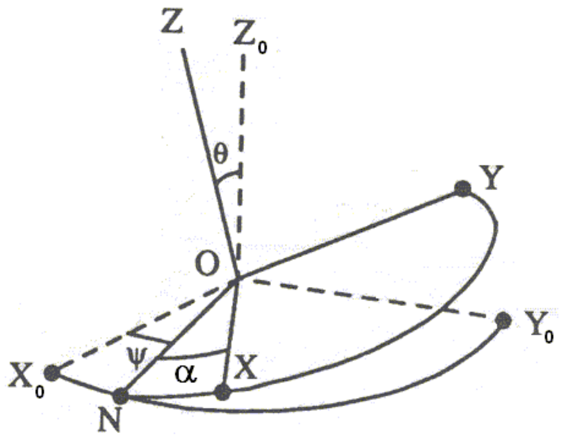

Figure 2: Geometry of the face milling.

The relationship between these coordinate systems has been defined using three angles (Euler angles). Two angles allow locating the cutting plane (XY) in the space and third one defines the feed direction in the cutting plane. Considering the rotation matrix $[Q_2]$ associated with these angles, the dynamics defined in fixed coordinate system $[\Phi_0]$ can be oriented to obtain the frequency response matrix considering process coordinate system $[\Phi]$.

$$[\Phi] = [Q_2]^{-1} \cdot [\Phi_0] \cdot [Q_2] \tag{16}$$

$$[Q_2] = \begin{bmatrix} \cos\psi \cdot \cos\alpha - sin\psi \cdot \cos\theta \cdot sin\alpha & -\cos\psi \cdot sin\alpha - sin\psi \cdot \cos\theta \cdot \cos\alpha & sin\psi \cdot sin\theta \\ sin\psi \cdot \cos\alpha + \cos\psi \cdot \cos\theta \cdot sin\alpha & -sin\psi \cdot sin\alpha + \cos\psi \cdot \cos\theta \cdot \cos\alpha & -\cos\psi \cdot sin\theta \\ sin\theta \cdot sin\alpha & sin\theta \cdot \cos\alpha & \cos\theta \end{bmatrix} \tag{17}$$

4 Dynamic characterization method.

To characterize dynamically big milling machines and to establish the limit of stability is not simple matter. There are a lot of variables that influence on stability [24][26]. The number of tests has to be optimized in order to be applied in industrial environment. In this context, the time is limited and if many variables are introduced the effect of machine dynamics could not be seen clearly. The proposed method is divided into four tests.

4.1 Test 1: dynamic stiffness.

The dynamic stiffness is measured by means of frequency response functions (FRF) obtained in machine tool cartesian axes ($X_0Y_0Z_0$). In a general case 18 FRF are required to define the dynamic stiffness of each point of the workpiece. It is convenient to neglect the effect of the flexibility of the part choosing stiff test parts. Therefore, in some machine architectures the flexibility of the side of the part can be neglected and only 9 FRFs are required.

In big milling machines the dynamic stiffness of the structure changes a lot inside the workspace. In many cases the variation of the stiffness with one axis can be refused and FRFs can be obtained only in a plane. Hence, the first step is to discretize the workspace to obtain the FRFs in each position.

The method is focused in structural chatter. Common face mills are quite stiff in the range that chatter grows in this kind of operations. Anyway, a normalized mandrel is used to obtain FRFs. This mandrel allows clamping accelerometers and shakers, and assures that the excitation and response is introduced and measured in machine tool cartesian axes($X_0Y_0Z_0$). The mandrel is a 73x73x130mm bar attached to a stardard tool holder (for instance ISO50, see Figure 4). The weight (7 kg) is similar to a 125mm diameter face milling cutter.

