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Cutting forces measurements during discontinuous machining process

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Abstract: He presents the various cases where the measurement of cutting forces in fast transitory mode is necessary. He exposes also a method of compensation of inertial effects which denatures the signals provided by the most used piezoelectric dynamometers. The possibilities of this method, based on the accelerations measurements of the upper part of the dynamometer, are exposed for the case of a discontinuous turning process. It is shown that the undesirable effects related to the resonance frequencies of the dynamometer and to its base movements are mainly eliminated.

Keywords: compensation of inertial forces; cutting forces measurements machining; discontinuous turning process.

Biographical notes: B. Benmohammed is working as an Associate Professor in the Department of Mechanical Engineering, Faculty of Technology at University of Batna, Algeria. He completed his graduation in Mechanical Engineering from University of Batna and his Post-graduation in Modelling the Dynamic Behaviour of Machining Processes from Ecole Nationale Supérieure d'Arts et Métiers (ENSAM), Paris, France and he received his PhD in 1996. He has also received his HDR in 2006 in Mechanical Engineering from University of Batna, Algeria. He has contributed a number of research papers at the international level in the area of modelling the dynamic cutting processes and especially for the turning case. He is actually developing new research methods at Product Research Laboratory (PRL) at the Mechanical Engineering Department, University of Batna, Algeria.

1 Introduction

Measuring cutting forces generated during cutting or abrasion processes is one of the best ways to characterise these operations. The analysis of cutting forces is necessary to understand the cutting phenomenon, to control static and dynamic behaviour of a whole piece-tool machine system and is a natural way of monitoring and diagnosis for cutting operations.

Many authors such as Chung and Spiewak (1994) have studied the possible measurement inaccuracies, when using piezoelectric dynamometers in the machining processes.

The main measurement inaccuracies result from the fact that dynamometers behave approximately as a weakly damped second order system. It presents a resonance frequency depending on the sensors type and their layout.

Resonance is all the more perceptible as damping is weak (about 500 N.s.m^{-1}). During cutting operations, the fluctuating frequencies of cutting forces approach or exceed the resonance frequency, making a difference between the dynamometer signals and the exerted forces.

Three cases of dynamometer disturbances could be described:

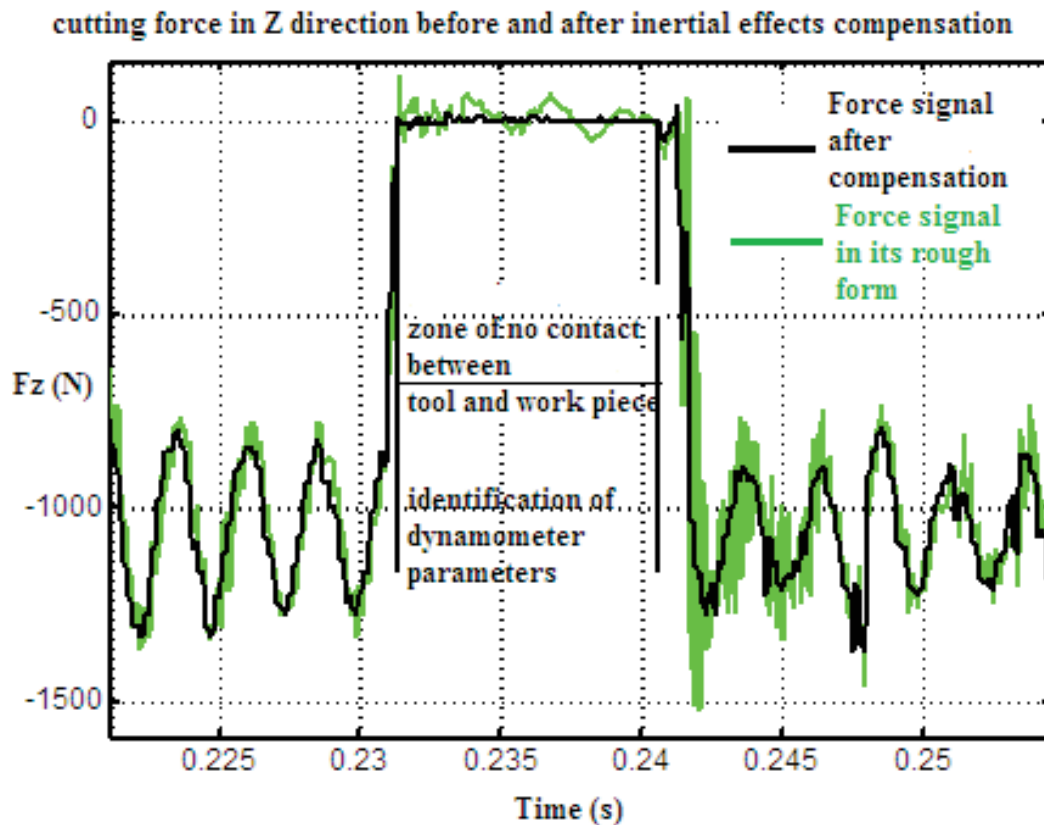
- During discontinuous cutting operations; even if the frequency of force signals repetition is weak. The presence of transitions with abrupt variations produces a wide range of frequencies and could generate unacceptable distortions. This is the case for milling, discontinuous turning and broaching operations. The following example (Figure 1) shows a typical response signal in the case of discontinuous turning process. It seems obvious that when the rotating speed increase, the repetition frequency increases and could reach the dynamometer resonance frequency.
- During the appearance of a vibration phenomenon such as chatter, the frequency could range from some hundred Hertz to some kilo Hertz. Nevertheless, this analysis is very important because it emphasises a damping phenomenon, which has to be taken into account for a good simulation of vibration phenomena. We could meet such conditions in the case of an external vibration source such as in ultrasonic abrasive machining.
- During the superposition of a random force to the continuous force. Generally, we meet this case during the study of influence of the local material micro hardness variations on the cutting force evolution. It is very important to take into account the dynamometer distortions. Distortions attenuation could be considered in several ways. The first one consists in making use (or creating) a dynamometer having a satisfying bandwidth (Gautschi, 1971). We could then set up preliminary calibrating operations (Kegg, 1965; Komanduri, 1971). These operations are difficult and took a long time to be implemented. They are only valid for a given measuring system configuration. Most efficient methods are divided into two classes:
 - 1 Those of the first class aim at attenuating inertial errors and damping forces. For this, it is necessary to carry out simultaneous measurements of different accelerations. They allow firstly the evaluation of inertial effects and secondly the evaluation of speed and consequently damping forces. This inertial effects compensation has been particularly studied for one dimensional system by

