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Optimization of Hard Material Roughing by means of a Stability Model

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Abstract
Self-excited vibrations limit the productivity of aluminium and steel roughing. During aluminium roughing, chatter is related to tool modes (high frequency), whereas in the case of steel they are associated to the structure of the machine tool (low frequency). For aluminium, the dynamic characterisation of the machine is highly dependent on the tool and not on the machining position, whereas in steel roughing the dynamics are independent of the tool and depend on the position.

Considering these differences the present research studies different stability models and proposes the most suitable for steel roughing. This model has been used in the process planning and tool selection for steel machining. This method differs from the usual one applied in light alloys. The proposed model is verified experimentally in a practical case, optimizing material removal rate, choosing the optimal tool and work conditions.

1 INTRODUCTION
Self-excited or chatter vibrations are well known among milling machine tool users. Nowadays, the chatter is one of the most important restrictions of the milling process.

Machine tool chatter vibrations avoid obtaining the required accuracy in workpiece, reducing the lifetime of the cutter and the mechanical components of the machine.

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 300 and 3000 Hz). However in steel roughing the critical modes are related to the whole machine tool structure (the chatter frequency is roughly between 15 and 100 Hz).

Stability lobes have been successfully used in aluminium roughing process planning to maximise the Material Removal Rate (MRR). Those diagrams relate the spindle speed to the chatter-free maximum depth of cut.

Taking into account the shape of the critical modes, the main dynamic properties remain constant with position changes. However, their performance change with variations in the tool and toolholder. In this kind of process, a two dimensional model obtains good predictions neglecting the dynamic flexibility in tool axis direction.

As far as the excitation range and the shape of the critical modes are different in steel and cast iron roughing, the stability lobes can not be used in the same way. In the work presented, the stability lobes will be applied to plan steel face milling.

2 LITERATURE REVIEW
A review of the state of the art has been carried out to define the most suitable stability model to steel roughing.

Chatter first basic research works were carried out in mid 40's [1]. Initially it was thought that chatter was originated by a negative damping caused by the cutting process.

Later on, 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. Merritt [4] designed a closed-loop scheme, which made chatter easier to understand and predict. Nevertheless, almost all theoretical and experimental research 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.

The first milling process modeling attempts were carried out by Tlusty [5], Sridhar et al. [6], and Opitz and Bernardi [7]. Opitz [7] applied turning stability theory to milling process stability. They replaced the time dependent cutting coefficient by a constant term, the average term over the real cutting period of each cutter.

Time domain simulations permit to analyse nonlinearities 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. This aspect introduces a discontinuous nonlinearity in the system equations. 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 [8], [9], [10], [11]. 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 [12] 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 [13] developed a stability method, which led to an analytical determination of stability limits. It was based on a two dimensional single frequency analysis and allowed calculating milling lobes diagram faster than by means of time domain models.

Later, Jensen and Shin [14], using a similar methodology to Altintas and Budak [13], presented a model that allowed simulating a three-dimensional system. This model included the axial axis of the tool and an additional cutting edge lead angle that permitted to locate the direction of the chip regeneration in the third axis. Altintas [15] extended its equations to a three-dimensional system, obtaining a similar solution applied to cylindrical inserts.

These frequency domain models suppose that the vibration has an only a dominant harmonic (single frequency model) and this simplification allows an 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 that limit the stability.

The stability of these interrupted cuts has been determined by way of alternative methods. There are authors who have obtained the stability lobes based on a temporary finite element analysis [16] or by means of the semidiscretisation method [17].

The same problem can be approached within the frequency domain considering that vibrations are comprise multiple harmonics of the tooth passing frequency from a generic chatter frequency. These multifrequency models [18], [19], [20], [21], [22] produce additional lobes that are more important as the milling becomes more intermittent.

Frequency domain models obtain stability lobes very fast, specially when single frequency models can be used. However, these models can not describe the nonlinearities of the system. The contact loss of the inserts stabilises some cuts but does not affect the location of the high stability pockets [22], [23]. The process damping becomes dominant when spindle speed is located in the fifth or higher lobes, or when the chatter frequency is about 4-5 times higher than the tooth passing frequency [23].

