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NUMERICAL STUDY ON DYNAMIC BEHAVIOR OF BORING BAR OF UNEQUAL BENDING STIFFNESS

Abdelghafour BOURDIM[1,*], Abed MEGHDIR[2], Abdelkrim BOUIZEM[3]

[1)] Doctor, EOLE Laboratory, department of mechanical engineering, faculty of technology, University of Tlemcen, Algeria
[2)] Doctor, Institute of Maintenance and industrial security, University of Oran Es-senia, Algeria
[3] Doctor, EOLE Laboratory, department of mechanical engineering, faculty of technology, University of Tlemcen, Algeria

Abstract: *The effects of tool tip orientation with respect to principal axis of non circular boring bar, on the stability of self-excited regenerative chatter vibration are numerically and experimentally investigated. The principal formulation of the problem is based on the finite element method and the resolution of the differential equations governing the vibratory movement of the boring bar use "Theta Wilson" method. The superposition of relative displacements of the cutting tool tip, obtained for various orientations of the tool with respect to their principal axes, in curves of lissajoux characterizes the elliptic trajectory of the cutting tool tip while machining. It is clearly shown that there exist a couple of frequency and orientation angle of the boring bar making it possible to delay considerably the appearance of the phenomenon of chattering, there also existed other couples of angles and dangerous frequencies corresponding to the proper modes of the bar. Results of the numerical simulation of the boring bar motion are successfully confirmed by experimental investigations.*

Key words: *chatter vibratio, regenerative chatter, dynamic stability, boring process, wedging angle.*

1. INTRODUCTION

Chatter in metal cutting process, is in general, the result of both forced and self-excited vibrations [1]. Forced vibration is due to the unbalance of rotating members, such as unbalanced driving system, wear of the kinematic elements (rings of the bearing, teeth of the gears...), wear of machine tools slides, or impacts from a multi-tooth cutter. In practice, the forced vibration sources can be traced by comparing the frequency of chatter with the frequency of the possible force functions. Corresponding measures can then be taken to reduce such vibration sources.

Certain modes of machining are at the origin of appearance of self-excited vibrations known as «chatter"[1]. These type of vibrations consists of two types, namely primary (or non-regenerative type) and regenerative type. The primary or non-regenerative type of self-excited vibration occurs when there is no interaction between the vibratory motion of the system and the undulatory surface produced in the revolution of the workpiece [2,5,6,7, and 9], such as that in threading. Hence it is inherently related to the dynamics of the cutting process. While the regenerative type is due to the interaction of the cutting force and the workpiece surface undulations produced by previous tool passes [1 and 3]. The regenerative type of self-excited vibration is found to be the most detrimental phenomena in most machining process, hence it becomes the focus of this paper.

A considerable amount of work has been done on regenerative chatter in order to explain the origin and to find the remedies [1, 2, 3, 4, 5, ...]. The current theories are not able to predict the conditions of appearance of this phenomenon, because of complexity of the structure of the machine tool, which has a great number of degree of freedom which changes while machining. We will quote in example, the displacement of the machine tool carriage, the wear of the cutting tool, heating of the rotating shafts, etc. Moreover, the structures of the machine tool consists of nonlinearities (discontinuous contact of workpiece/tool, solid frictions , variations of the cutting forces with respect to the amplitudes of displacements, etc.) and instantaneous variations of the dynamic characteristics (damping, tightening, etc.) which escapes calculations.

In order to reduce these dispersions, the analysis of this phenomenon is focused on a boring bar of unsymmetrical bending, to which various rotations around his central axis are made by the variation of the angle of orientation (Figs. 1 and 2).

