

F.E.M. OF THE DRILLING MACHINE - TOOL INCLUDING THE GYROSCOPIC EFFECT

Kessentini A.^{*,**}; Chevalier G.^{**}; Louati J.^{*}; Rivière A.^{**} & Haddar M.^{*}

^{*}Mechanics Modelling and Production Research Unit (U2MP),
University of Sfax, Sfax Engineering School (E.N.I.S.), B.P 3038 - Sfax - Tunisia

^{**}Laboratoire d'Ingénierie des Structures Mécaniques et des Matériaux,
Institut Supérieur de Mécanique de Paris (Supmeca), 3 rue Fernand Hainaut, 93 407 Saint
Ouen Cedex

E-Mail: amir_kessentini@yahoo.fr

Abstract:

In this paper, a new theoretical approach for dynamical study in drilling operation is proposed. We use a finites element of beam design to develop a dynamical model for drilling machine - tool. This model takes in to account of the gyroscopic effect and its influence on the vibrations phenomenon appearing in drilling operation. It has also a big influence on the quality of machining surfaces and on the dispersion of the machine - tool when little deformations are taken in account. As a result of this modelling, initially modal analysis is carried out for the study of two cases: the simple model of the machine structure and a model with mass modifications. In the end of this paper, we present a simulation of the gyroscopic effect and the speed effect on the cutting forces fluctuation in drilling, axial and radial vibrations, the deformation energy and the hole circularity.

Key Words: Drilling vibrations, Modelling, Machine tool dynamics, Vibratory behaviour, Finite element

1. INTRODUCTION

Drilling is one of machining processes by chip removal the most widespread. It generates vibrations, which influence the quality and the cost of the product like over the machine - tool life. The interactions drill - part is the main cause of the chatter vibration phenomenon but not the only. It was essentially affected by the entire machine - tool dynamic. Existing dynamic models in the literature are limited to the cell of machining for the vibrations study (tool - part system consideration). Dilley and Bayly in [1, 2] uses an existing dynamic model that includes acceleration and velocity terms, and then adds an end support of increased spring stiffness to model the margin and hole interaction as hole depth increases. They also make experimental results of the effect between the chisel point and workpiece contact on the chatter frequency in non - piloted hole drilling. Tooraj [3] was developed a dynamic model to obtain the limit of stability for the bending vibration mode. Pirtini and Lazoglu in [4] was developed a mathematical model based on the mechanics and dynamics of the drilling process is developed for the prediction of cutting forces and hole quality. In [5] Juhchin A Yang and al. presents a new computer simulation model for drilling and reaming processes. The model is made of four parts: the force model for the cutting lips, the force model for the chisel edge, the dynamic model for the machine tool (including the cutter) and the regenerative correlation between the force and machine tool vibration. Gupta and Al. [6] have developed a dynamic deflection model that incorporates the effect of the alignment errors that arise when a drill is held in the spindle assembly. Jochem C. Roukema, Yusuf Altintas [7] are presented a time domain model of the torsional - axial chatter vibrations in drilling. The model considers the exact kinematics of rigid body, and coupled torsional and axial vibrations of the drill. The tool is modeled as a pretwisted beam that exhibits axial and

torsional deflections due to torque and thrust loading. A mechanistic cutting force model is used to accurately predict the cutting torque and thrust as a function of feedrate, radial depth of cut, and drill geometry.

Dynamic design aims at obtaining desired dynamic characteristics in machines and structures, which may include shifting of natural frequencies, desired mode shapes and vibratory response. The conventional dynamic design is basically hit and trial method in which many researchers [8, 11, 15] try to achieve desired dynamic characteristics by making several prototypes. The disadvantage of this technique is that actual design cycle takes a lot of time and therefore it is not cost effective. However, it is very important to simplify the drilling machine model in the purpose to reduce the time of calculation. We develop a theoretical dynamic design in which the dynamic parameters change in time.

R.S. Bais et al [8] were making a FE study of the drilling machine in the static way. They develop updated FE models of a drilling machine using analytical and experimental results. These updated FE models have been used to predict the effect of structural dynamic modifications on vibration characteristics of the drilling machine. Two stages have been carried out on the machine. In the first stage, modal tests have been carried out on a drilling machine using instrumented impact hammer. For analytical FE modelling of the machine, they was developed a computer program. Analytical FE model has been updated, with the help of a program, which has been developed using direct methods of model updating. In the second stage, they make modal testing using random noise generator and modal exciter. Global method has been used for modal identification. They used I-DEAS software to do the analytical FE modelling. Updating of the analytical FE model has also been done using the above software. The updated FE models, obtained from both the stages have been used for structural dynamic modifications (SDM).

Ewins [9] and Maia and Silva [10] have explained the basic concepts of modal testing, which is an experimental approach to obtain mathematical model of a structure. In a modal test, the structure under test is excited either by an impact hammer or by a modal exciter, and the response of the structure is recorded at several experimental points, in the form of frequency response functions (FRFs), using a dual channel FFT analyzer. The experimental modal model gives information about the natural frequencies, corresponding mode shapes and modal damping factor and is useful for model updating. The model updating techniques helps them to bring analytical finite element models closer to real systems. In model updating an initial analytical FE model constructed for analyzing the dynamics of a structure is refined or updated using test data measured on actual structure such that the updated model describes the dynamic properties of the structure more correctly. The inaccuracies in FEM, when applied to dynamic problems are due to uncertainties in boundary conditions and structural damping etc.

Sestieri [11] has discussed SDM application to machine tools and engines. Kundra [12] gave the method of structural dynamic modification via models. Modak [13, 14] has discussed SDM predictions using updated FE model for an F-structure. He used constrained nonlinear optimization method for updating of a machine tool using stiffness parameters at the boundary [15].

In addition to factors like drill geometrical parameters, drill mounting error, gyroscopic effect created by the rotation of the spindle - drill assembly can have a significant effect on the dynamic of the machine tool system.

The main objective of this paper is to study the drilling dynamics by F.E.M. including the machine - tool vibrations and the drilling forces. The first part of this work presents a mathematical development and F.E. discretization of the machine - tool. This part is completed by a modal analysis of the considered system in the purpose to determine the natural frequency. We make also a comparison between the simple model and the model with mass modifications. These mass modifications are introduced in the goal to have a model near to the reality.

The influence of the gyroscopic effect on the cutting forces fluctuations has to be investigated. Also, we present the influence of this phenomenon on the radial and axial vibrations of the drill.