Face milling of steel is carried out using big mills, high number of inserts and high radial immersion. Therefore, the milling process is far to be interrupted [24]. Considering these aspects, the three-dimensional frequency domain model proposed by Altintas [15] has been implemented taking into account different feed directions.

3 THREE DIMENSIONAL SINGLE FREQUENCY STABILITY MODEL

Cutting forces can excite vibration modes in anyone of the three directions X, Y and Z, and consequently a superficial undulation in the workpiece is produced. Therefore, the chip thickness will not be constant and will vary depending on the frequency of vibration, the
spindle speed and the mode direction with respect to the chip thickness direction.

### 3.1 Dynamic chip thickness

Firstly it is necessary to define the vibration \((v_j)\) in the direction of the chip thickness depending on the position of the flute \(j\). The dynamic displacement of the system in this direction is presented in the following form:

\[ u_j = (x \sin \phi_j + y \cos \phi_j) \sin \gamma - z \cos \gamma \]  

(1)

\(\phi_j\): instantaneous angular immersion of tooth \(j\) measured from Y axis.

\(\gamma\): cutting edge lead angle.

\(j\): tooth number.

The resulting chip thickness consists of a static part, attributed to the rigid motion of the cutting tool \((c \cdot \sin \phi_j \cdot \cos \gamma)\), and a dynamic component corresponding to the vibrations caused by the displacements of the present tooth period and the previous one \((v_{j,o} - v_j)\).

The static component will not be considered because it does not affect the regenerative mechanism. Consequently, the chip thickness can be represented as,

\[ h(\phi, \gamma) = [u_{j,o} - u_j] \cdot g(\phi) \]  

(2)

\(u_{j,o}\); \(u_j\): Dynamic displacement of the cutter at the previous and present tooth periods.

The function \(g(\phi)\) is a function that determines whether the tooth is cutting or not. Replacing \(u_{j,o}\) and \(u_j\) by the displacements in X, Y and Z, the chip thickness is

\[ h(\phi, \gamma) = [\Delta x \sin \phi + \Delta y \cos \phi \sin \gamma - \Delta z \cos \gamma] \cdot g(\phi) \]  

(3)

where:

\(\Delta x = x(t) - x(t-T)\)

\(\Delta y = y(t) - y(t-T)\)

\(\Delta z = z(t) - z(t-T)\)

\(x(t), y(t), z(t)\) and \(x(t-T), y(t-T), z(t-T)\) represent the relative dynamic displacement between the theoretical and real cutting trajectory.

### 3.2 Modeling of the dynamic milling forces

A linear force model is considered. The three components of force are proportional to the chip area.

\[ K_t, K_r, K_a\] are the cutting coefficients.

Hence, if the forces \((F_r, F_a, F_t)\) are projected on an insert of the cutter, in the XYZ Cartesian axes:

\[ \begin{bmatrix} \Delta F_r \\ \Delta F_a \\ \Delta F_t \end{bmatrix} = K_t \cdot a \begin{bmatrix} K_c \\ K_r \\ K_a \end{bmatrix} \cdot \sin(\phi) \cdot \sin(\gamma) + \begin{bmatrix} K_c \\ K_r \\ K_a \end{bmatrix} \cdot \sin(\gamma) \cdot \cos(\gamma) \cdot \sin(\phi) \]  

(4)

Adding the cutting forces contributed by all teeth:

\[ Fx = \sum_{j=0}^{N-1} Fx_j \quad ; \quad Fy = \sum_{j=0}^{N-1} Fy_j \quad ; \quad Fz = \sum_{j=0}^{N-1} Fz_j \]  

(5)

where \(\phi_j = \phi + j \phi_p\)

\(\phi_p\): cutter pitch angle, \(\phi_p = 2\pi/N\)

Rearranging the expressions in matrix form, new cutting force equation is obtained. In which the different directional coefficients of the matrix \([A]\) depend on the position of the mill, the lead angle and the relation between the cutting coefficients.

\[ \begin{bmatrix} Fx \\ Fy \\ Fz \end{bmatrix} = a \cdot K_i \cdot [A] \cdot \begin{bmatrix} \Delta x \\ \Delta y \\ \Delta z \end{bmatrix} \]  

