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Passive vibration absorber effect on the machining surface quality of a flexible workpiece

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A B S T R A C T

This paper presents a study of the vibratory behaviour of a flexible workpiece subject to a milling end operation. Indeed, this vibratory behaviour is critical, especially when the excitation frequency is near to the resonance. For this reason, passive vibration suppression is considered in order to attenuate the dynamic response of the milled workpiece and decrease the dynamic effect on the resulting machined surface roughness and flatness. In order to confirm the efficiency of the passive vibration suppression, the vibratory behaviour and the quality (roughness and flatness) of a machined surface are studied without and with passive absorber (TMD) using a finite-element model.

1. Introduction

The presence of regenerative chatter in machining process causes poor surface quality and inaccuracy. Therefore, many numerical and experimental investigations have been conducted to attenuate vibrations in machining in order to improve surface quality. For this reason, many researchers have studied the impact of the cutting parameters on the surface quality. For example, Sutherland and DeVor [1] developed a model for the prediction of the cutting force system and surface error in end milling. He took into account the effect of the cutter deflections on chip load, of the cutting forces, and consequently of the surface error. Montgomery and Altintaş [2] presented a model of the milling process. The kinematics of the cutter and workpiece vibrations are modelled, which identifies the orientation and velocity direction of the cutting edge during dynamic cutting. This model consists in identifying the forces and surface finish under rigid or dynamic cutting conditions. Based on Nyquist's stability criterion, Altintaş and Budak [3] proposed an analytical method for the stability prediction of the regenerative chatter in milling process using a two-dimensional vibrational model. They expanded the time-varying directional coefficients coming from the milling kinematics into Fourier series. Moreover, Altintaş and Ko [4] presented a frequency domain and chatter stability prediction theory for plunge milling operations. Recently, many other researchers have also studied the impact of the cutting parameters and of the vibratory behaviour in milling process on the surface quality. For example, Sukhev et al. [5] investigated experimentally the effect of cutting parameters on the tool vibration and surface roughness. It was shown that the feed rate that is the most dominating parameter affecting the surface roughness, whereas the cutting speed parameter is the most influent in tool vibration. Sheth and George [6] and Simunovic et al. [7] studied also the effect of the machining parameters during the milling operation on surface flatness and roughness.

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Chaari et al. [8] modelled a procedure for the quantification of geometric defects considering the machining dynamic effects on the milled surface quality.

Indeed, when a milling workpiece is flexible, the vibratory behaviour generated during machining is crucial. Consequently, it is difficult to mitigate this vibratory behaviour, especially when the excitation frequency is near to the resonance. Hence, it is important to use a passive absorber (TMD) in order to attenuate the dynamic response of a milled workpiece and decrease the dynamic effect on the resulting surface flatness. The passive absorber is one of the passive control techniques recently used in many machining applications. Several researchers studied the integration of passive absorbers into the milling mounts in order to reduce machining vibrations and to improve the stability of the milling tool. Sims [9] demonstrated the performance of his new analytical solution to vibration absorbers introduced to solve the chatter problem. So, he showed a great improvement in the critical limiting depth of cut compared to the previous tuned vibration absorber. While Wang [10] introduced a nonlinear tuned mass damper in order to improve the machining chatter suppression. Hamed et al. [11–13] studied also the problem of chatter suppression in the milling process by introducing linear and non-linear models of passive vibration absorbers in order to achieve better surface quality and larger material removal rate. Recently, Gafsi et al. [14] modelled passive linear absorbers composed of a single mass-spring-damper implemented in the spindle. He showed that the absorbers decrease the vibration amplitude of the milling tool and reduce the cutting time when the feed rate was doubled. Indeed, few researchers have studied the passive absorbers effect on the machined surface flatness. For example, Sathishkumar et al. [15] studied the effect of an impact-damping absorber on the dynamic behaviour of a boring bar and the influence of the particle parameters on the machining surface roughness.