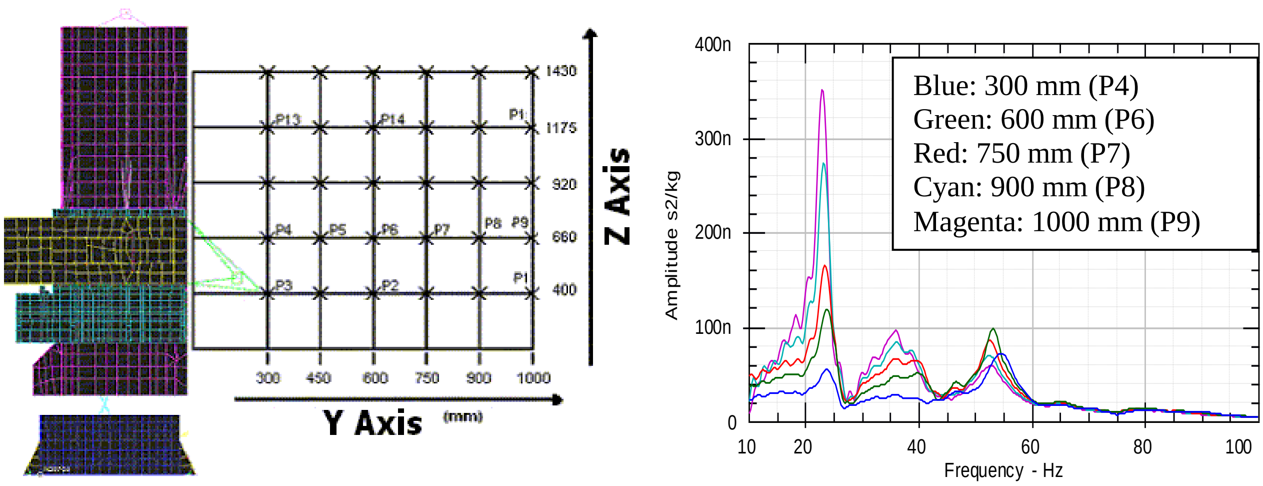

Figure 3: Discretization and Variation of Dynamic Stiffness (ZZ).

The machine tool structure is considered like a linear system [26][27], hence a hidraulic/electromagnetic shaker or dynamometric hammer can be used to obtain FRFs. The hammer is easier to apply in industrial environment taking into account that position changes are required. Some authors [13] remark the influence of the preload between part and tool in the dynamic properties. In the proposal, these effects are neglected and the structure is excited without preload. The excitation range depends on many factors (machine architecture, milling force range, workspace,..). For instance, a exitation range between 10 and 200Hz is recomended for big milling machines.

The value of dynamic stiffness can be found for direct FRFs taking into account the value of minimum stiffness in excitation range. These values can be used to compare changes suffered by the structure in the workspace. In cartesian machines these values are very important and it is usual considering direct static and dynamic stiffnesses. However, it seems more suitable to work in modal or natural coordinates. This way the minimum and maximum stiffness can be obtained considering all the possible directions and calculating the eigenvalues of frequency response matrix for each frequency of the range.

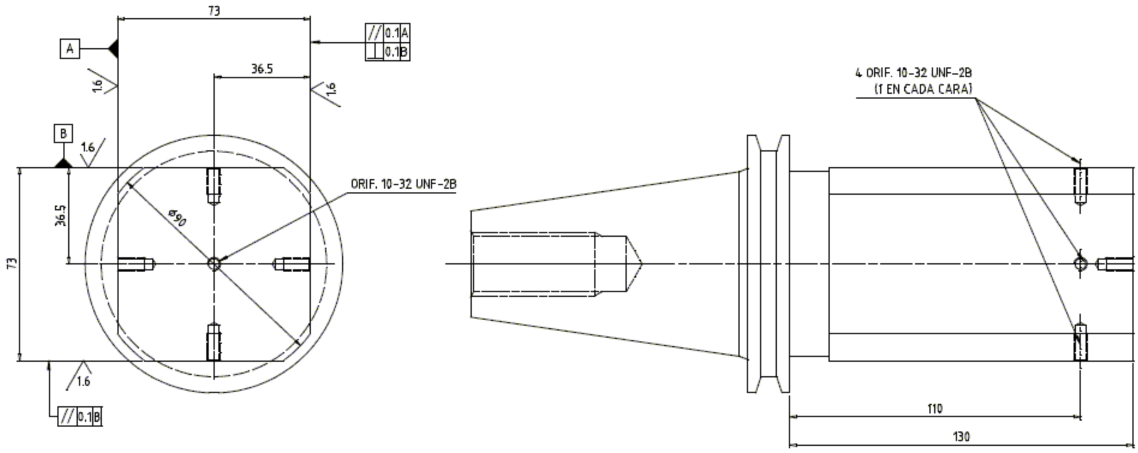

Figure 4: Normalized mandrel for ISO50.

4.2 Test 2: stability lobes.

The tridimensional single frequency method is proposed to describe the cutting capabilities of universal milling machines. The stability lobes relate maximum stable depth of cut to different spindle speeds. The chatter frequency diagram is also important, because it confirms which is the unstable mode. The stability lobes depend on dynamics of the machine, tool/part pair and cutting process parameters.