2. MODELLING OF THE ENTITY (SPINDLE - DRILL)

2.1 General assumptions

The structure of the drilling machine is very complicated with the various assemblies and the accessories. Consequently, exact modelling and the analysis of the real structure are difficult and require an enormous computing time. However, for a dynamic study by finite element, one considers the simplified model of the system machine - tool - part. This model uses the finite element of beam for the unit (pillar - arm - spindle - drill). A pillar type of drilling machine was used for this study.

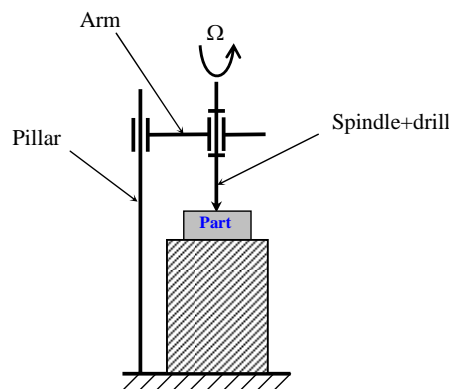


Figure 1: Pillar drilling machine.

2.2 Modelling of the entity (spindle – drill)

The (spindle - drill) entity is the only element of the considered structure which is in a rotating movement. It is modelled by a beam of section circular and characterized by its kinetic energy and its deformation energy, its behaviour compared to a fixed reference frame $R(\vec{X}, \vec{Y}, \vec{Z})$ results from the superposition of the following effects:

- 1.) movements of the small deformations due to the elasticity of the drill:
 - Traction and compression along the axis \vec{z} : displacement w .
 - Flexion around the \vec{x} axis (in *the yz plan*): displacement v and rotation θ_x .
 - Flexion around the \vec{y} axis (in *the xz plan*): displacement u and rotation θ_y .
 - Torsion θ_z around the \vec{z} axis.
- 2.) Rotation around the axis \vec{z} :

$$\theta = \Omega t \quad (1)$$

Ω is the spindle speed.

- 3.) A rigid movement body characterized by the displacements X_D , Y_D and Z_D following the three directions and being carried out compared to a fixed reference mark $R(\vec{X}, \vec{Y}, \vec{Z})$.

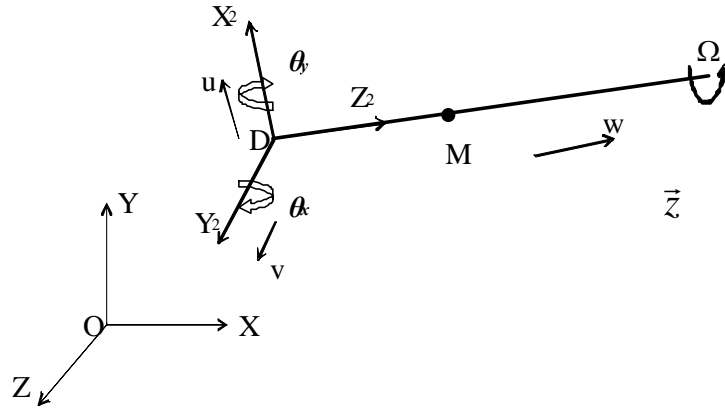


Figure 2: Frames orientations.

This figure defines the orientations of the rotating referential related to the drill:

- $R_3(\vec{X}_2, \vec{Y}_2, \vec{Z}_2)$ compared to the fixed referential $R(\vec{X}, \vec{Y}, \vec{Z})$.
- $R(\vec{X}, \vec{Y}, \vec{Z})$ referential fixes related to the frame.
- $R_3(\vec{X}_2, \vec{Y}_2, \vec{Z}_2)$ referential related to the drill.

If D is the extreme point of the entity (spindle - drill), and M is an arbitrary point of the drill,

$$\overline{OM} = \overline{OD} + \overline{DM} \quad (2)$$

The rigid movement body of the entity (spindle - drill) is given by:

$$\overline{OD} = X_D \vec{X} + Y_D \vec{Y} + Z_D \vec{Z} \quad (3)$$

Before deformation (in referential R_3),

$$\overline{DM} = X_{2M} \vec{X}_2 + Y_{2M} \vec{Y}_2 + Z_{2M} \vec{Z}_2 \quad (4)$$

After deformation,

$$\overline{DM}(R_3) = (X_{2M} + u) \vec{X}_2 + (Y_{2M} + v) \vec{Y}_2 + (Z_{2M} + w) \vec{Z}_2 \quad (5)$$

We suppose that X_{2M} and Y_{2M} are negligible, so we have,

$$\overline{DM}(R_2) = [u \cos \theta - v \sin \theta] \vec{X}_1 + [u \sin \theta + v \cos \theta] \vec{Y}_1 + (Z_{2M} + w) \vec{Z}_2 \quad (6)$$

$$\vec{V}(M)/R = \vec{V}(D)/R + \vec{V}(M)/R_2 + \vec{\omega}(R_2)/R \wedge \overline{DM} \quad (7)$$

$$\vec{V}(M)/R = \dot{X}_D \vec{X} + \dot{Y}_D \vec{Y} + \dot{Z}_D \vec{Z} + \vec{V}(M)/R_2 + \vec{\omega}(R_2)/R \wedge \overline{DM} \quad (8)$$

$$\vec{\omega}(R_2)/R = \dot{\theta}_Y \vec{Y}_1 + \dot{\theta}_X \vec{X} = \dot{\theta}_Y \vec{Y}_1 + \dot{\theta}_X \cos \theta_Y \vec{X}_1 + \dot{\theta}_X \sin \theta_Y \vec{Z}_2 \quad (9)$$

If we suppose also that are very small, then

$$\sin \theta_x \approx \theta_x, \sin \theta_y \approx \theta_y, \cos \theta_x \approx 1, \cos \theta_y \approx 1 \quad (10)$$

$$\vec{V} = \vec{V}(M)/R = \begin{pmatrix} \left(\begin{array}{l} \dot{X}_D + \dot{u} \cos \theta - u\Omega \sin \theta - \dot{v} \sin \theta - v\Omega \cos \theta \\ +(z + \omega)\dot{\theta}_y - \dot{\theta}_x \theta_y (u \sin \theta + v \cos \theta) \end{array} \right) \\ \left(\begin{array}{l} \dot{Y}_D + \dot{u} \sin \theta - u\Omega \cos \theta + \dot{v} \cos \theta - v\Omega \sin \theta \\ + \dot{\theta}_x \theta_y (u \cos \theta - v \sin \theta) - \dot{\theta}_x (z + \omega) \end{array} \right) \\ \left(\begin{array}{l} \dot{Z}_D + \dot{w} + \dot{\theta}_x (u \sin \theta + v \cos \theta) - \\ \dot{\theta}_y (u \cos \theta - v \sin \theta) \end{array} \right) \end{pmatrix} \quad (11)$$