Tanaka and Ikawa (1970) and by Tlustý et al. (1987). The general case was envisaged by Gyssels (1980).

- 2 The second class of distortions attenuation methods is based on a special adaptive filters use and it was done by Spiewak (1992) and Chung and Spiewak (1994).

Possibilities of the first class method used in laboratory for the fine analysis of cutting forces are presented in this research work.

Figure 1 Discontinuous turning process (see online version for colours)



2 Modelling of dynamometer signals

The dynamometer employed in our investigation is 9257B ‘Kistler’ model (Figure 2, Benmohammed, 1996). It is made up from a system of four ‘three-directional measure elements’, which are inserted between the dynamometer base and superior plate ‘platform’. On the dynamometer higher plate is fixed the work piece or the cutting tool. The dynamometer base and superior plates are considered as rigid elements, and every measure element is assimilated in each of the three directions to a spring associated to a viscous damper.

The platform (the dynamometer superior plate) is considered as a rigid body, which has six degrees of freedom, three translations and three rotations. Considered as a dynamic system, the platform exhibits six resonance frequencies associated with its vibration modes. The considered dynamometer installed on a machine tool is a part of a larger multi degrees of freedom system, as explained in (Figure 3).

Figure 2 Mechanical design of the considered dynamometer: a ‘9257B model’