MACHINE CLASS	*MAXIMUM TORQUE [N·m]*	*TOOL*		
		DIAMETER [mm]	*Z*	*κ / LEAD ANGLE [º]*
A. VERY LIGHT	<95	50	4	45
B. LIGHT	95-185	63	5	45
C. MEDIUM	185-365	100	6	45
D. HEAVY	365-720	125	8	45
E. VERY HEAVY	>720	160	10	45

Table 2: Tool Characteristics.

- Machine Tool Dynamics: the same FRF obtained in the previous test are used to obtain stability lobes. The stability model allows using experimental FRFs directly.
- Tool Selection: Uriarte et al [29] defines the characteristic fullfilled by the tool according with the torque of the spindle head. These values (see Table 2) are adopted in this proposal.
- Cutting Coefficients: A linear model is considered (see point 3.1). The cutting coefficient depends on workpiece material/insert material-geometry pair. Therefore all the tools described by Table 2 can mount the same inserts with the same angles and geometry, and the same cutting coefficients can be used. The recomended part is a block of F1140(C45) steel and for instance, SANDVIK R245-12 T3 M-PM-4030 inserts are used mounted in SANDVIK R245-080Q27-12M. To obtain the cutting coefficients an inverse coefficient identification have to be performed [11][30]. A set of 30 cutting tests has been carried out to obtain different cutting coefficients which are shown in Table 3.

F1140	K_{tc}	K_{rc}	K_{ac}	K_{te}	K_{re}	K_{ae}
ABSOLUTE	1889N/mm^2	775.6 N/mm^2	364.3 N/mm^2	63.1 N/mm	78.2 N/mm	82.6 N/mm
RELATIVE	1889N/mm^2	0.4106	0.1928			

Table 3: Cutting coefficients for F1140/ SANDVIK R245-080Q27-12M/ R245-12 T3 M-PM-4030

- Cutting conditions: The cutting speed range is determined taking into account the cutting speed range defined by cutting tools supplier. The minimum cutting speed is defined multiplying by 0.8 the recommended minimum speed and the maximum speed multiplying by 1.2 the recommended maximum speed. For instance, for 125mm diameter tool with R245-12 T3 M-PM-4030 inserts a cutting speed range between 418 r/min and 764 r/min is proposed.

  When the feed rate is very small the cutting coefficients are increased and the stability is lower. This kind of effect can be taken into account by means of an exponential model. The feed rate is a secondary variable in stability but it has to be controlled. In this case also the recommeded feed rate range must to be considered and average one has to be chosen. In the determination of the cutting coefficients the same feed rate range has to be used. In cutting test with R245-12 T3 M-PM-4030 inserts a feed rate of 0.2 mm/flute has been adopted. Finally, dry cutting tests have been simulated neglecting the influence of the lubrication.

- Multiple simulations: Theoretically, the machine has different stability every plane of the space. If the machine has only three axes the cutting plane is automatically defined. In the case of machines with orientable spindle head, in most cases, the face milling operation is carried out in machine tool cartesian plane (X_0Y_0, Y_0Z_0 or X_0Z_0). Hence, the simulations have to be done in these planes. In each plane two kind of operations have to been simulated:

  1. Simulation set 1: A slotting operation is considered in all the directions of a plane. In this way the worse and the best directions can be found and milling direction (up or down milling) can be neglected.
  2. Simulation set 2: The real face milling conditions are reproduced. A radial immersion of 80% is considered in both milling directions (up and down milling). For each milling direction, only feed directions coincident with the machine tool axis are simulated.

  The simulations are performed for each position in the workspace. For any considered cutting position, cutting plane and feed rate direction the stability lobes are obtained, and minimum and maximum values of depth of cut are collected with the related spindle speeds, feed directions and chatter frequencies. Representing these values in the workspace, the variation of the cutting capabilities of the milling machine can be described (see Figure 6) and the critical modes can be identified.