With: $z = Z_{2M}$

We will neglect in this study the inertial effect due to the rotational movement (small deformations), so the speed formula becomes:

$$\vec{V} = \vec{V}(M)/R = \begin{pmatrix} \left(\begin{array}{l} \dot{X}_D + \dot{u} \cos \theta - u\Omega \sin \theta - \dot{v} \sin \theta - v\Omega \cos \theta \\ \dot{Y}_D + \dot{u} \sin \theta - u\Omega \cos \theta + \dot{v} \cos \theta - v\Omega \sin \theta \end{array} \right) \\ (\dot{Z}_D + \dot{w}) \end{pmatrix} \quad (12)$$

The kinetic energy could be written in the following form::

$$E_c = \frac{1}{2} \int_v \vec{V} \cdot \vec{V} dm = \frac{1}{2} \int_v \rho V^2 dv \quad (13)$$

$$E_c = \frac{1}{2} \rho S \int_L \left[\begin{array}{l} \dot{X}_D^2 + \dot{Y}_D^2 + \dot{Z}_D^2 + \dot{u}^2 + \dot{v}^2 + \dot{w}^2 + u^2 \Omega^2 \\ + v^2 \Omega^2 + 2\dot{X}_D \dot{u} \cos \theta - 2\dot{X}_D \dot{v} \sin \theta \\ + 2\dot{Y}_D \dot{u} \sin \theta + 2\dot{Y}_D \dot{v} \cos \theta + 2\dot{Z}_D \dot{w} \\ - 2\dot{X}_D u \Omega \sin \theta - 2\dot{X}_D v \Omega \cos \theta \\ + 2\dot{Y}_D u \Omega \cos \theta - 2\dot{Y}_D v \Omega \sin \theta - 2\dot{u} \dot{v} \Omega + 2u \dot{v} \Omega \end{array} \right] dz \quad (14)$$

3. FINITE ELEMENT FORMULATION OF DRILLING MACHINE

3.1 Model presentation

The eigenvalues and the eigenvectors will be calculated starting from the mass and rigidity global matrices of the system $[M]$ and $[K]$.

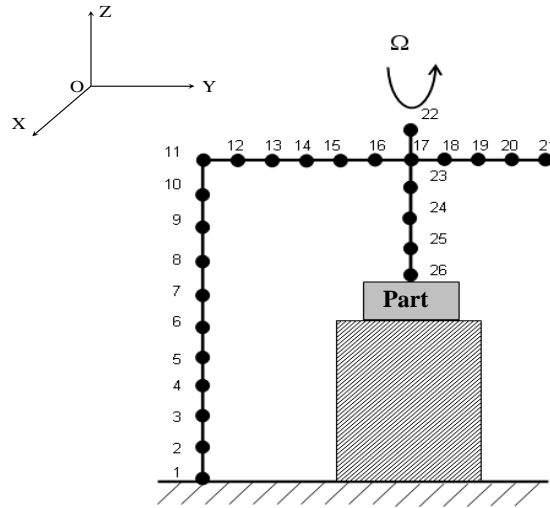


Figure 3: Drilling machine structure for FE analysis.

The relevant data used for the drilling machine is given below:

Drilling machine pillar type, height $1.4m$, length of arm $1.4m$, mass density $\rho = 7800 \text{ Kg/m}^3$, young modulus $E = 210GPa$, number of nodes=26, number of degree of freedom per node=6, number of nodes per element=2, number of elements=25.

3.2 Discretization

The drilling machine will be discretized by beam finite elements with two nodes and 6 degrees of freedom per node (Fig. 3.). The figure 4 shows the structure of the drilling machine. The eigenvalues and the eigenvectors will be calculated starting from the mass and rigidity global matrices of the system $[M]$ and $[K]$.

The vector δ of the nodal displacements is:

$$\delta = [u_i \quad v_i \quad w_i \quad \theta_{xi} \quad \theta_{yi} \quad \theta_{zi}]^T, \quad i = 1, 2. \quad (15)$$

The elastic deformation was characterized as a superposition of a number of assumed global shape functions [17]. So the displacements u , v and w are presented in function of the nodal coordinates using the following expression:

$$\begin{Bmatrix} u \\ v \\ w \end{Bmatrix} = [N]\{\delta\}, \quad \text{With } [N] \text{ is the classic matrix of interpolation function of the beam:}$$

$$[N] = \begin{bmatrix} \beta_i & 0 & 0 & 0 & \gamma_i & 0 \\ 0 & \beta_i & 0 & -\gamma_i & 0 & 0 \\ 0 & 0 & \alpha_i & 0 & 0 & 0 \end{bmatrix}, \quad i = 1, 2. \quad (16)$$

With β_i and γ_i , are the classic functions of displacement of beam in flexion, and α_i the classic functions of beam in traction - compression [18],

$$\beta_1(z) = 1 - \frac{3z^2}{l_e^2} + \frac{2z^3}{l_e^3}, \quad \beta_2(z) = \frac{3z^2}{l_e^2} - \frac{2z^3}{l_e^3} \quad (17)$$

$$\gamma_1(z) = -z + \frac{2z^2}{l_e} - \frac{z^3}{l_e^2}, \quad \gamma_2(z) = -\frac{z^2}{l_e} + \frac{z^3}{l_e^2} \quad (18)$$

$$\alpha_1 = 1 - \frac{z}{l_e}, \quad \alpha_2 = \frac{z}{l_e} \quad (19)$$

The rotations θ_x and θ_y are presented in function of the nodal coordinates using the following expression:

$$\begin{Bmatrix} \theta_x \\ \theta_y \end{Bmatrix} = \begin{Bmatrix} -\frac{\partial u_y}{\partial z} \\ \frac{\partial u_x}{\partial z} \end{Bmatrix} = \begin{bmatrix} -1 & 0 \\ 0 & 1 \end{bmatrix} [D] \{\delta\} \quad (20)$$

with

$$[D] = \frac{\partial}{\partial \xi} \begin{bmatrix} \beta_i & 0 & 0 & 0 & \gamma_i & 0 \\ 0 & \beta_i & 0 & -\gamma_i & 0 & 0 \end{bmatrix}, \quad i=1,2. \quad (21)$$