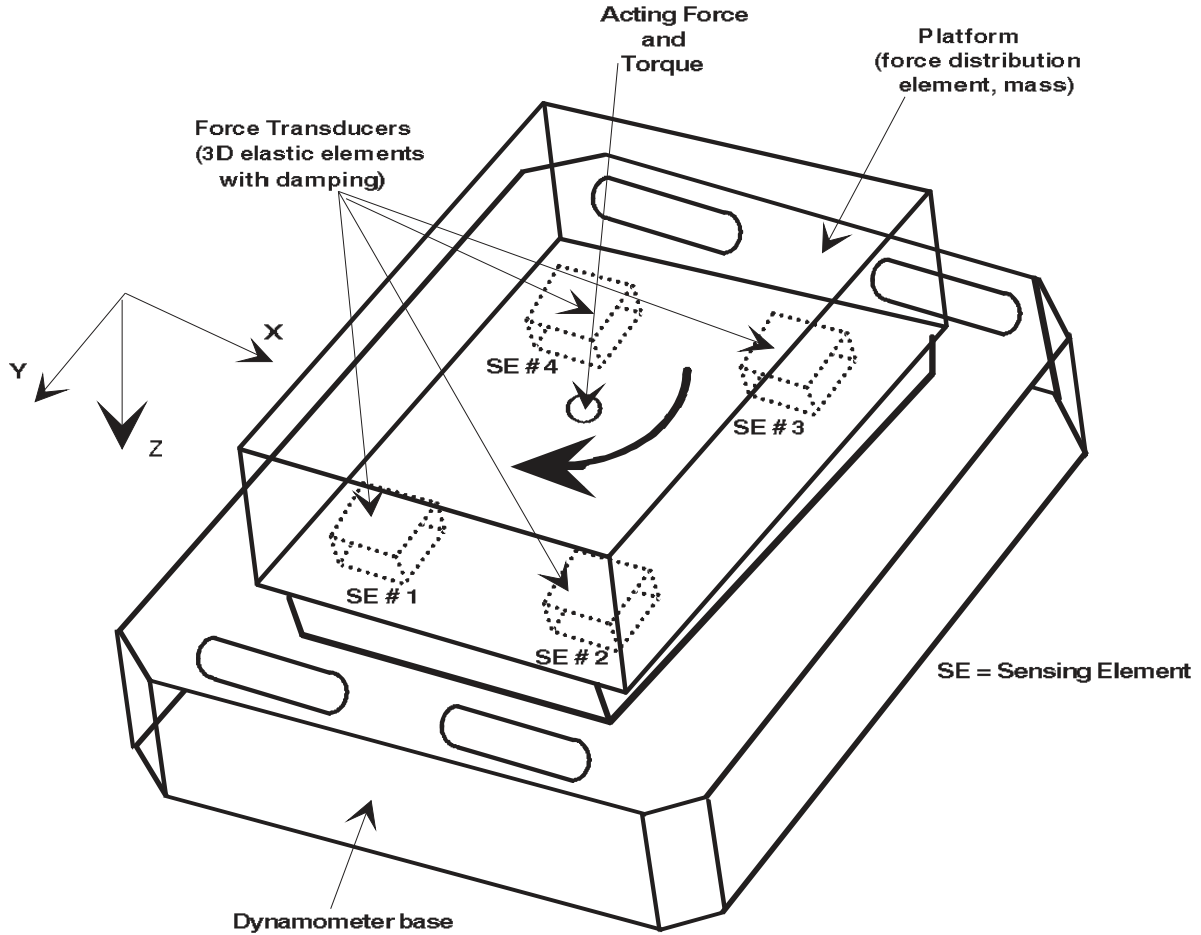
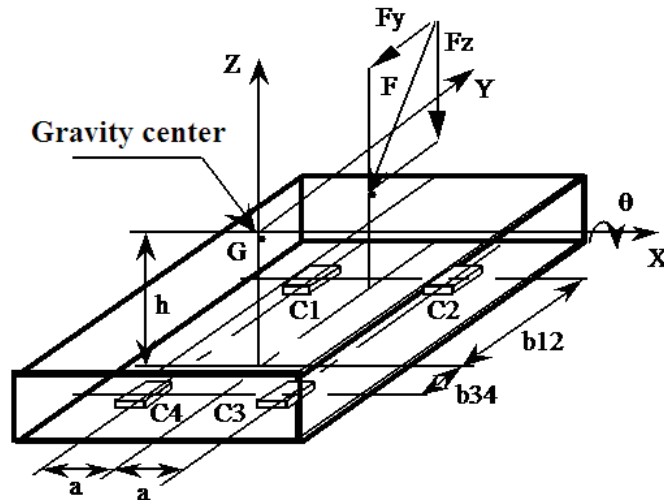


Figure 3 Geometrical dynamometer layout



In this present paper, a complete dynamic model encompassing all the six signals from dynamometer has been established and limited to the simplified case only. The developed model is characterised by the following hypothesis:

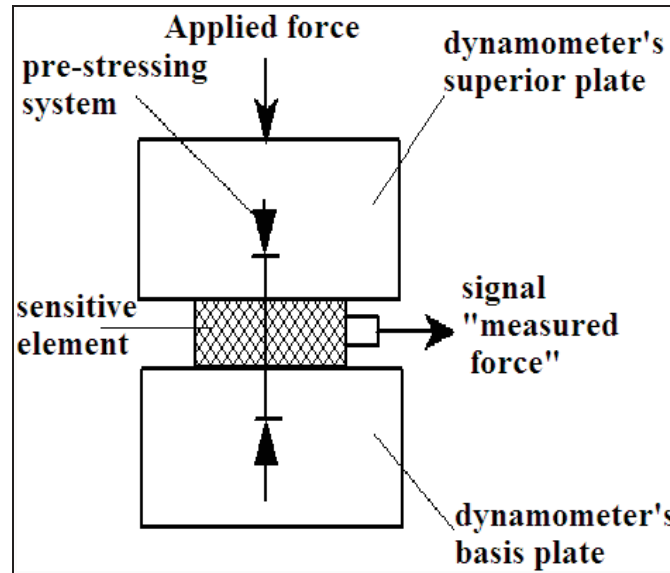
- the masses distribution of the cutting tool and the dynamometer’s higher plate are symmetrical to the dynamometer YZ plane of symmetry (Figure 3)
- all the dynamometer characteristics (the sensor’s: stiffness, damping coefficients and sensitivities) are symmetrical to the dynamometer YZ plane of symmetry
- external force is applied in the contact point of the cutting tool on the work piece

- it has been assumed that the dynamometer base plate is immobile (it is supposed as a rigid body).

He presents, in the beginning of his research work, the acceleration compensation principal.

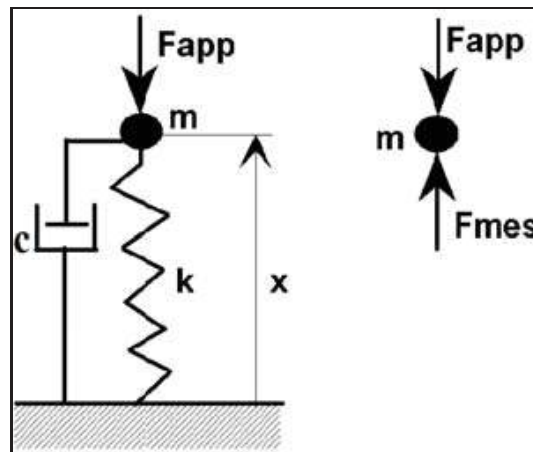
The principle of the acceleration compensation is exposed for the case of one dimensional configuration. The sensor is composed of a sensitive element pre-stressed between the base and the higher plates (Figure 4).

Figure 4 The piezoelectric sensor as a pre-stressing system



He could assimilate the sensitive element to a very simple model: mass-spring-damper (Figure 5).

Figure 5 Mathematical modelling of the sensitive element



Under these conditions, he has (Benmohammed, 1996) obtained a plane movement characterised by the two translations (y, z), of the gravity centre of the dynamometer platform; and also its rotation (θ) around the X axis.

The dynamic equation giving the dynamometer superior (or higher) plate movement is:

$$M.\ddot{q} = F_e(t) + \sum F_{ci}(t) \quad (1)$$

$$\mathbf{q} = [y \quad z \quad \theta]^t, \mathbf{F}_e = \begin{bmatrix} F_y & F_z & M_x \end{bmatrix}^t, \mathbf{F}_{ci} = \begin{bmatrix} F_{cyi} & F_{czi} & M_{cxi} \end{bmatrix}^t \text{ and } \mathbf{M} = \begin{bmatrix} m & 0 & 0 \\ 0 & m & 0 \\ 0 & 0 & I_{xx} \end{bmatrix}.$$

Usually, each of the F_{cyi} and F_{czi} force components is decomposed into two values: one corresponding to the elastic deformation of sensing elements and the other is due to their damping characteristics.

He has used the following relationship: $F_{cyi} = F_{kyi} + F_{ayi}$ and $F_{czi} = F_{kzi} + F_{azi}$.

By calling z_i and z_{bi} the sensitive (i) displacements in the Z direction, which are respectively linked to the dynamometer superior and base plates, he has found the following relationship:

$$F_{czi} = k_{zi} \cdot (z_i - z_{bi}) + c_{zi} \cdot (\dot{z}_i - \dot{z}_{bi}) \quad (2)$$

The same relationship is valid for the Y direction:

$$F_{cyi} = k_{yi} \cdot (y_i - y_{bi}) + c_{yi} \cdot (\dot{y}_i - \dot{y}_{bi}) \quad (3)$$

where

$(z_i - z_{bi})$ and $(y_i - y_{bi})$ are the displacements of sensor (i) in Z and Y directions.