4.3 Test 3: cutting test.

Several cutting tests have to be carried out to compare experimental results with predictions of the stability lobes. The main characteristics of the test are summarized in the next points:

- Reproduced simulations: It is difficult to compare with simulation set 1 with experimental results. Some authors [26][29] propose cutting test in form of circular interpolations. These results can be compared with simulation set 1 but experimental results can differ with simulations because when big face mill cutters are used the dynamic stiffness can change a lot during cutting tests. Therefore, taking into account the industrial enviroment, only simulation set 2 have to be reproduced experimentally.
- Workpiece: The proposed workpiece is a steel solid block. The idea is to divide the block in different zones. In each zone a different depth of cut is machined [24]. The lenght of the zone has to assure a

minimum quantity of 30 revolutions of the tool with all the inserts cutting (see equation 18) considering the diameter of the tool *(D)*. This way, chatter can be clearly determined by means of marks left in the surface, characteristic noise or accelerometer spectrum.

$$l = \frac{D}{2} + f_t \cdot Z \cdot 30 \tag{18}$$

- Cutting plane / feed direction: The cutting test must be performed only in one cutting plane.. When the spindle head is orientable, the cutting plane is defined considering two axes. The first one is chosen selecting the axis that produces a greater variation on dynamic stiffness. The second one, however, is the axis with lower stiffness variation, and it will be used as feed direction. Succesive tests have to be carried out in different positions of the machine moving the machine according to most flexible axis.

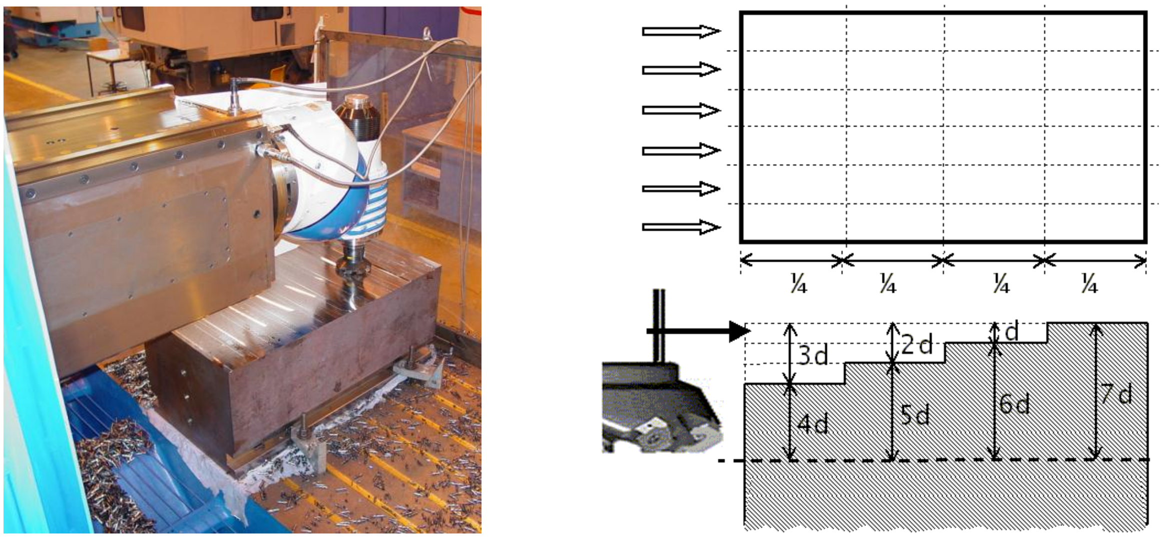

Figure 5: Cutting Tests.

- Experimental set-up: The milling machine will be sensorized locating accelerometers in points of the structure as close as possible to the spindle head. A microphone can be an alternative to accelerometers. The objective is to measure the chatter frequency in unstable cutting tests.
- Cutting conditions: the cutting speed is chosen considering the results of stability lobes. Cutting conditions related to minimum and maximum stability have to be used in tests (simulation set 2). This way the spindle speed and cutting direction is selected. Average value of feed rate is maintained constant in this test and no lubrication is used.

The experimental results will be compared with stability lobes. The reliability of the predictions can be determined and the identification of the critical modes confirmed measuring chatter frequencies.

4.4 Test 4: modal analysis.

The target of the test is to obtain the shapes and modal parameters of the critical modes. All the details of the modal analysis are not defined in the paper. There are authors that describe the basis of the technique [31][32] and the application to machine tool [5][26]. Anyway, some comments can be interesting:

- Selection of the machine position and excitation direction: The critical modes have been identified with stability lobes and experimental tests. Analyzing the FRF function in different directions and positions, the best position and excitation direction can be selected.
- Experimental set-up: vibration modes of the milling machine structure have to be modelled. The response of the structure must be measured in more than 100 points and hence the use of hidraulic/electromagnectic shaker drives to better results.

5 Application.

The method was applied to a medium size universal milling machine. The stroke of the machine is 6000x1000x1400 in $X_0Y_0Z_0$ coordinate system. The method demands the manipulation of a great amount of data. Therefore, a software application has been implemented based on MATLAB 7.0.