So, this paper presents a contribution to the improvement of the attenuation of the vibratory behaviour of flexible machined workpieces and consequently to the improvement of the flatness and roughness of the machined surfaces. Hence, a study of the effect of a passive absorber attached to a thin workpiece on the milled surface flatness is presented. Then, a comparison of the simulated values of the machined surface quality (flatness and roughness) without and with a passive absorber (attached to the machined surface) is also presented. Indeed, this comparison confirms the efficiency of the passive vibration suppression technique to attenuate the vibratory level of a flexible machined workpiece, especially when the excitation frequency is near to the resonance.

2. Formulation of the problem

2.1. Dynamic modelling

This work investigates the impact of a linear absorber attached to a thin workpiece on the flatness of a machined surface. The dynamic behaviour of the thin workpiece excited by the cutting forces is modelled when a vibration absorber is attached to the milled workpiece. The motion of the global system (the thin workpiece and the absorber) is governed by the dynamic equation derived using Lagrange's formulation:

$$[M_{\mathrm{syst}}]\{\ddot{X}\}+[C_{\mathrm{w}}]\{\dot{X}\}+[K_{\mathrm{syst}}]\{X\}=\left\{F(t)\right\} \tag{1}$$

where $\{X\}$ is the nodal displacement of the workpiece, $[C_{\mathrm{w}}]$ is the damping matrix of the workpiece, and $\{F(t)\}$ is the external force applied to the system such as cutting forces and clamped forces. $[M_{\mathrm{syst}}]$ and $[K_{\mathrm{syst}}]$ represent, respectively, the mass and the stiffness matrix of both the workpiece and the absorber.

$$[M_{\mathrm{syst}}]=[M_{\mathrm{w}}]+[M_{\mathrm{a}}] \tag{2}$$

$$[K_{\mathrm{syst}}]=[K_{\mathrm{w}}]+[K_{\mathrm{a}}] \tag{3}$$

2.2. Cutting force model

For a face-milling operation, we have adapted the cutting force models modelled by Tang and Lu [16] and Altintaş and Budak [3]. The cutting force consists of three components in the tangential, radial, and axial directions, which are denoted by F_{t}, F_{r}, and F_{a}, respectively.

$$F_{\mathrm{t}}=K_{\mathrm{t}}\cdot A \tag{4}$$

knowing that: $A=h(t)\cdot b$

$$F_{\mathrm{r}}=K_{\mathrm{r}}\cdot F_{\mathrm{t}} \tag{5}$$

$$F_{\mathrm{a}}=K_{\mathrm{a}}\cdot F_{\mathrm{t}} \tag{6}$$

A is the instantaneous cross section area of cut, $h(t)$ is the instantaneous undeformed chip thickness, and b is the nominal undeformed width of the chip. K_{t}, K_{r}, and K_{a} are the tangential, radial, and axial milling coefficients, respectively.

$$h(t)=\begin{cases} f_z \sin\phi(t)\sin K_r & \text{if } \phi_{\rm st}\le\phi(t)\le\phi_{\rm ex} \\ 0 & \text{if } \phi_{\rm st}>\phi(t) \text{ or } \phi_{\rm ex}<\phi(t)\end{cases} \tag{7}$$

$$b=a_{\rm p}/\sin(K_{\rm r}) \tag{8}$$

f_z is the feed rate by tooth, $a_{\rm p}$ is the depth of cut, $K_{\rm r}$ is the tool entering angle, t is the cutting time, $\phi_{\rm st}$ is the initial cut-in angle, $\phi_{\rm ex}$ is the end cut-out angle, and then $\phi(t)$ is the instantaneous rotation angle of tool cutting.