$(\dot{z}_i - \dot{z}_{bi})$ and $(\dot{y}_i - \dot{y}_{bi})$ are the speeds of the sensor (i) displacements in Z and Y directions.

He supposes that the emitted signal from sensor (i) in Z direction is proportional to the term $k_{zi} \cdot (z_i - z_{bi})$. Under these conditions, the dynamometer base movement is taken into consideration and he could then write the equation (1) as follows:

$$\mathbf{M} \cdot \ddot{\mathbf{q}} = \mathbf{F}_e(t) + \sum \mathbf{F}_{mesi}(t) + \sum \mathbf{F}_{ai}(t) \quad (4)$$

Equation (4) makes possible the calculation of the applied force $F_e(t)$ component from the three other terms. The term $\sum \mathbf{F}_{mesi}(t)$ is given by the dynamometer, providing that he separately process the signals emitted by the different sensors. The term $\mathbf{M} \cdot \ddot{\mathbf{q}}$ is obtained from accelerations measurements. The valuation of the damping term $\sum \mathbf{F}_{ai}(t)$ is more difficult.

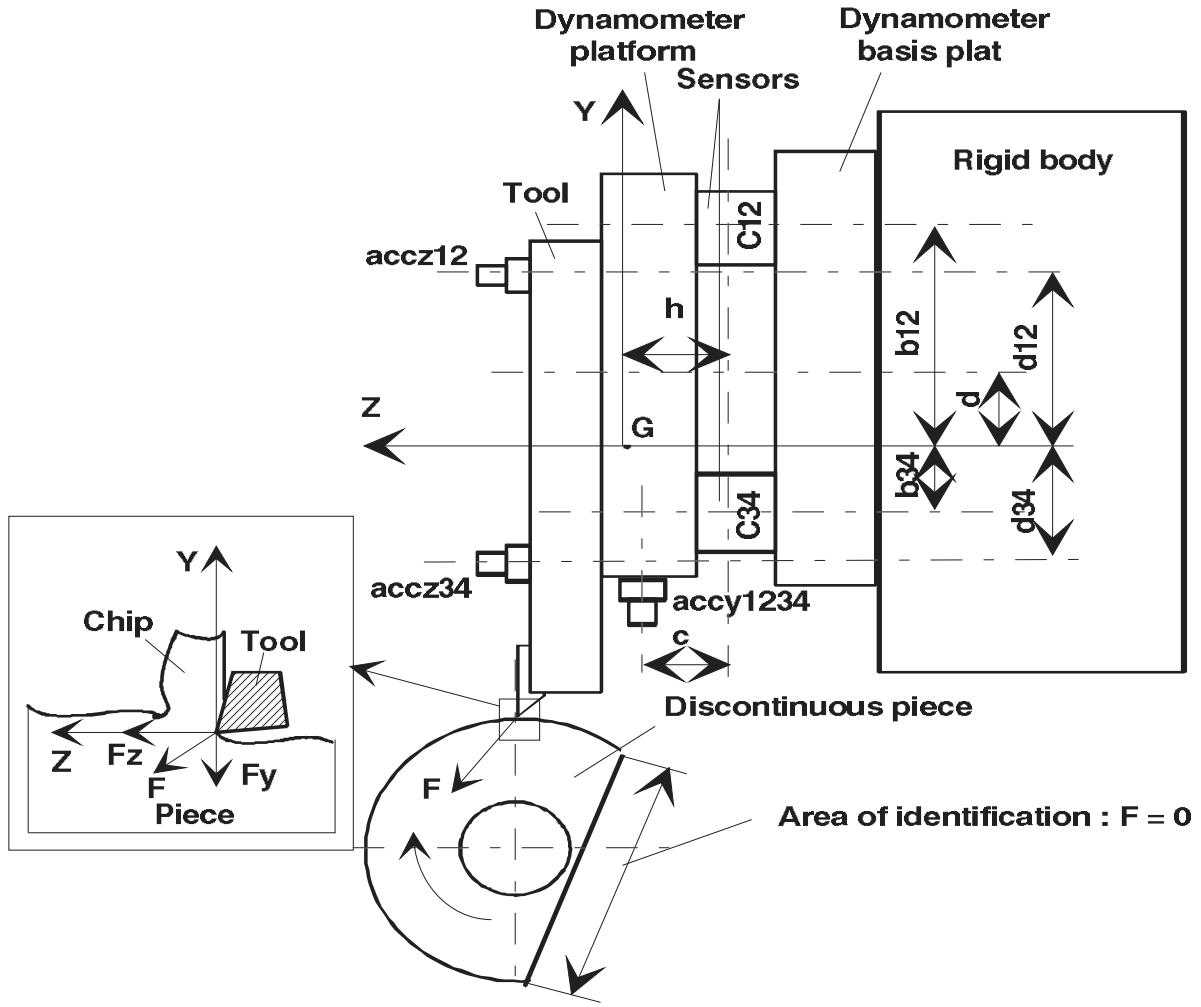
He could then evaluate the dynamometer platform speeds from the accelerations measurements, but it would be necessary to know the dynamometer base plate movements.