POSITION		*FRF XX*		*FRF ZZ*	
Z axis (mm)	*Y axis (mm)*	*Frequency (Hz)*	*Dynamic stiffness (N/μm)*	*Frequency (Hz)*	*Dynamic stiffness (N/μm)*
400	*300*	43	16,8	26,25	18,0
	600	42,5	11,5	26	9,0
	1000	40,75	4,1	24	3,2
660	*300*	47,25	18,1	23,75	17,8
	600	44	9,8	24	8,3
	1000	40,5	4,2	23	2,8
1175	*300*	40,25	15,2	22,25	19,0
	600	42	8,2	22	9,8
	1000	40,25	3,7	21,5	4,5

Table 4: Dynamic stiffness measurements

- Dynamic Stiffness: The workspace has been discretized and different FRFs obtained using the test device. The results have been summarized in Table 4. Four modes have been iddentified in different FRFs obtained betwen 0 and 200Hz. However, the presence of two dominant modes must be remarked. The first one has a main proyection on Z axis and its frequency varies from 26 to 21Hz. (see Figure 3) The second one has a main component in X direction when the averhang of the ram is bigger than 600. Its frequency fluctuates between 40 y 44Hz.

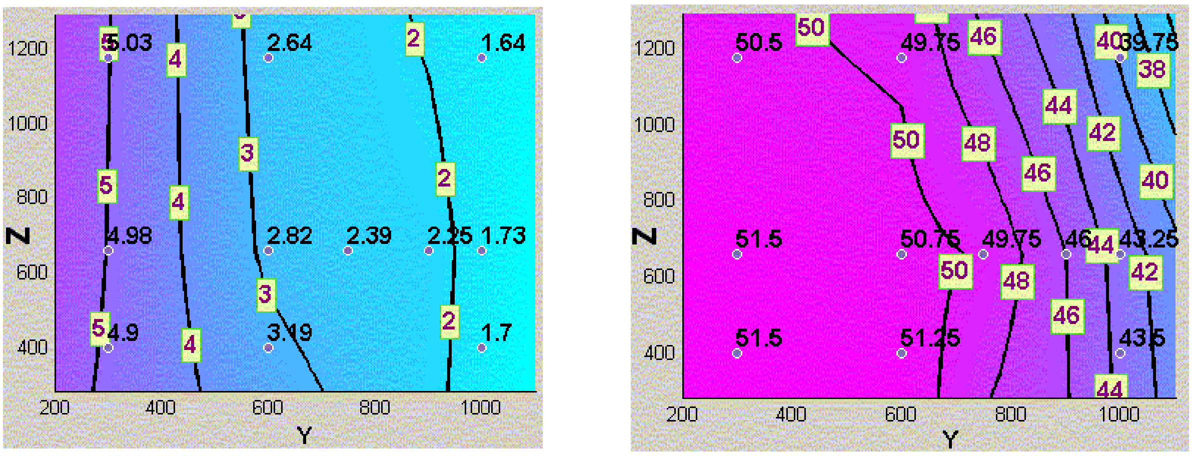

Figure 6: Variation of the minimum depth of cut and related chatter frequency with the position.

- Stability Lobes: According with Table 2, a face-milling cutter with a diameter of 125mm, 8 flutes and a lead angle of 45º was selected. The simulations predicted that if any feed direction in XY plane is considered in slotting conditions (simulation set 1), the two main modes limit the stability. However, if only cartesian directions (X and Y) are considered in up and down milling direction with a radial immersion of 100mm (simulation set 2), the chatter grows due to 40-44 Hz mode in the recommended range of cutting speeds. These simulations remarked the importance of the ram cantilever on the stability.
- Cutting tests: Taking into account the results of the stability simulations two cutting conditions have been chosen. The worse stability conditions have been obtained in simulations with a spindle speed of 400 -460 r/min in up milling operation. The best stability conditions have been obtained in high cutting speeds (650r/min) in down milling operation.

  The cutting test has been confirmed that the chatter grows between 43 and 48Hz. The stability of the process has been reduced as the overhang of the ram increases. In the low spindle range, the limiting depth of cut and chatter frequencies have been predicted properly. However, some discrepancies appear in high cutting speed and the predicted maximum stability can not achieve.