The design of this boring bar presents a phenomenon of deviated deflection and whose moments of inertia with respect to the fixed axes are given by the following relations [6]:

$$\begin{cases} I_{xx} = \dfrac{I_R + I_T}{2} + \dfrac{I_R - I_T}{2}\cos 2\theta \\ I_{yy} = \dfrac{I_R + I_T}{2} - \dfrac{I_R - I_T}{2}\cos 2\theta \end{cases} . \qquad (1, 2)$$

[*] Corresponding author: Faculty of Technology,B.P.230 13000 – Tlemcen - Algeria,
Tel.:+213(0)43285689;
Fax: +213(0)43285685
E-mail addresses: *bourdim56@yahoo.fr* (A. Bourdim), *Meghdir.abed@yahoo.fr* (A. Meghdir)

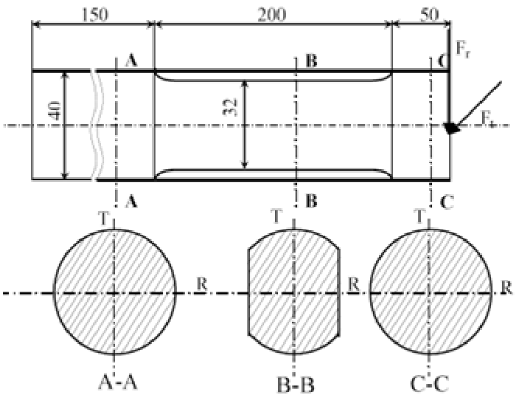

Fig. 1. Bar of unsymmetrical bending.

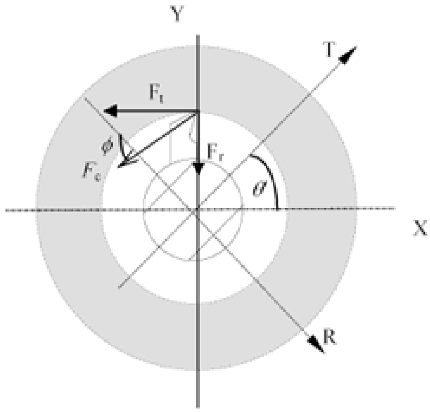

Fig. 2. Cutting forces system.

2. MODELISATION

The bar being fixed with an overhang in the turret of the lathe machine, can be represented perfectly as being modeled as a cantilevered beam in bending, having two identical circular sections, but of different lengths and a noncircular section and is subjected to two lateral deflections due to the action of the efforts F_r and F_t (Figs. 1 and 2).

In order to understand the mechanism of the appearance of the regenerative chatter, it is necessary to analyze the behavior of a structure subjected to the action of an =effort which is itself function of the behavior of the structure, this phenomenon is then analysed by combination of cutting process dynamic and the mechanical receptance of the boring bar in a closed feedback loop that instantaneously controls the tool-workpiece motion [7, 8, and 9] (Fig. 3).

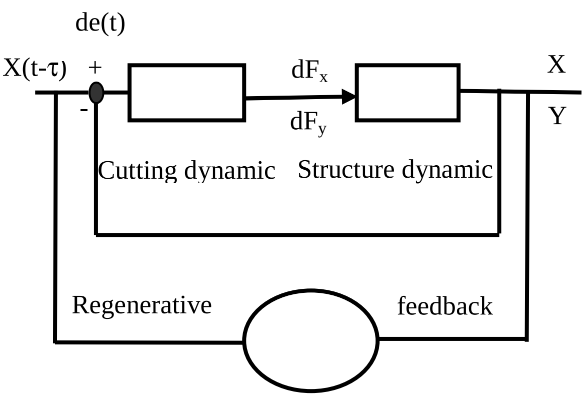

Fig. 3. Block diagram for regenerative chatter.

Therefore if a buckled system is subjected to a variation of the effort d$F(t)$ related to a variation of the displacement of the tool d$e(t)$ by a transfer function of machining T_U (Cutting dynamic) such as:

$$\mathrm{d}F(t) = T_U\, \mathrm{d}e(t). \tag{3}$$

And the amplitude of current vibration $X(t)$ is caused by the transfer function of the machine T_M (structure dynamic) such as:

$$X(t)=T_M \mathrm{d}F(t). \tag{4}$$

Which combined with that of the precedent cycle $X(t-\tau)$, regenerate the displacement d$e(t)$.