After discretization, the formulation of the kinetic energy will be in the following form:

$$\begin{aligned} E_c = & \langle \delta \rangle^T [C_e] \langle \delta \rangle + \langle \dot{\delta} \rangle^T [M_b] \langle \dot{\delta} \rangle + \langle \dot{\delta}_R \rangle^T [M_b] \langle \dot{\delta}_R \rangle + \\ & \langle \dot{\delta}_R \rangle^T [M_{cr}] \langle \dot{\delta} \rangle + \langle \dot{\delta} \rangle^T [G_b] \langle \delta \rangle + \langle \dot{\delta}_R \rangle^T [G_{cr}] \langle \delta \rangle \end{aligned} \quad (22)$$

$\delta_R = [X_{R/i} \quad Y_{R/i} \quad Z_{R/i} \quad 0 \quad 0 \quad 0]$, $i=1,2$; is the vector of the rigid body displacements.

$[M_b]$ and $[K_b]$ are the classic mass and rigidity matrix of beam.

$$[M_{cr}] = \rho S \int_0^L [N]^T [P_m] [N] dz \quad (23)$$

$[M_{cr}]$ is the mass coupling matrix between the rigid body movement and the elastic deformations.

$$[P_m] = \begin{bmatrix} 2 \cos \theta & 2 \sin \theta & 0 \\ 2 \sin \theta & 2 \cos \theta & 0 \\ 0 & 0 & 2 \end{bmatrix}$$

$$[G_{cr}] = \rho I \Omega \int_0^L [D]^T [P_g] [D] dz \quad (24)$$

$[G_{cr}]$ is the gyroscopic coupling matrix between the rigid body movement and the elastic deformations.

$$[P_g] = \begin{bmatrix} -2 \sin \theta & 2 \cos \theta \\ -2 \cos \theta & -2 \sin \theta \end{bmatrix}$$

$$[G_b] = \rho I \int_0^L [D]^T \begin{bmatrix} 0 & -1 \\ 1 & 0 \end{bmatrix} [D] dz \quad (25)$$

$[G_b]$ is the gyroscopic matrix.

$$[Ce] = \Omega^2 \begin{bmatrix} 1 & 0 & 0 \\ 0 & 1 & 0 \\ 0 & 0 & 0 \end{bmatrix} \quad (26)$$

$[Ce]$ is the matrix of the centrifugal effect of beam.

3.3. Global equation of motion

The global motion equation of all considered system (pillar - arm - spindle - drill) in the presence of external forces (cutting force) is put then in the following form:

$$[M] \{\ddot{\delta}\} + (\Omega[G] + \alpha[K]) \{\dot{\delta}\} + ([K] - \Omega^2[Ce]) \{\delta\} = \{F\} \quad (27)$$

$\{\delta\}$, $\{\dot{\delta}\}$ and $\{\ddot{\delta}\}$ are respectively the nodal vectors of displacement, speed and acceleration.

$[M]$, $[G]$ and $[K]$ are respectively the global matrices of mass, rigidity and gyroscopic, of the drilling machine (pillar - arm - spindle - drill).

$[Ce]$ the matrix of the centrifugal effects

$\{F\}$ is the vector which translates the cutting force exerted on the node $N^\circ 26$. The cutting force model is the same used in [15, 16]. It depends of feed speed, chip thickness and geometrical parameters of helical drill.

$$\{F(U, t)\} = [Kf] \{\delta(t) - \delta(t - \tau/2)\} \quad (28)$$

$[Kf]$ is a coupling matrix.

$$[Kf] = \frac{2(IC_l + IC_c)}{d \tan p} \begin{bmatrix} \sin \theta \cos \theta & \sin^2 \theta & 0 & 0 & 0 & 0 \\ \cos^2 \theta & \sin \theta \cos \theta & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 \end{bmatrix} \quad (29)$$

IC_l and IC_c are the integral parts of the torque on the lips and the chisel edge[16]. They depend of geometrical parameters of helical drill and material type of the piece. “d” and “p” are respectively the diameter and the half of the drill point angle.

It is supposed that the (spindle - drill) entity has as an average diameter equalizes with 150mm.

$$l^e = 140 \text{ mm}, \rho = 7800 \text{ Kg} / \text{m}^3.$$

In addition to 26 nodes×6 degree of freedom=156, a three displacement X_D , Y_D and Z_D of the rigid body motion. Total of matrices size 159×159.

$$[M] = \begin{bmatrix} [M_{b(frame)}] & [0] & [M_{c(f)}] \\ [0] & [M_{b(s+d)}] & [M_{cr(s+d)}] \\ [M_{cr(f)}] & [M_{cr(s+d)}] & [M_{b(rigid)}] \end{bmatrix} \quad (30)$$

$[M_{b(frame)}]$ is the matrix of mass of the frame (pillar - arm), $[M_{b(s+d)}]$ is the matrix of mass of the entity (spindle - drill) relating to small displacements, $[M_{b(rigid)}]$ is the matrix of mass of the entity (spindle - drill) relating to rigid body displacement, $[M_{cr(s+d)}]$ is the matrix masses which represents the coupling between small displacement and the rigid body motion for the entity (spindle - drill), $[M_{cr(f)}]$ is the matrix masses which represents the coupling between the small displacement and the rigid body motion for the frame.

This description is the same for the stiffness and gyroscopic matrices, respectively $[K]$ and $[G]$.

$$[K] = \begin{bmatrix} [K_{b(frame)}] & [0] & [0] \\ [0] & [K_{b(s+d)}] & [0] \\ [0] & [0] & [0] \end{bmatrix} \quad (31)$$

$$[G] = \begin{bmatrix} [G_{b(frame)}] & [0] & [-G_{cr(f)}] \\ [0] & [G_{b(s+d)}] & [-G_{cr(s+d)}] \\ [G_{cr(f)}] & [G_{cr(s+d)}] & [G_{b(rigide)}] \end{bmatrix} \quad (32)$$

$$[Ce] = \begin{bmatrix} [0] & [0] & [0] \\ [0] & [Ce_{s+d}] & [0] \\ [0] & [0] & [0] \end{bmatrix} \quad (33)$$

4. NUMERICAL RESULTS AND DISCUSSION

4.1. Modal analysis

Table I show the natural frequency of the drilling machine (pillar - arm - spindle - drill). The values from 1 to 6 represent the rigid modes relating to embedding of the node N°1. We consider in this study two different structures study. The first column present the natural frequency realized for a simple beam of structure modelling. The second is realized for a structure different from the precedent by the taking into account of the masses from the gear box (masses total concentrated assumed to 300Kg in nodes 16, 17, 18, 22, 23 and 24). This is to have a dynamical characteristic near to the reality.