Y AXIS (mm)	VC = 400 R/MIN (UP MILLING)				VC = 650 R/MIN (DOWN MILLING)			
	Simulation		Test		Simulation		Test	
	a_{lim} (mm)	$\omega_{chatter}$ (Hz)	a_{lim} (mm)	$\omega_{chatter}$ (Hz)	a_{lim} (mm)	$\omega_{chatter}$ (Hz)	a_{lim} (mm)	$\omega_{chatter}$ (Hz)
750	3.8	46.7	3.5	48Hz	5.8	48.6Hz	4.5	45,2Hz
850	-	-	3.5	46.5Hz	-	-	4	45,2Hz
900	3	44.7	-	-	5.5	48.2Hz	-	-
950	-	-	2.5	43.5Hz	-	-	4	44Hz
1000	2	42.8	-	-	5.8	47.4Hz	-	-

Table 5: Cutting Test versus Stability Lobes.

- Modal Analysis: A point with maximum overhang and medium heigh from the floor (see point 9 in Figure 3) has been selected as the most suitable one. In order to excite the main modes of the machine, the modal analysis has been carried out exciting the machine in the direction X+ and Z+. The goal is to excite the two main modes but the shape and parameters of four modes have been obtained.

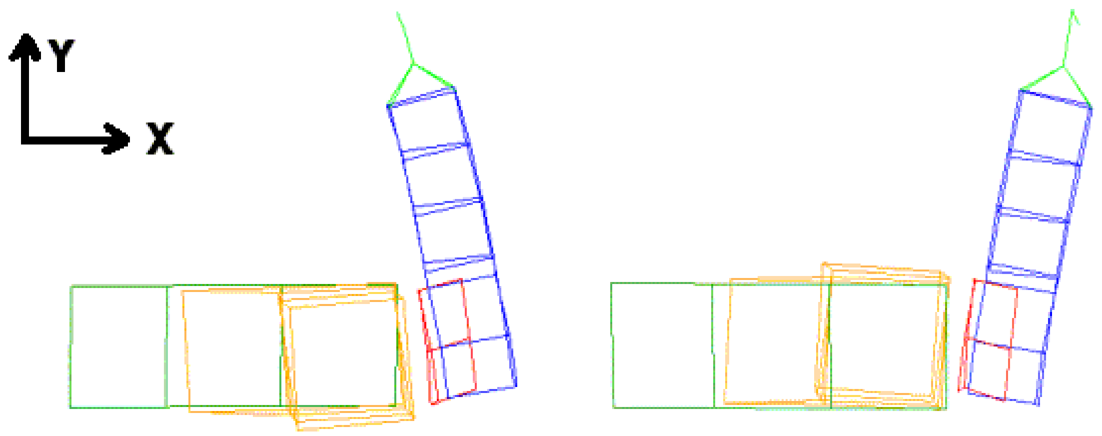

Figure 7: Critical mode 41Hz.

Studying the modal analysis, it is determined that the chatter vibrations are related to the twisting of the column and bending of the ram in XZ plane. The other critical mode (23.5Hz) is a turn respect to X linear guidance.

	FREQUENCY (Hz)	RELATIVE DAMPING(%)	BRIEF DESCRIPTION
1	23.5	3.4	Turn in YZ plane respect to X linear guidance.
2	41.0	5.7	Twisting of the column and bending of the ram in XZ plane
3	53.5	3.4	Balancing of the ram & saddle in YZ plane around Z guidance.
4	173	4.4	Twisting of the ram.

Table 6: Results of Modal Analysis.

6 Conclusions.

A method to characterize the dynamic behaviour of the universal milling machine is proposed. The method is made up by four tests. In the first one, the dynamic stiffness of the structure is obtained in whole workspace.Obtained data is used in the stability model to predict the limiting depth of cut and chatter frequency for a standard face milling operation. The stability lobes also allow determining the modes that limit the stability in the useful range of the cutting tool. Studying the bibliography a three dimensional single frequency model has been chosen to simulate the stability of the universal milling machine. These predictions are correlated with experimental test chosen considering the simulations performed. Finally using modal analysis the shape and parameters of the critical modes are obtained.

The method has been apllied to medium size universal milling machine. The presence of two main modes whose flexibility grows with the overhang of the ram has been measured. The stability simulations confirm that these two modes limit the stability. In cutting tests performed with a feed rate direction coincident with X axis only the mode of 40-44Hz produced chatter in the cutting speed range. Experimental results match the predicted chatter frequencies and minimum depth of cuts, although some differences have been obtained in the prediction of maximum depth of cuts.

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