(7)
Considering that either $[A]$ matrix or $\Delta$ vector change with time and angular velocity, the expression becomes,

$$\{F(t)\} = a \cdot K_i \cdot [A(t)] \cdot \{\Delta(t)\}$$

(8)

where: $\{\Delta(t)\} = \{\Delta x \ \Delta y \ \Delta z\}^T$

Since the matrix $[A(t)]$ is periodic over a tooth period $T=2\pi/\omega$, it is expanded into Fourier series:

$$[A(t)] = \sum_{r=-\infty}^{\infty} [Ar] \cdot e^{-ir\omega t}$$

(9)

where $[Ar] = \frac{1}{T} \int_{0}^{T} [A(t)] \cdot e^{-ir\omega t} \ dt$

(10)

If the expression is approximated to the average term, a new matrix $[a]$ formed by the average directional coefficients is obtained. This matrix is time invariant and only depends on the radial immersion of the mill.

$$\{F(t)\} = \frac{N}{4\pi} \cdot a \cdot K_i \cdot [a] \cdot \{\Delta(t)\}$$

(11)

3.3 Stability of the process. Solving the eigenvalue problem

The vibration vector $\{\Delta(t)\}$ is presented as,

$$\{\Delta(t)\} = \{1-e^{-i\omega t}\} \cdot [G(i\omega)] \cdot \{F(t)\}$$

(12)

where the matrix $[G(i\omega)]$ represents the frequency response function relative to the contact area of the tool-workpiece. The $x$ direction is the feed direction.

$$[G(i\omega)] = \begin{bmatrix} G_{xx}(i\omega) & G_{xy}(i\omega) & G_{xz}(i\omega) \\ G_{yx}(i\omega) & G_{yy}(i\omega) & G_{yz}(i\omega) \\ G_{zx}(i\omega) & G_{zy}(i\omega) & G_{zz}(i\omega) \end{bmatrix}$$

(13)

The dynamic system of the cutting forces is reduced to:

$$\{F(t)\} = \frac{N}{4\pi} \cdot a \cdot K_i \cdot \{1-e^{-i\omega t}\} \cdot [a] \cdot [G(i\omega)] \cdot \{F(t)\}$$

(14)

The development of the equation leads to a problem of eigenvalues and therefore the system stability limit is determined by means of a characteristic equation:

$$\det \{[I] + \Lambda_\omega [\Phi]\} = 0$$

(15)

where $[\Phi] = [a] \cdot [G(i\omega)]$

The eigenvalues of the system are obtained from a cubic equation, independent of the number of the machine tool modes. Hence, the problem with some assumptions leads to an analytical solution.

3.4 Relation between the chatter frequency and the axial depth of cut limit

For each chatter frequency supposed ($\omega_c$) the three eigenvalues ($\Lambda = \Lambda_i + \Lambda_k$) can be obtained, which can be complex. Replacing these values, the limit of the axial depth of cut for a certain chatter frequency $\omega_c$ is obtained. Considering that the depth of cut has to be a positive real value:

$$\Lambda_i \cdot (1-\cos \omega_c T) - \Lambda_k \cdot \sin \omega_c T = 0$$

(16)

Defining $\kappa = \frac{\Lambda_i}{\Lambda_k} = \frac{\sin \omega_c T}{1-\cos \omega_c T}$

These simplifications yield,

$$a_{lim} = -\frac{2\pi \Lambda_k}{N \cdot K_i} \cdot (1+\kappa^2)$$

(17)

3.5 Relation between chatter frequency and rotational speed of the cutter

Since,

$$\kappa = \frac{\Lambda_i}{\Lambda_k} = \tan \psi = \frac{\cos(\omega_c T/2)}{\sin(\omega_c T/2)}$$

$$= \tan \left(\frac{\pi}{2} - \left(\omega_c T/2\right)\right)$$

(18)

The phase shift between present and previous vibration marks $\psi$ is

$$\psi = \frac{\pi}{2} - 2\psi$$

(19)

Therefore, if $k$ is the integer number of full vibration waves between teeth, then

$$\omega_c \cdot T = \varphi + 2\pi \cdot k$$

(20)

Hence, the spindle speed $n$ is calculated as,

$$n = \frac{60}{N \cdot T} = \frac{60 \cdot \omega_c}{N \cdot (\varphi + 2\pi \cdot k)}$$

(21)

Several spindle speeds are obtained depending on an integer value $k$ for each chatter frequency.