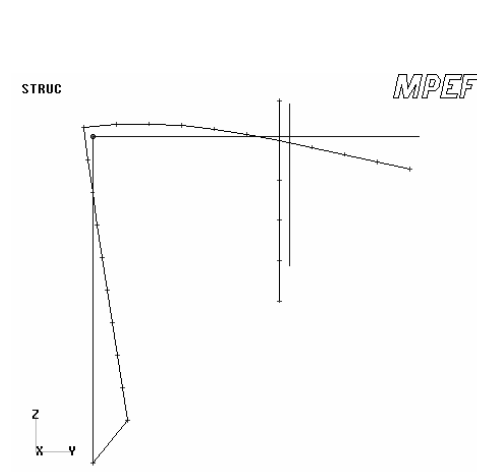
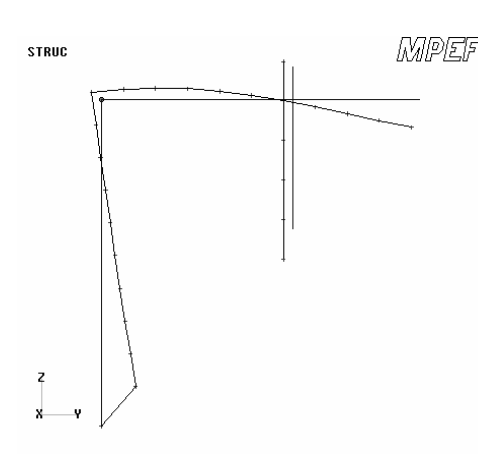
The first mode is noted for the fist $f_{1n} \approx 18 \text{ Hz}$ model of the machine - tool structure and $f_{1n} \approx 15 \text{ Hz}$ for the model with mass modifications.

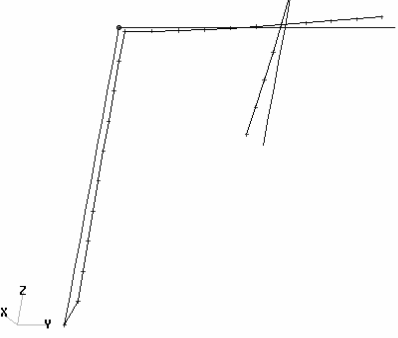
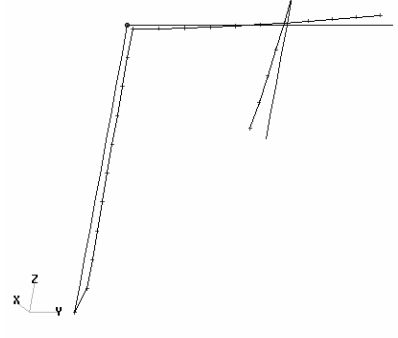
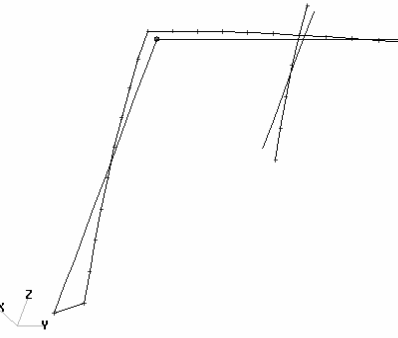
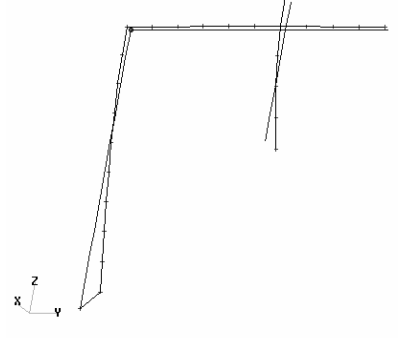
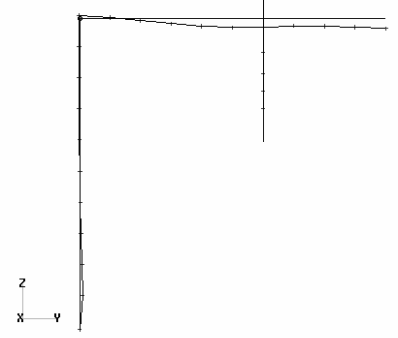
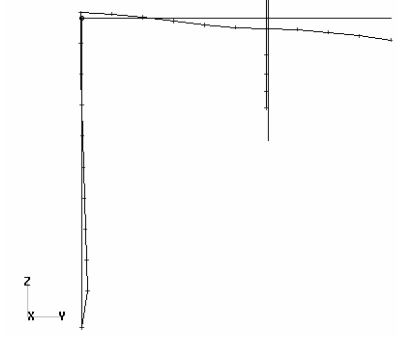
Table I: Natural frequency.

Mode N°	fn(Hz) (with mass modifications)	fn(Hz) (with mass modifications)
1 to 6	0	0
7	18.131	15.237
8	22.53	18.414
9	55.536	43.847
10	69.352	44.344
11	77.042	50.105

Next figure shows the modal shapes in the tow cases: simple beam structure and simple beam structure with mass modifications.

By using a modal analysis, we note that there is big concordances between the modal shapes in the tow cases in spite of the frequency are modified.