According to the theory of control systems we will have:

$$\frac{\mathrm{d}F(t)}{X(t-\tau)} = \frac{T_M}{1-T_U T_M}. \tag{5}$$

The complete analysis of the dynamic behavior of the system, is comparable with that controlled linear system, composed of an elastic system characterized by its dynamic flexibility in interaction with the cutting process [2], defined by the sensitivity of the tangential component variation dF_t(t) with respect to the chip thickness variation de(t) such as:

$$\mathrm{d}F_t(t)=K_1 \mathrm{d}e(t). \tag{6}$$

The vibratory movement $x(t)$ relating to the position of the cutting tool tip with respect to the workpiece, affects the chip thickness by a value $-\,x(t)$. We introduces a overlapping factor " μ " of the cutting marks, and τ the lag time to represent the preceding cutting surface "regenerated ", where the instantaneous chip thickness variation is given by:

$$de(t) = -\,x(t) + \mu\, x(t-\tau). \tag{7}$$

According to equations (6) and (7), we will have:

$$\mathrm{d}F_t(t) = K_e b\,[-x(t) + \mu x(t-\tau)]. \tag{8}$$

Figures 4.a and 4.b represent the effects of a primary and regenerative chatter [3] and [4].

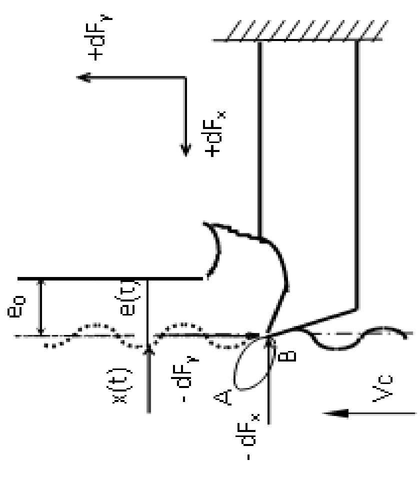

Fig. 4.a. Effect of primary chatter (μ = 0).

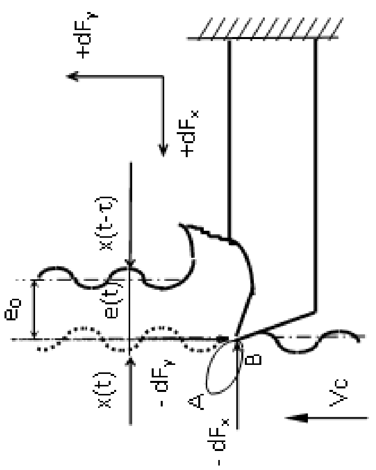

Fig .4.b. Effect of regenerative chatter (μ≠0).

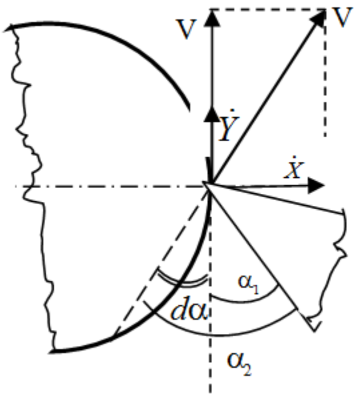

Fig. 5. Clearance angle variation dα(t).

And the dependence of the radial component variation d$F_r(t)$ with respect to the clearance angle variation dα(t) [5] (Fig. 5) is given by:

$$dF_r\,(t) = K_2 d\alpha(t). \tag{9}$$

The proportionality factors K_1 and K_2 of respectively tangential and radial cutting rigidity are evaluated experimentally [6,7].