3.6 Determination of the stability lobes diagram

If the critical depths of cut are related to the speeds corresponding to the same chatter frequency and sweeping this last frequency, it is possible to obtain the stability lobes [25].
4 CUTTING PROCESS CHARACTERISATION

The milling forces are calculated linearly composed of two elements: the cutting force proportional to the chip area (Fc) and the ploughing force proportional to the depth of cut (Fc). This last force corresponds to the force generated ploughing the surface (null chip thickness). This component is usually neglected for the stability analyses, since it does not depend on the chip thickness.

The milling forces for each tooth in the radial, tangential and axial directions are,

\[ F_r(\phi) = K_{rc} \cdot a \cdot h(\phi) + K_{re} \cdot a \]
\[ F_t(\phi) = K_{tc} \cdot a \cdot h(\phi) + K_{te} \cdot a \]
\[ F_a(\phi) = K_{ac} \cdot a \cdot h(\phi) + K_{ae} \cdot a \]

(22)

The cutting coefficients must be obtained, in order to determine correct cutting forces. Steel and cast iron face milling operations are carried out using tools with inserts. To identify the cutting constants a mechanistic identification method has been used [26]. This method is based on average cutting forces.

The average cutting forces in the three Cartesian axes are obtained by means of full immersion cutting tests for different feed rates. The measurement of the cutting forces (axial, radial and tangential) is done using a dynamometer. Analytically, the average cutting coefficients for full immersion milling can be expressed as,

\[ \bar{F}_x = -\frac{N \cdot a}{4} \cdot K_{rc} \cdot c - \frac{N \cdot a}{\pi} \cdot K_{re} \]
\[ \bar{F}_y = -\frac{N \cdot a}{4} \cdot K_{tc} \cdot c + \frac{N \cdot a}{\pi} \cdot K_{te} \]
\[ \bar{F}_z = -\frac{N \cdot a}{\pi} \cdot K_{ac} \cdot c + \frac{N \cdot a}{2} \cdot K_{ae} \]

(23)

N: Number of teeth.
a: Axial depth of cut.
c : Feed per tooth.

Then, if the results of the feed-dependent average cutting forces are printed in a chart, a straight line is obtained. The factors multiplying the independent variable c compose the slope of the obtained straight line, whereas the independent term is represented by the intersection of the straight line with the OY axis. Solving the equations with the experimentally obtained slope and independent term, the values of the cutting coefficients \( K_c \) and \( K_e \) are obtained. In this work, the cutting coefficients shown in the Table 1 were obtained with SANDVIK-Coromill R245-125Q40-12M tool and R245-12T3M-PH 4030 inserts cutting F1140 (C45) steel.

<table>
<thead>
<tr>
<th>Steel F1140</th>
<th>( K_{tc} )</th>
<th>( K_{vc} )</th>
<th>( K_{ac} )</th>
<th>( K_{te} )</th>
<th>( K_{re} )</th>
<th>( K_{ae} )</th>
</tr>
</thead>
<tbody>
<tr>
<td>Absolute (N/mm²)</td>
<td>1883</td>
<td>725</td>
<td>468</td>
<td>97.6</td>
<td>186</td>
<td>147</td>
</tr>
<tr>
<td>Relative</td>
<td>1883</td>
<td>0.385</td>
<td>0.249</td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

Table 1: Steel F1140 cutting coefficients for the defined cutting tool.

These cutting constants depend on the workpiece material and characteristics of the insert (geometry, material, wear,…). Hence these coefficients can be used to simulate the cutting forces for many tools with different diameters and number of inserts, milling the same workpiece material and mounting the same inserts.

5 MACHINE DYNAMICS CHARACTERISATION

The machine dynamics must be characterised in order to introduce it in the stability model. In this case the workpiece is supposed to be totally rigid in roughing processes. The flexibility of the piece can be taken into account easily.

Considering these simplifications, it is only necessary to obtain the frequency response functions (FRF) of the cutting tool to characterise the system response at different frequencies.