N° mode	Modal shapes (without mass modifications)	Modal shapes (with mass modifications)
1	 <p>F=18.131Hz</p>	 <p>F=15.237Hz</p>

N° mode	Modal shapes (without mass modifications)	Modal shapes (with mass modifications)
2	<p>STRUC MPEF</p>  <p>F=22.53Hz</p>	<p>STRUC MPEF</p>  <p>F=18.414Hz</p>
3	<p>STRUC MPEF</p>  <p>F=55.536Hz</p>	<p>STRUC MPEF</p>  <p>F=43.847Hz</p>
4	<p>STRUC MPEF</p>  <p>F=69.352Hz</p>	<p>STRUC MPEF</p>  <p>F=44.344Hz</p>

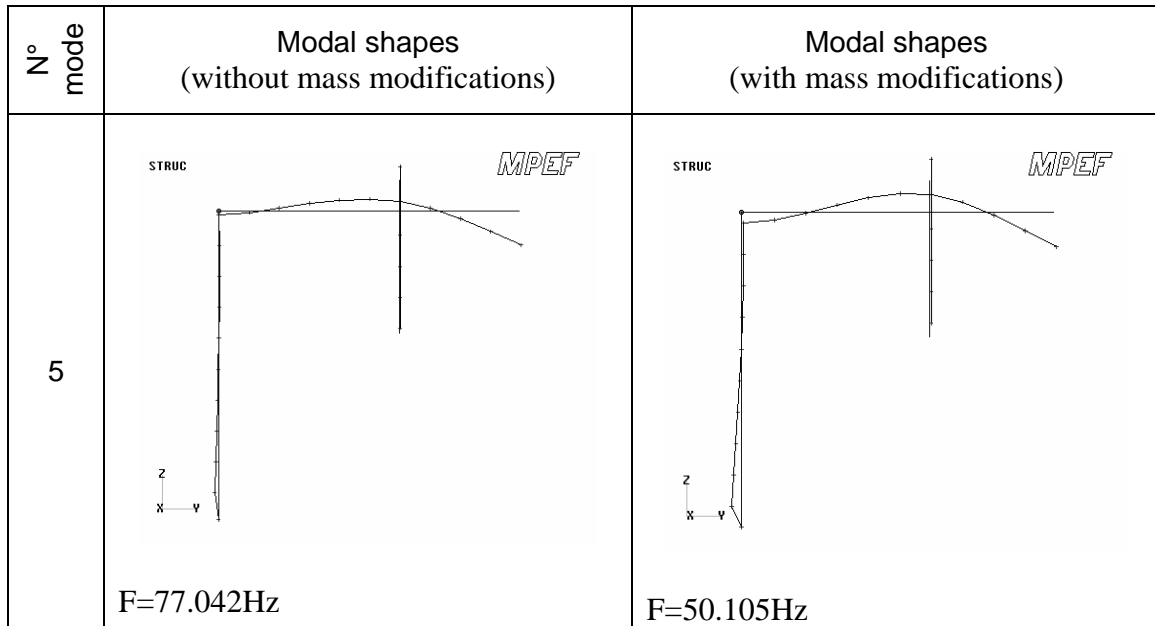


Figure 4: Modal shapes.

4.2. Vibratory analysis and dynamical behaviour

The resolution of the equation of motion (27) is carried out by using the method of the finite differences which is an explicit method of direct integration. The parameters uses for simulations are follows: the drill diameter $d = 30mm$, the feed speed $f = 0.16mm/rev$, the half of the drill point angle $p = 59^\circ$.

This resolution permit to view the gyroscopic effect on the cutting forces fluctuation evolution by simulation with two speeds, $1000rpm$ and $3000rpm$. For each speeds, we note that the fluctuation forces increases when the gyroscopic effect is taken in account.

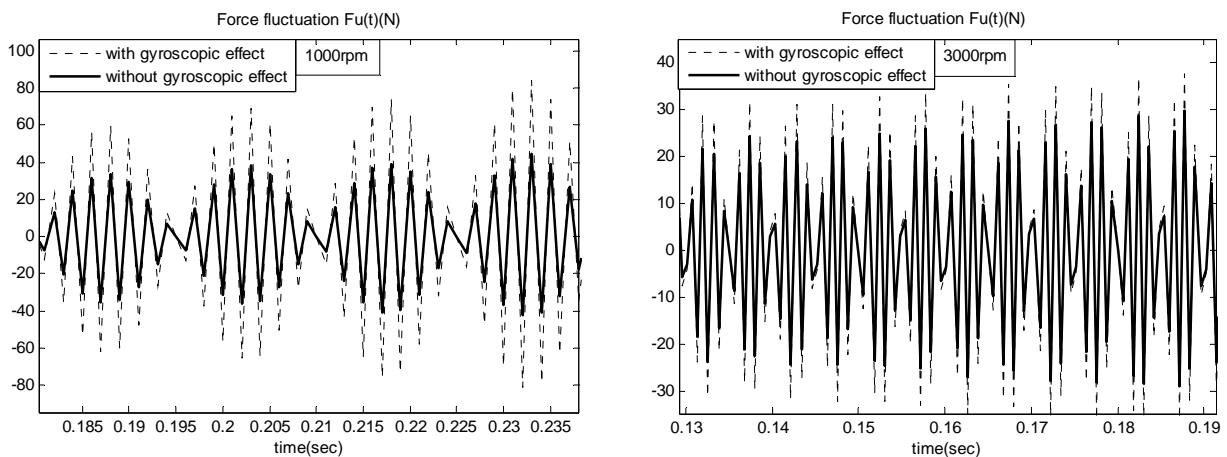


Figure 5: Gyroscopic effect on the forces fluctuations:
(a) Spindle speed 1000rpm, (b) 3000 rpm..

The simulation with spindle speed $3000rpm$ (Fig. 5b) shows that the pick to pick value of the cutting force fluctuation decreases with comparison to the simulation with speed $1000rpm$ (Fig.5a). When the chatter vibration phenomenon in drilling occurs, we find two types of

vibrations: Axial vibrations caused by the fluctuation of the cutting forces in the axial direction. Radial vibrations caused by the fluctuation of the cutting forces in the radial direction.

Next figure shows the simulation of the dynamical response in these two directions.

Furthermore, we can note that the radial vibrations are very more important than the axial vibrations.