$$\phi(t)=(2\pi N/60)\cdot t \tag{9}$$

$$\phi_{\rm st}=\pi/2-\arcsin(2d/D) \tag{10}$$

$$\phi_{\rm ex}=\pi/2+\arcsin\big(2(W-d)/D\big) \tag{11}$$

knowing that W is the cutting width of workpiece, N is the rotation speed of the cutting tool, d is the vertical distance from the tool's insert point to the centre of the tool, and D is the diameter of the cutter tool. The instantaneous cutting force acting on an individual insert (i) engaged in cutting at an instant can be expressed by:

$$\begin{cases} F_x(i,\phi)=F_{\rm t}(i,\phi)\sin(\phi)-F_{\rm r}(i,\phi)\cos(\phi) \\ F_y(i,\phi)=F_{\rm t}(i,\phi)\cos(\phi)+F_{\rm r}(i,\phi)\sin(\phi) \\ F_z(i,\phi)=F_{\rm a}(i,\phi)\end{cases} \tag{12}$$

2.3. Quantification procedure of the flatness and roughness of the machined surface

In machining, the geometric defects are generally generated by small kinematics displacements, static and also dynamic displacements of the machined workpiece.

The presented quantification procedure of the flatness and roughness is based on the following assumptions:

- the Hertzian model holds for all contacts between workpiece and fixturing system;
- Coulomb friction is present at all contacts between workpiece and fixturing system;
- all deformations are linearly elastic and small.

Indeed the procedure presented in Fig. 1 consists in:

- describing the machined surface by N points based on the principle of three-dimensional measurement machines (MMT);
- quantifying the geometric defects caused by fixture errors and form defects of datum surfaces based on a tensorial model developed by Chaari et al. [8];
- quantifying the static and dynamic displacements of the machined surface at each tool displacement based on the dynamic model. The coordinates of the machined surface points (which are stored in a matrix) will be also updated for each tool displacement;
- quantifying the flatness and roughness of the machined surface using algorithms based on geometric interpretation and linear least squares estimation methods.

2.3.1. Geometric model of the machined surface

The geometry model used is based on the principle of three-dimensional measurement machines (MMT). It consists in describing the machined surface by a set of surface points. Then, the coordinates of these points are extracted from the mesh model generated from the FEA software and are also stored in matrix X (Fig. 2):

$$X=\begin{bmatrix} p_1 & p_2 & \ldots\ldots & p_k\end{bmatrix}_0=\begin{bmatrix} x_1 & \ldots & x_{\rm m} \\ y_1 & \ldots & y_{\rm m} \\ z_1 & \ldots & z_{\rm m}\end{bmatrix} \tag{13}$$

2.3.2. Modelling of the geometric defects

The geometric defects caused by the fixture errors are modelled by the homogeneous matrix $\mathrm{T_f}$ [8]:

$$\mathrm{T_f}=\begin{bmatrix} 1 & \delta\theta_{\rm zf} & -\delta\theta_{\rm yf} & \delta x_{\rm f} \\ -\delta\theta_{\rm zf} & 1 & \delta\theta_{\rm xf} & \delta y_{\rm f} \\ \delta\theta_{\rm yf} & -\delta\theta_{\rm xf} & 1 & \delta z_{\rm f} \\ 0 & 0 & 0 & 1\end{bmatrix} \tag{14}$$