This modelling is not valid unless the dynamometer platform has a rigid body and also a special attention to the cutting tool fixation, on the dynamometer platform, should be paid.

3 Applicability of the developed model to turning process

The layout of the cutting tool on the dynamometer higher-plate is described in Figure 6 (Benmohammed and Lapujoulade, 1996a, 1996b).

Figure 6 Discontinuous turning process



Note: Layout of the sensitive elements (C_{12} , C_{34}) and of the accelerometers ($accz_{12}$, $accz_{34}$, $accy_{1234}$)

In his current study, only Z component of cutting force has been evaluated. Otherwise it would have been necessary to add an accelerometer in the Y direction. He supposes that symmetry hypothesis around the dynamometer YZ plane are satisfied. Especially, the turning cutting tool does not generate a force in the X direction. Under these conditions, the emitted signals by C_1 and C_2 sensors are summoned as they are emitted by a unique sensor C_{12} , and similarly for the sensors C_3 and C_4 ; they are summoned to a unique sensor C_{34} .

By taking into consideration only the Z direction force component, equation (4) could be rewritten as follows:

$$F_z = m \cdot \ddot{z} - F_{z12} - F_{z34} - F_{a12} - F_{a34} \quad (5)$$

With the sign conventions linked to the dynamometer, he could express the F_{z12} and F_{z34} components as a function of the measured values F_{mz12} and F_{mz34} delivered by the sensors:

$$F_{z12} = -F_{mz12} \text{ and } F_{z34} = -F_{mz34}.$$

The acceleration of the gravity centre of the dynamometer platform in Z direction is calculated from the measured accelerations acc_{12} and acc_{34} , as follows:

$$\ddot{z} = \frac{\text{acc}_{12} \cdot b_{34} + \text{acc}_{34} \cdot b_{12}}{b_{12} + b_{34}}.$$

He could also write:

$$\ddot{z} = B_{12} \cdot \text{acc}_{12} + B_{34} \cdot \text{acc}_{34},$$

with

$$B_{12} = \frac{b_{34}}{b_{12} + b_{34}} \text{ and } B_{34} = \frac{b_{12}}{b_{12} + b_{34}}.$$

He could then evaluate the sensors speeds after the accelerations integration. The damping coefficient (C) is determined by observing the signal oscillations decrease after a shock, and the damping forces corresponding to the sensors C_{12} and C_{34} are evaluated from these data.