The variation of the clearance angle dα(t) is given by:

$$\tan d\alpha(t) = \frac{\dot{X}(t)}{Vc},$$

$$\tan d\alpha(t) = d\alpha(t) \quad \text{for small variations}$$

$$dF_r(t) = \frac{K_2}{R\Omega}\dot{X}(t). \tag{10}$$

The action of the cutting process defined by the relations (8) and (10) on the elastic system appears by the deformation of the boring bar, thus causing relative displacements X and Y of the tool tip with respect to the machined surface. Since the deflections of the bar are inversely proportional to the moments of inertia Ixx and Iyy, displacements of the tool tip X and Y vary with respect to the orientation of the bar, i.e. the wedging angle (θ) between the principal axes of the bar (R, T) and fixed axes referring to the tool tip.

We must thus solve a system of differential equation [11, 12] of the vibratory movement which is given by:

$$[M]\{\ddot{x}\}+[C_x]\{\dot{x}\}+[K_x]\{x\}=\{F_r(t)\}, \tag{11}$$

$$[M]\{\ddot{y}\}+[C_x]\{\dot{y}\}+[K_x]\{y\}=\{F_t(t)\}. \tag{12}$$

To solve this type of equation, we developed a computation software based on the finite element method. The boring bar consists of a long slender beam carrying the tool at its end, and having by its nature a comparatively low stiffness, is discretized in three elements and having two degrees of freedom per node.

The software code enables us to calculate the elementary matrices of masse and rigidity and link them in order to find the global matrices of masse and rigidity of the bar.

The structural damping has been neglected.

For the resolution of our homogeneous system of equations, we used the method of integration known as Thêta Wilson. A solution of the form $x(t) = x_0 \cos \omega t$ has been considered, we could approximate the expressions of the cutting forces given by equations (8) and (10) as follows:

$$F_r\,(t) = F_0 \cos(\omega t + \phi), \tag{13}$$

$$F_t\,(t) = F_0' \sin \omega t. \tag{14}$$

In order to be able to reach the permanent mode, the step of time Δt is found to be equal to 10^{-4}sec.

Substituting these two expressions of the cutting forces in our differential equations (11) and (12) and after introduction of the boundary conditions, the results of simulation have enabled us to determine couples of frequency and wedging angle of the bar corresponding to the beginning of the resonance and the proper modes of the bar (Fig 6a).

In order to locate the peaks of resonance, and to determine the couples " θ and F ", the graph of Fig. 6a is projected on plan (θ,F). These peaks are represented on the graph of Fig. 6b by rhombuses allowing us to determine the corresponding angle and frequency.

This graph delimits the interval of the dangerous frequencies for various wedging angles of the bar. So it is easy to be able to avoid resonance by suitable choice of both frequency and wedging angle.

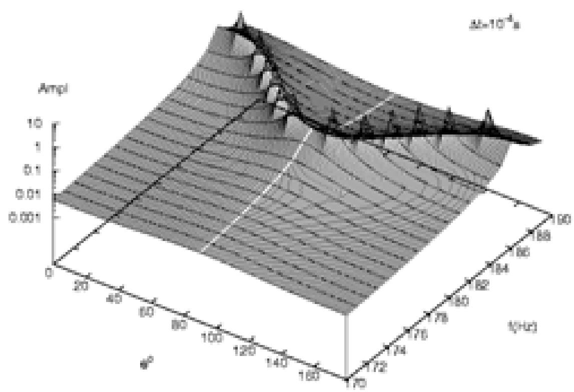

Fig. 6a. Variation of X with respect to θ and F.

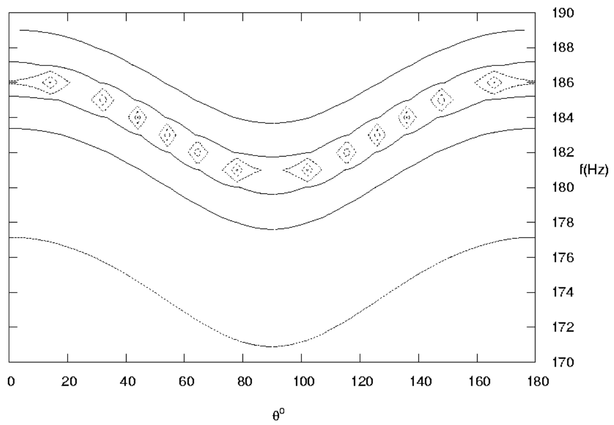

Fig. 6b. Iso-Amplitudes diagram.