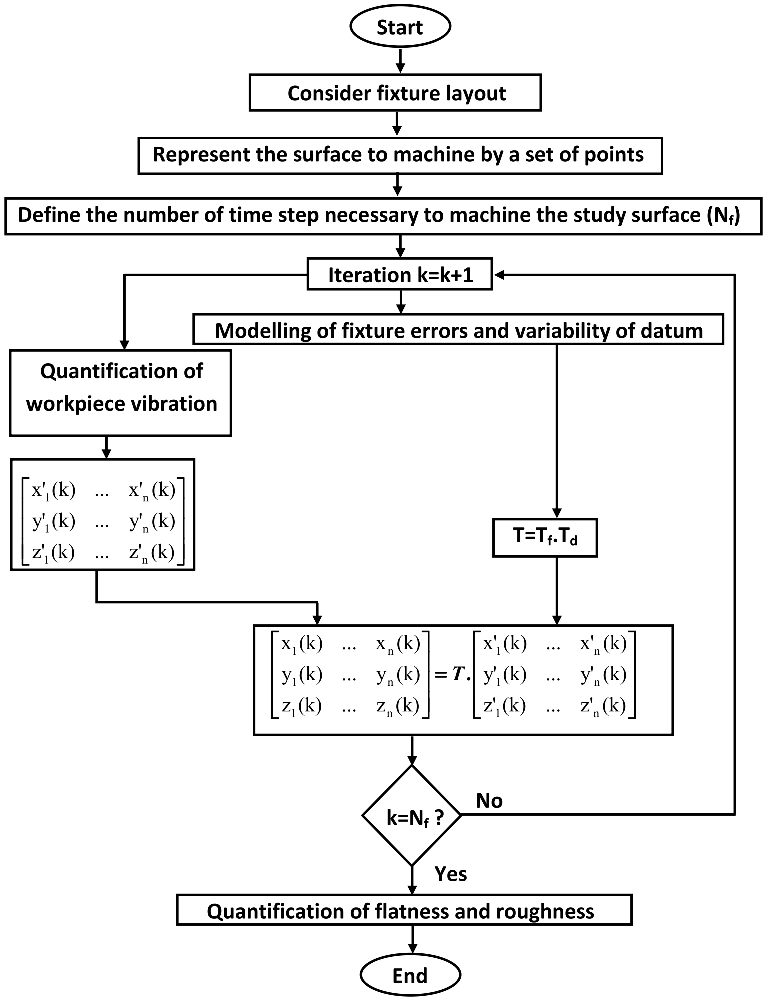

Fig. 1. Quantification procedure of flatness and roughness accounting for the vibratory behaviour of the machined workpiece.

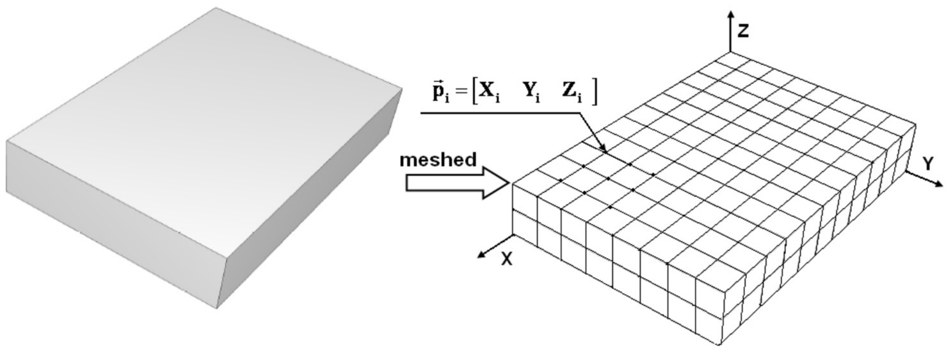

Fig. 2. Geometric representation of the machined surface [8].

Then the geometric defects caused by form defects of datum surfaces are also modelled by the homogeneous matrix T_d [8]:

$$T_d = \begin{bmatrix} 1 & \delta\theta_{zd} & -\delta\theta_{yd} & \delta x_d \\ -\delta\theta_{zd} & 1 & \delta\theta_{xd} & \delta y_d \\ \delta\theta_{yd} & -\delta\theta_{xd} & 1 & \delta z_d \\ 0 & 0 & 0 & 1 \end{bmatrix} \tag{15}$$

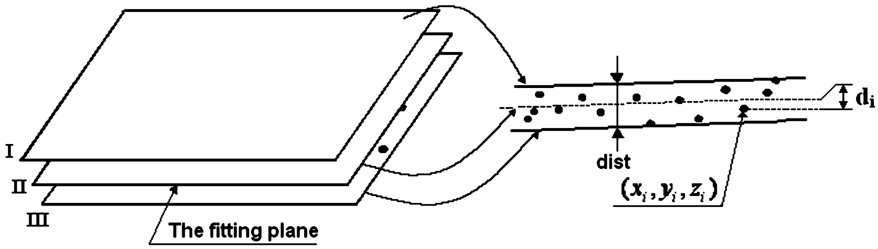

Fig. 3. Geometric interpretation of the quantification method of the flatness [8].