The stability of the cut depends on the dynamics of the machine structure, which involves dependence on the machine position [27]. Therefore, each machine position will need its stability lobes, so that, in each position nine FRF would be needed to characterise machine dynamics. The FRFs are obtained by means of an impact hammer and a triaxial accelerometer.

These modes usually have big modal masses, and therefore the FRFs are independent of the tool. Hence the same FRF can be used to obtain the stability lobes of different tools.

The feed direction also has a strong influence on the stability of these operations. If local coordinate system (xyz) is defined considering
the x axis as the feed axis and z as the axial direction of the tool, and the dynamics of the system are defined by means of FRFs $[G^0]$ obtained in machine coordinate system XYZ, the FRF matrix in local coordinate system can be obtained defining the rotation matrix between the two coordinate systems $[Q]$

$$[G] = [Q] \cdot [G^0] \cdot [Q]^T$$

(24)

6 PROCESS PLANNING.

When the machine tool user requires a face milling process planning of a steel block, a high number of tools with different diameters, number of tooth and inserts are available.

The FRF functions and cutting constants obtained for each position can be used to obtain the stability lobes of different tools. With these charts it is possible to consider the stability of the process in the election of the tool and maximise the MRR choosing the most suitable tool and spindle speed.

For instance, the face milling of the steel block will be studied. The block will be mechanised in the machining center described in figure 3. In this case the feed direction coincides with the machine X axis ($[G]=[G^0]$). The ram travel and the vertical position (height over table) of the tool were maintained constant for the straight cut tests in X-axis. The machine is driven by linear motors in X direction, so that considering the shape of the modes, it is possible to affirm that during the cutting test the dynamic parameters of critical vibration modes do not vary. Therefore, in this machine the FRF are independent of the X axis position.

Figure 3: Horizontal machining center.

The same stability lobe can predict the stability of tools with different diameters. The lobes depend on radial immersion angles, number of inserts, position and cutting constants. Hence the figure 4 shows a stability lobe valid for three different tools, milling F1140 steel in down milling with a 80% radial immersion and 8 inserts characterised in table 1.

Figure 4: Steel roughing process planning with stability lobes.

Depending on the recommended cutting speeds for the inserts, the available spindle speed range varies according to the cutting tool diameter.

It is possible to predict the optimal speeds for each tool and the chatter-free maximum depth of cut using this kind of charts.

The Figure 4 shows that the selection of a 125 mm diameter tool takes advantage of the maximum stability area, allowing 5 mm depth of cut without chatter. However, the chatter limit is roughly on 3 and on 2 mm for the 80 mm and 200 mm cutting tools, respectively.

With the same FRF and cutting constants, stability lobes for tools with different number of inserts and radial immersion angles can be calculated. For instance, if the number of inserts increases, the asymptotic borderline of stability decreases [4] and the high stability zones move to the left side of the diagram.

7 EXPERIMENTAL VALIDATION

In order to validate experimentally the model, a 125 mm diameter tool (SANDVIK-CoromillR245-125Q40-12M) with eight inserts (R245-12T3M-PH 4030) has been used. The selected radial immersion was 100mm in down milling (80% of radial immersion).
The experimental tests made to validate the model are shown in the figure 5. The results determine that the obtained lobes match the laboratory tested. Nevertheless some differences are detected. Even though, lobes predict reasonably well the sweep spot, the point where the maximum depth of cut is obtained (540 rpm and 5 mm). The area of minimum stability is also well defined to 2 mm.

![Figure 5: Experimental validation of the stability lobes diagram.](image)

The experimental results match the predicted lobes. The stability lobes also allow determining the modes that limit the stability in the useful range of the cutting tool.

8 CONCLUSIONS

The dynamic stiffness of the structure of the milling machine limits the productivity in steel and cast iron roughing, due to chatter vibrations.

A three dimensional single frequency model has been chosen to simulate the stability of steel roughing. It has been implemented introducing the influence of the feed direction.

The fact that the FRF only depends on machining position allows simulating stability lobes for different tools and cutting conditions with the same FRF.

For instance, face milling of a steel block has been planned taking into account the stability of the process by means of stability lobes.

The experimental results match the predicted lobes. The stability lobes also allow determining the modes that limit the stability in the useful range of the cutting tool.

9 REFERENCES