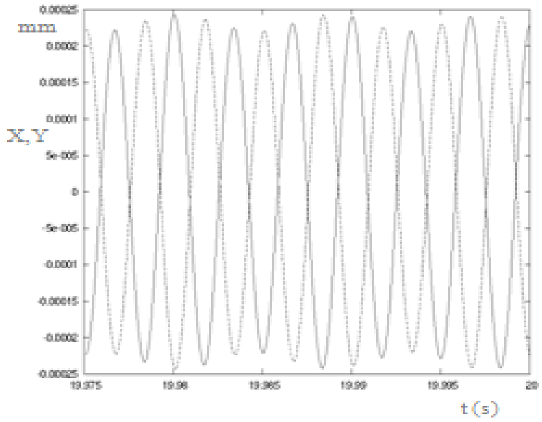

Fig. 8b. X,Y = *f* (*t*) for θ = 30° and *F* = 300 Hz.

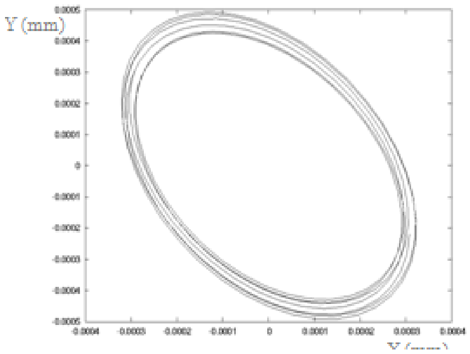

Fig .7a. Trajectory of the tool tip
(θ = 48° and *F* = 262.5 Hz)

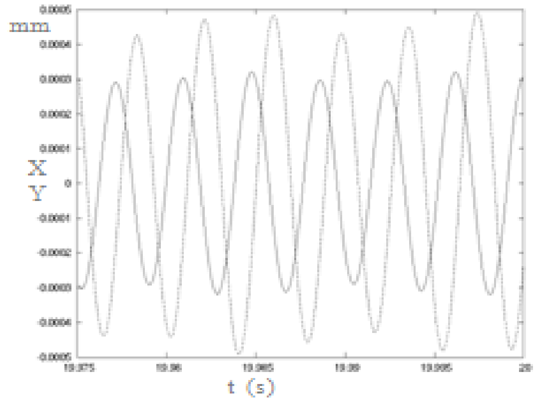

Fig. 8a. *X*,*Y* = *f* (*t*) for θ = 48° and *F* = 262.5 Hz.

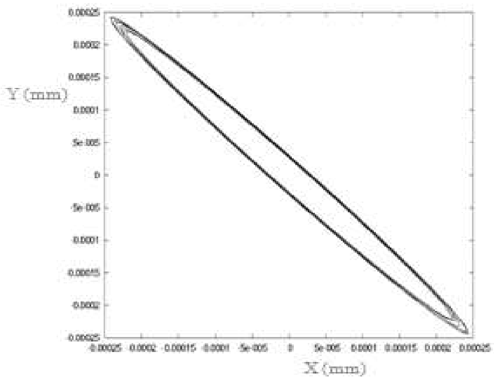

Fig .7b. Trajectory of the tool tip
(θ = 30° and *F* = 300 Hz)

The results of simulation also enable us to find a couple of frequency and angle of orientation allowing to reduce these vibrations (Fig 7a and b)

It is clearly seen the flatness of the curves in lissajoux for a couple of frequency and angle of orientation of the tool tip (Figs. 6a and b), which means that the stability of the machining operation depends not only on the cutting conditions, but also of the position of the tool tip with respect to the principal axes of the bar. The slope of these curves depends on lag angle between displacements *X* and *Y* (Figs. 8a and b).