Table 1
Coordinates of the applied points of the fixture and clamps [8].

Fixtures	Coordinates (X, Y, Z) (mm)	Clamps	Coordinates (X, Y, Z) (mm)
L1	(60, 20, 0)	C1	(50, 0, 11)
L2	(60, −20, 0)	C2	(−50, 0, 11)
L3	(−60, 0, 0)	C3	(60, −30, 7)
L4	(60, 30, 7)	C4	(−60, −30, 7)
L5	(−60, 30, 7)		
L6	(70, 0, 7)		

The homogeneous matrix T_f and T_d are generated using the Monte Carlo simulation given the tolerance of fixture and the variability datum surfaces, respectively.

The geometric defects caused by different variation sources can be superposed under a small displacement assumption. The superposition of geometric defects caused by fixture errors and variability datum surfaces are modelled by multiplying the two homogeneous transformation matrixes: T_f and T_d, whereas the dynamic displacements of the machined surface are represented by the variability of the instantaneous coordinates of the machined surface points at each time step. So, the generated surface at the ith time step is represented as:

$$\begin{bmatrix} x_1(i) & \dots & x_n(i) \\ y_1(i) & \dots & y_n(i) \\ z_1(i) & \dots & z_n(i) \end{bmatrix} = T_f \cdot T_d \begin{bmatrix} x'_1(i) & \dots & x'_n(i) \\ y'_1(i) & \dots & y'_n(i) \\ z'_1(i) & \dots & z'_n(i) \end{bmatrix} \quad \text{for } i = 1 \dots n \tag{16}$$

2.3.3. Quantification algorithm of flatness defects

The quantification algorithm of the geometric flatness defects is based on the linear least squares estimation method and some geometric interpretations. Assuming that the fitting plane of the surface points is represented by a point (x_f, y_f, z_f) (point of the fitting plane) and the normal vector (a, b, c), the deviation from a surface point (x_i, y_i, z_i) to this plane is:

$$d_i = a(x_i - x_f) + b(y_i - y_f) + c(z_i - z_f) \quad i = 1 \dots n \tag{17}$$

The estimated flatness is defined as the minimum distance between two parallel planes (that are parallel to the least square fit plane) that covers the measured surface. For a set of n points, the distance between two parallel planes is:

$$dist = \max(d_i) - \min(d_i); \qquad i = 1 \dots n \tag{18}$$

The deviation ($dist$) has to be smaller than the tolerance specification for the flatness. Fig. 3.

3. Studied case

A numerical simulation is presented to show the considerable impact of a passive absorber on the flatness of a milled surface. For this reason, an end-milling operation is conducted on the top surface of a hat-shaped workpiece with 3-axis machining centre. The material of the workpiece is aluminium Al-7085. The cutting tool with a diameter of 60 mm has six triangles, CVD coated inserts with a face cutting edge angle of 30°, a lead angle of 0°, an axial rake angle of 5°, and a radial rake angle of 1°. The simulated workpiece is the same example as that treated by Zhong and Hui [17] and Zhong et al. [18] in order to validate our simulated results without absorber. Table 2, Fig. 4, Table 3, Fig. 5, Table 4.

A 3-2-1 fixture layout and four clamps were used in order to ensure a stable fixturing for the studied workpiece. Table 1, Fig. 6.

4. Results and discussion

In this section, the finite-element method and Matlab software are used to simulate the vibratory behaviour of the simulated workpiece without and with the presence of vibration absorber.