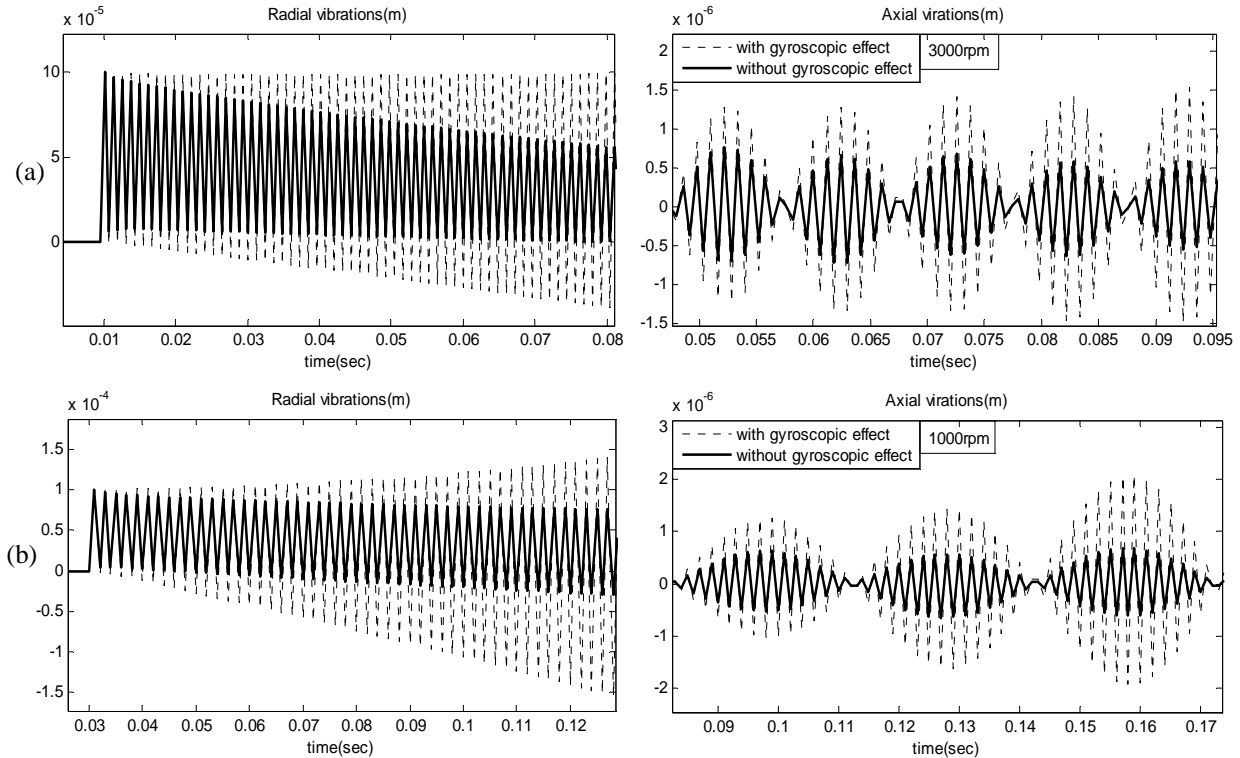


Figure 6: Gyroscopic effect on the axial and radial vibrations:
(a) Spindle speed 1000rpm, (b) 3000 rpm.

This fact is due to the weak deflection rigidity of the drill. The axial displacements shape evolution (Fig.6.) shows the waving form of the hole bottom surface. In one hand, it causes the regenerative vibrations. In the other hand, it affects more and more the vibrations in the radial direction because of the coupling who existing between the axial and radial displacements.

The deformation energy has been calculated for the three entities: drill, spindle and frame. It is clear in (Fig.7.) that the deformation energy of the frame is very weak compared to that of the drill and the spindle. For each given speed, the influence of the gyroscopic effect is significant especially when it supports less the deformation of the drill, the spindle and the frame.

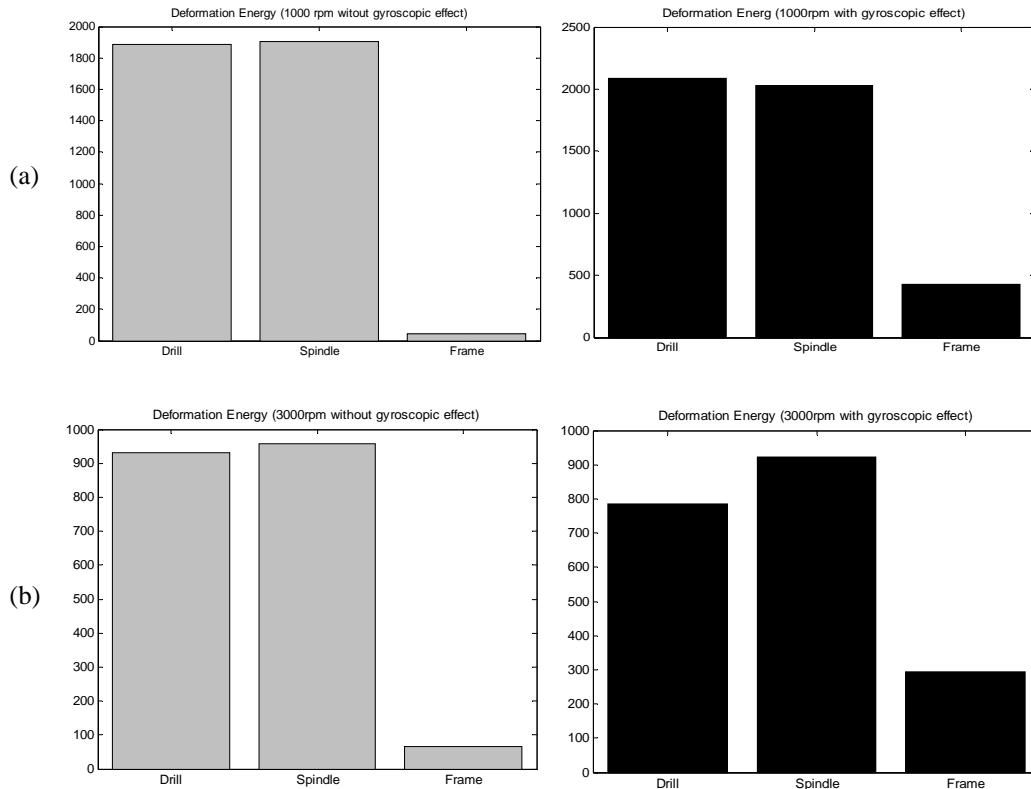


Figure 7: Gyroscopic effect on the Deformation energy: (a) Spindle speed 1000rpm, (b) 3000 rpm.

The circularity of the hole is one of some parameters who traduce the quality of the drilling operation. However, we make a simulation of the gyroscopic effect on this parameter using spindle speed equal to 1000 rpm. So, (Fig.8.) shows that the hole circularity is strongly affected by the gyroscopic effect. This simulation was carried out with dynamical response amplified 5000 time.