3. EXPERIMENTAL RESULTS [7]

3.1. Experimental apparatus

The experimental apparatus consisted of a Harding Cazeneuve lathe type HB 725 of 16 kW, displacement transducers with their associated electronics and a digital spectrum analyzer, Hewlett Packard 3566A. All experiments involved only right-handed orthogonal cutting. Positive rake tool inserts, Sandvik type Tmax PCNMG 120412, were employed supported by a cylindrical bar of 40 mm in diameter, with flat part thickness 0.8 D tool holders. The rake and clearance angles were respectively 10° and 5°. Cylindrical work pieces of XC 38 steel as shown in figure 9 were machined under the following cutting conditions:

n = 125 rpm; f_n = 0.1 mm/rev; a_p = 1 mm.

The orientation angle is varied from 0° to 114° in step of 6°

All the operations were carried out without lubrication with new tools to avoid the influence of wear on the phenomenon.

In order to ensure the appearance of chatter vibration, the boring bar is voluntarily fixed with an overhang (Q_0 = 250 mm). Horizontal and vertical displacements of the tip of the tool were recorded using an incremental position sensor of type 591524 and D 412025 Bently Nevada for each orientation.

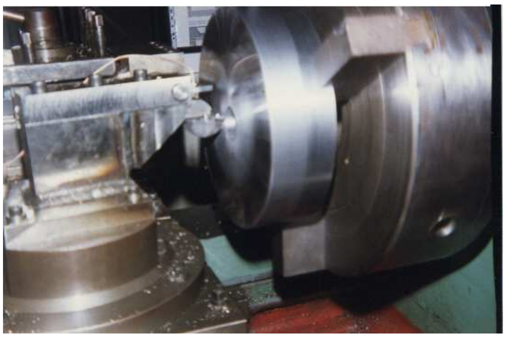

Fig. 9. The experimental setup.

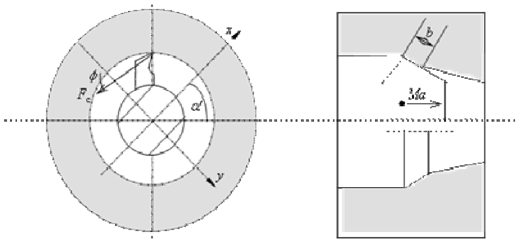

Fig. 10. Experimental conditions.

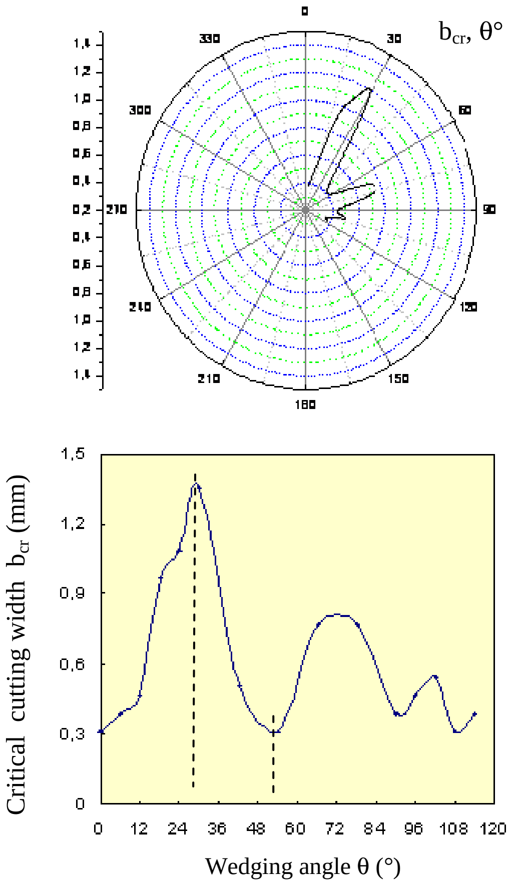

Fig. 11. Variation of cutting width.