Table 2
Cutting conditions.

V_f: feed rate (mm/s)	a_p: depth of cut (mm)	Z: number of inserts	N: spindle speed (rpm)
5.2	0.5	6	3200

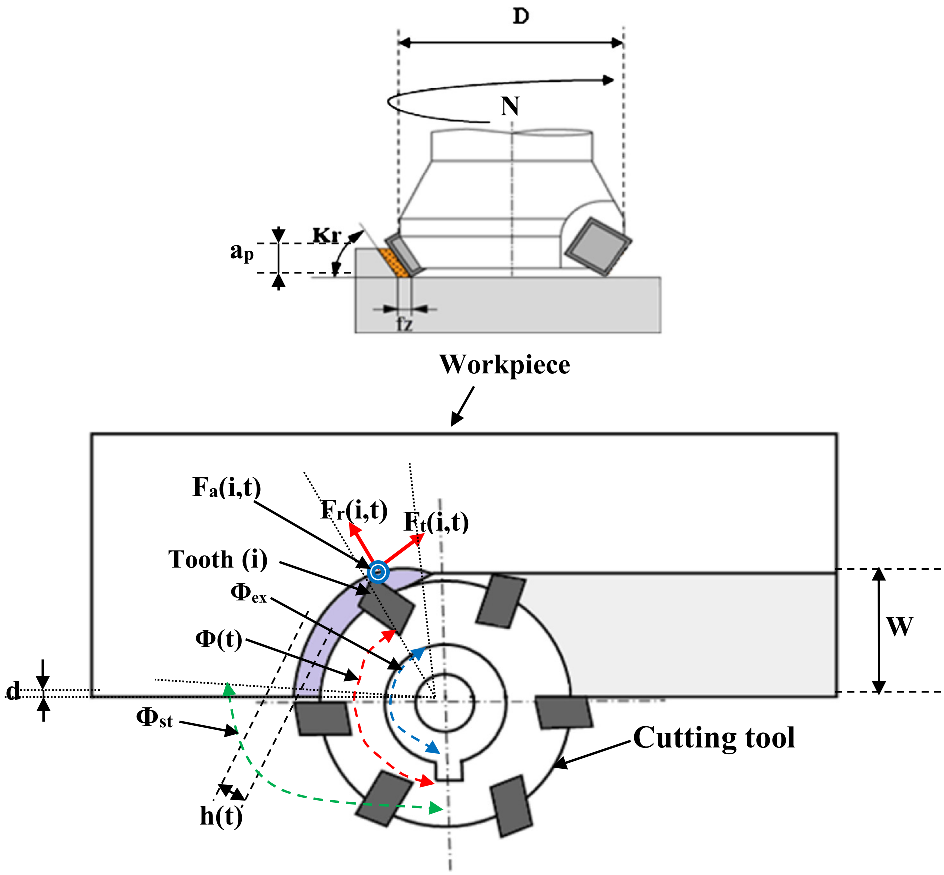

Fig. 4. Schematic representation of the geometry parameters of the cutting tool and the machined workpiece.

Table 3
Definition of the geometry parameters of the milling operation.

K_r (rad): tool entering angle	$\pi/6$
D (mm): diameter of the cutting tool	60
W (mm): cutting width of workpiece	35
d (mm): vertical distance from the tool's insert point to the centre of the tool	30
ϕ_{st} (rad): initial cut-in angle	0
ϕ_{ex} (rad): end cut-out angle	$\pi/2$
$\phi(t)$: instantaneous rotation angle of tool cutting	Variable according to time
$h(t)$: instantaneous undeformed chip thickness	Variable according to time

The first four natural frequencies of the simulated workpiece are 313.9 Hz, 461.1 Hz, 649.8 Hz and 762.7 Hz, respectively.