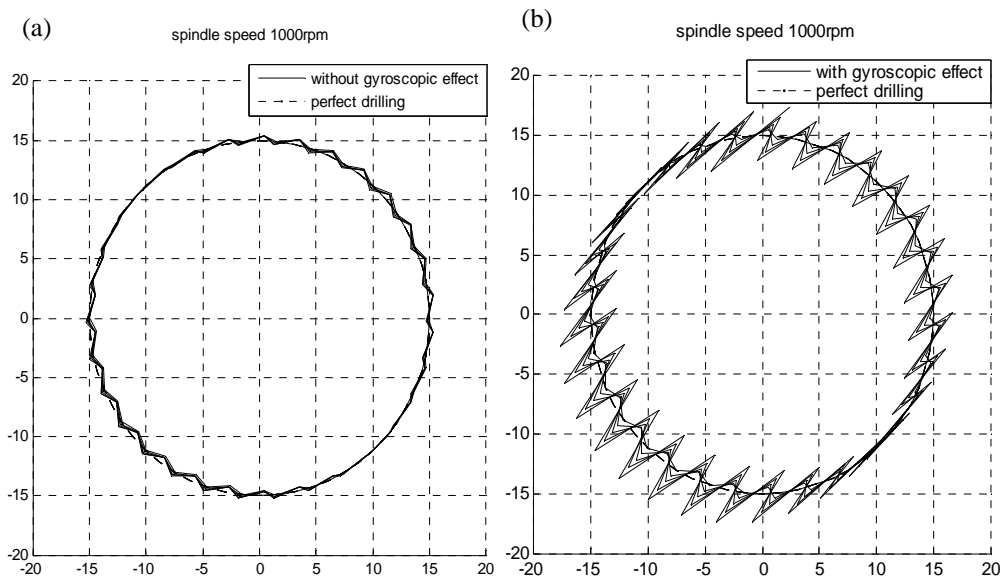


Figure 8: Gyroscopic effect on the hole circularity: (a) without gyroscopic effect, (b) with gyroscopic effect.

5. SUMMARY AND CONCLUSIONS

A dynamic drilling model for a drilling machine – tool that incorporate the effect of the gyroscopic phenomenon occurs by the rotation of the entity (spindle – drill) has been developed. This modelling was realized by using beam finite elements. The global mass matrix, rigidity matrix and gyroscopic matrix were built by taking account of the coupling existing between the little deformations and the rigid movement body of the rotating entity (spindle – drill). The resolution of the global equation of motion permit to make a vibratory investigations by simulation of the cutting forces, axial and radial vibrations, deformation energy and the hole circularity. These simulations were carried out in two cases: with and without gyroscopic effect.

The following conclusions are drawn from this work:

- The dynamical study of all the machine - tool - part system is necessary for the reason that the inertia of the rotating entity and the shape of the drilling machine structure can not be ignored.
- The gyroscopic phenomenon has a significant effect on the cutting forces and the axial and radial vibrations. Firstly, this phenomenon affects directly the vibratory behaviour of the drilling machine tool. Secondly, it decreases the quality of the drilling result.

REFERENCES

- [1] Dilley D.N.; Stephenson D.A.; Bayly P.V. and Schaut A.J. (2005). Frequency Shift in Drilling due to Margin Engagement, *Journal of Manufacturing Science and Engineering*, Vol. 127
- [2] Dilley D.N.; Bayly P.V.; Schaut A.J. (2005). Effects of the chisel edge on the chatter frequency in drilling. *Journal of Sound and Vibration* 281, 423–438
- [3] Tooraj A.; Ismail F. (2006). Machining stability in high – speed drilling—Part 1: Modeling vibration stability in bending. *International Journal of Machine Tools & Manufacture*, Volume 46, Issues 12-13, 1563-1572
- [4] Pirtini M.; Lazoglu I. (2005). Forces and hole quality in drilling. *International Journal of Machine Tools & Manufacture*, Volume 45, 1271–128
- [5] Juhchin A Yang et al. (2002). A new dynamic model for drilling and reaming processes. *International Journal of Machine Tools & Manufacture*, 42, 299–311
- [6] Gupta et al. (2003). Modeling and Prediction of Hole Profile in, Drilling, Part 1: Modeling Drill Dynamics in the Presence of Drill Alignment Errors. *Journal of Manufacturing Science and Engineering*, Vol. 125
- [7] Roukema J.C. Altintas Y. (2006). Time domain simulation of torsional–axial vibrations in drilling. *International Journal of Machine Tools & Manufacture*, Volume 46, Issue 15, 2073-2085
- [8] Bais R.S. et al. (2004). Studies in dynamic design of drilling machine using updated finite element models. *Mechanism and Machine Theory*, vol. 39, 1307–1320
- [9] Ewins D.J. (1997). *Modal Testing: Theory and Practice*. John Willey and Sons, New York
- [10] Maia N.M.M.; Silva J.M.M. (1977). *Theoretical and Experimental Modal Analysis*. John Willey and Sons, New York
- [11] Sestieri A. (2000). SDM application to machine tools and engines. *Sadhana (Academy Proceedings in Engineering Sciences)*, vol. 25, 305–317
- [12] Kundra T.K. (2000). Structural dynamic modification via model, *Sadhana (Academy Proceedings in Engineering Sciences)*, vol. 25 261–276
- [13] S.V. Modak (2001). *Studies in Finite Element Model Updating and Application to Dynamic Design*, Ph.D. Thesis, Department of Mechanical Engineering, IIT Delhi (Indian Institute of Technology).
- [14] S.V. Modak, T.K. Kundra. (2002). B.C. Nakra, Dynamic design of machine tool structures using an updated model Pro. *20th International Modal Analysis Conference (IMAC – XX)*, 1489–1494.
- [15] A. Kessentini et al. (2006). Simulation and modelling of the vibration effect on the hole form in drilling. *International journal of simulation and modelling*. 54, 133-144.
- [16] Mustapha Elhachimi, Serge Torbaty, Pierre Joyot. (1999). Mechanical modelling of high speed drilling. 1: predicting torque and thrust. *International Journal of Machine Tools & Manufacture*. 39 553–568.

- [17] K. Hu and N. Vlahopoulos and Z. PZ Mmourelatos. (2002). Formulation for coupling rigid and flexible body dynamics of rotating beam. *Journal of Sound and vibration* 253(3), 603 – 630.
- [18] J.L. Batoz, G. Dhatt (1990). *Modélisation des structures par élément finis, Volume 2: poutres et plaques*. Edition Hermès.