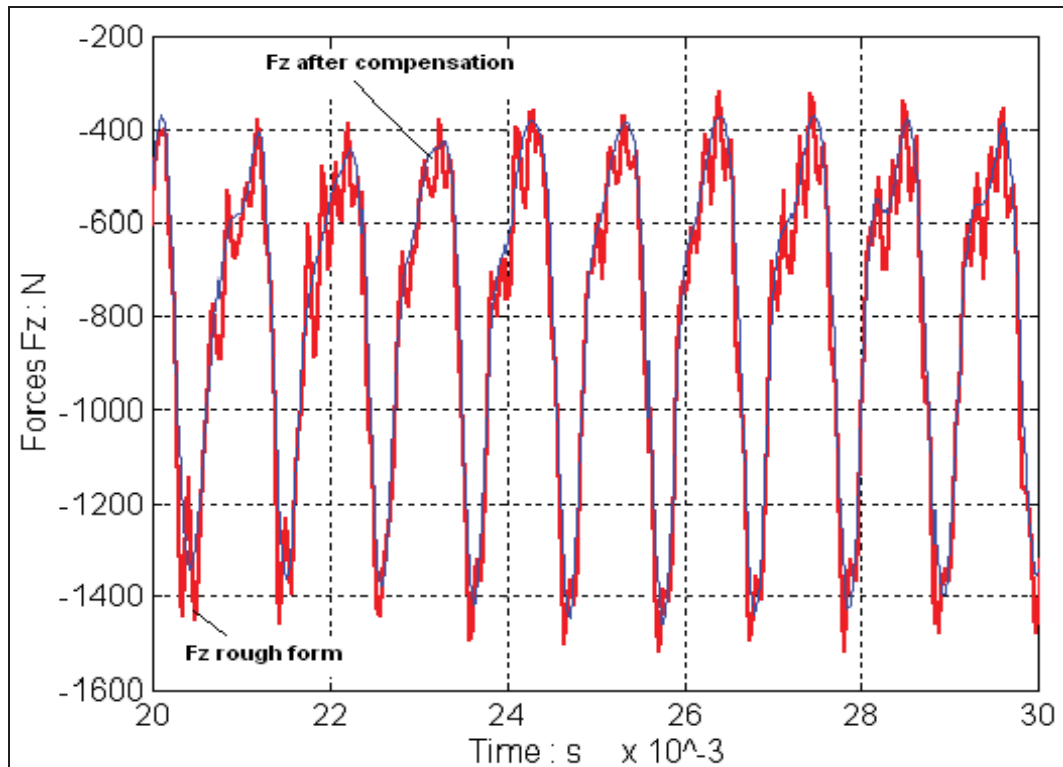
The mass of the dynamometer platform and the cutting tool, and the position of their gravity centre (G) are difficult to evaluate. It is necessary to operate by identification in the area where there is no machining, otherwise where the external force is null, (Figure 6).

Then he obtained the following relationship:

$$m \cdot B_{12} \cdot \text{acc}_{12} + m \cdot B_{34} \cdot \text{acc}_{34} + F_{mz_{12}} + F_{mz_{34}} - F_{a_{12}} - F_{a_{34}} = 0 \quad (6)$$

The objective is to ascertain values of coefficients (m, B_{12} and B_{34}) so that the relationship (6) is the best possible. To resolve this problem he has used the optimisation technique and he has chosen the least square method.

Figure 7 Discontinuous turning process (see online version for colours)



Note: The F_z component of a cutting force before and after compensation of the inertial effects.

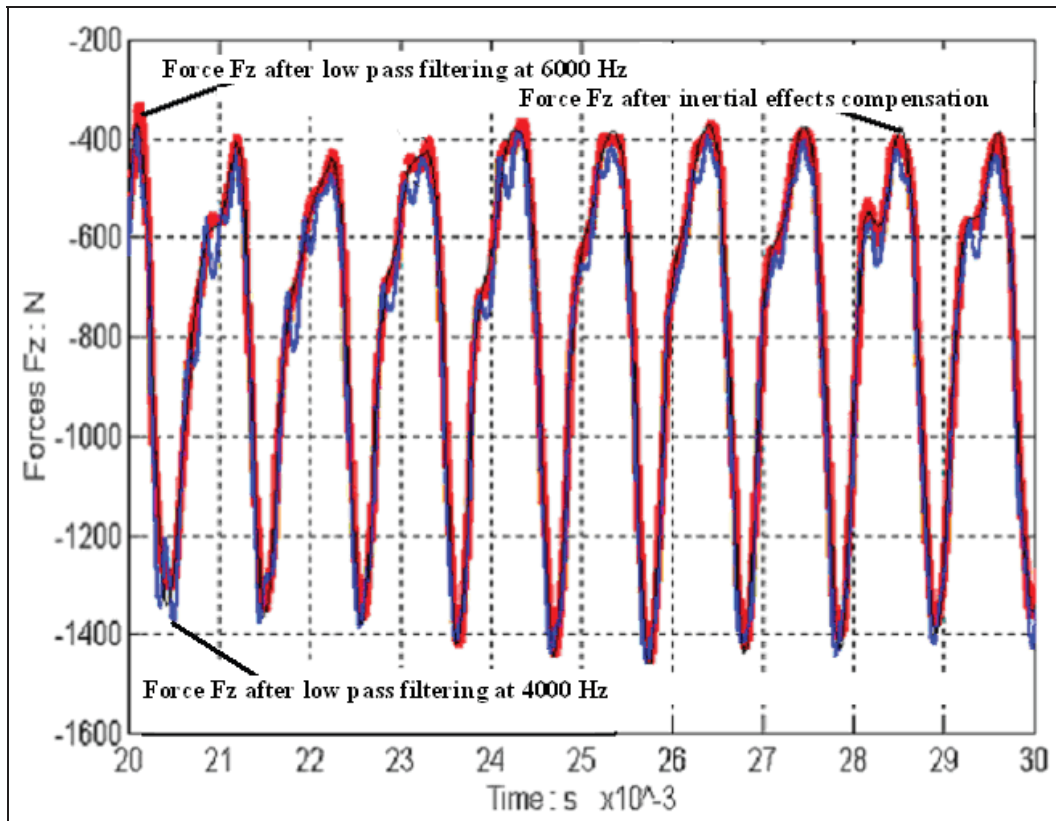
Then these coefficients are used to evaluate the external cutting force. Figure 7 shows the F_z cutting force component under its rough form given by the dynamometer and also after the inertial effects compensation.