When the wedging angle θ varies, the limit of stability characterized by the cutting width limits b_{cr} varies as indicated by the curves represented by Fig. 11.

According to this results, when θ = 30°, b_{cr} is maximum, and for θ = 0°, 54° and 108° it is minimal. The stability of cut is thus better for only one orientation (θ = 30°),

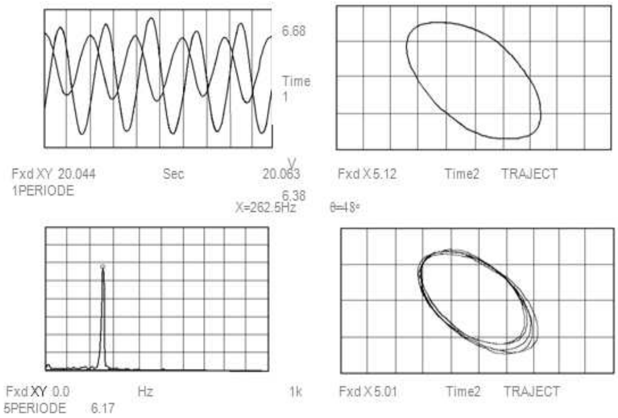

Fig. 12. Trajectory for θ = 48° and F = 262.5 Hz and displacement $Y = f\,(X)$.

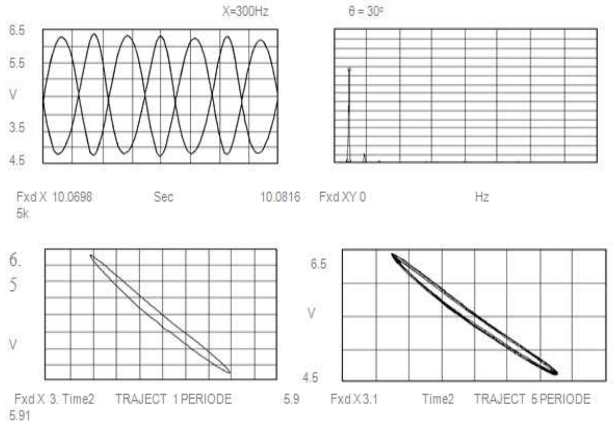

Fig. 13. Trajectory for θ = 30° and F = 300 Hz and displacement $Y = f\,(X)$.

The superposition of these displacements in curves of lissajoux, and their Fourier transform are represented by Figs. 12 and 13.

It should however be noticed that in mode of light chattering represented by Fig. 12, the ellipse characterizing the dynamic trajectory of the point of the tool is flattened than in mode of intense chattering (Fig. 11). It is noticed that in the case of a light chattering, the two movements are roughly in opposition of phase (φ~π).

4. CONCLUSIONS

The physical interpretation of the observed phenomenon in this work is basically linked to the instantaneous position of the cutting tool edge with respect to the direction of the minimum stiffness of the boring bar. When the critical position is reached, the workpiece tends to approach the cutting tool, which increases the thickness of the chip which itself allows the cutting force to increase and hence the cutting process becomes unstable. The other positions correspond to a situation where the workpiece tends to go further from the cutting tool edge and the opposite situation takes place. The thickness of the cut diminishes and the system becomes stable.

The used model allowed us to simulate the trajectories of the tool tip and to show that the threshold of chattering can be delayed by choosing a particular position of the principal axes of the bar tool holder with respect to the tool tip.

Displacements of the tool edge are polarized elliptically, and the curves which result from the superposition of these displacements, show that they are not of the same magnitude and are in phase lag to each others. The phase lag is thus one of the principal causes of the appearance of the phenomenon of chattering.

It has been well established that the computed results are in very good agreement with the experimental ones.

In the particular case of $k_u = k_v$ and $k_t = k_r$ the dynamic stability depends only on cutting conditions and dynamic characteristics of the boring bar, which is in this case of circular section.

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