The profile of the observed force signal is very close to the correct supposed form defined by calculation. He compares two signal forces, the first given directly by the dynamometer and the second obtained by calculation from relation (4):

$$F_e(t) = M.\ddot{q} - \sum F_{mesi}(t) - \sum F_{ai}(t) \quad (7)$$

The rising front in the cutting force signal is respected. It remains oscillations of weak amplitude and of a frequency superior to that of previous perturbations. Their source is due to the cutting process or to the dynamometer functioning, which is more complex in reality than the developed model. Especially, there are flexion vibrations of the dynamometer platform. On the other hand, the oscillations of lower frequency occur and they are corresponding to the cutting tool vibrations. The quality of the cutting force signal after compensation of inertial effects is to be compared to that given by a simple low-passes filtering of the initial cutting force signal (Figure 8).

Figure 8 Cutting force signals obtained by simple low-passes filtering (4,000 Hz and 6,000 Hz) and the force signal after inertial effects compensation (see online version for colours)



Besides the lag on the force signal, which could be eliminated by appropriate techniques, the filtered signal presents perturbations that are not a residue of the eliminating frequency, but the result of the stiff front filtering.

4 Conclusion

The compensation of inertial forces (Benmohammed, 1996) is vital for the study of cutting forces in discontinuous machining and particularly in the case of a discontinuous turning process. It is also a necessity for the study of cutting process in vibrations presence.

Within the framework of laboratory instrumentation, the compensation of inertial forces by acceleration measurements is possible. It is relatively simple to implement, each time that we could simplify the dynamometer modelling by creating a symmetry distribution of mass and force, as compared to the dynamometer median planes XZ or YZ.

Force signals obtained in discontinuous turning process allow a very fine analysis of the cutting force in the presence of abrupt variations.

This modelling is not valid unless the dynamometer platform has a rigid body. Also a special attention to the cutting tool fixation on the dynamometer platform should be paid.

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Nomenclature

a	Distance of sensing elements from G (gravity centre) in X direction
b_{12}	Distance of sensing elements (C_1, C_2) from G in Y direction
b_{34}	Distance of sensing elements (C_3, C_4) from G in Y direction
d	Distance of the dynamometer axis centre from G in Y direction
d_{12}	Distance of accelerometer (acc_{12}) from G in Y direction
d_{34}	Distance of accelerometer (acc_{34}) from CG in Y direction
G	The gravity centre of the cutting tool - dynamometer higher plate
C_1, C_2, C_3, C_4	Piezoelectric sensors
h	Distance of sensing elements form G in Z direction
x	Displacement of the gravity centre in X direction
y	Displacement of the gravity centre in Y direction
z	Displacement of the gravity centre in Z direction
θ	Cutting tool gravity centre rotation around X axis
F_e	Vector of external forces and moment applied to the dynamometer
F_{ci}	Vector of forces acting on the dynamometer platform
F_{ki}	Vector of elastic force components
F_{ai}	Vector of damping force components
K_{yi}, K_{zi}	The sensor (i) stiffness respectively in Y and Z directions
C_{yi}, C_{zi}	The sensor (i) damping coefficients respectively in Y and Z directions
I	Index denoting the location of piezoelectric sensors (C_1, C_2, C_3, C_4) inside dynamometer
M	Mass of the cutting tool – dynamometer higher plate
M	Elements of the matrix M of mass
q	Generalised coordinates vector.
I_{xx}	Moment of inertia of the cutting tool on the dynamometer platform in X direction