The simulated milling forces, shown in Fig. 7, are obtained from force model derived from [16,14]. When the excitation frequency is close to the resonance, the vibratory behaviour of the simulated workpiece is critical. In order to show the effectiveness of the absorber to mitigate the vibrations generated during milling, the spindle speed is chosen near the first workpiece natural frequency.

Figs. 8 and 9 show the geometric orientation and an estimation of the topography of the machined surface for two cases, in the absence and presence of absorber, respectively.

Table 5 shows the values of the estimated flatness of the machined surface in the absence and presence of the absorber.

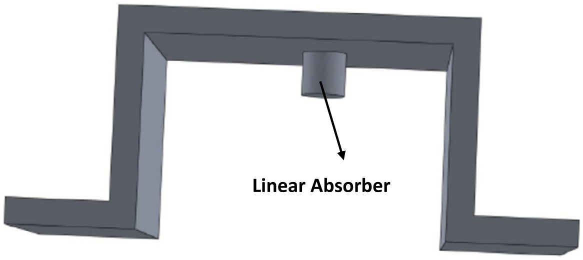

Fig. 5. Location of the linear absorber.

Table 4
Parameters of the linear vibration absorber.

Absorber mass	$M_a = 0.06$ kg
Spring stiffness	$K_a = 5910$ N/m

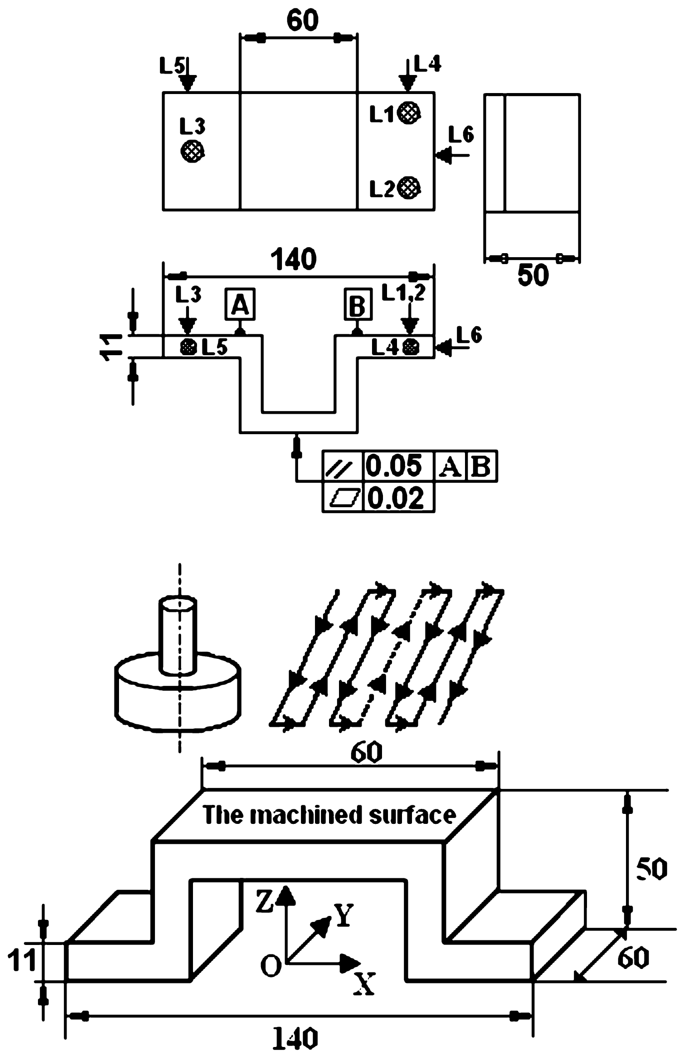

Fig. 6. Simulated example [17,8].

The obtained results show that the studied linear absorber is efficient in mitigating the vibrations generated during machining. Consequently, the attenuation of the vibratory behaviour of the machining workpiece improves considerably the flatness and roughness of the machined surface.

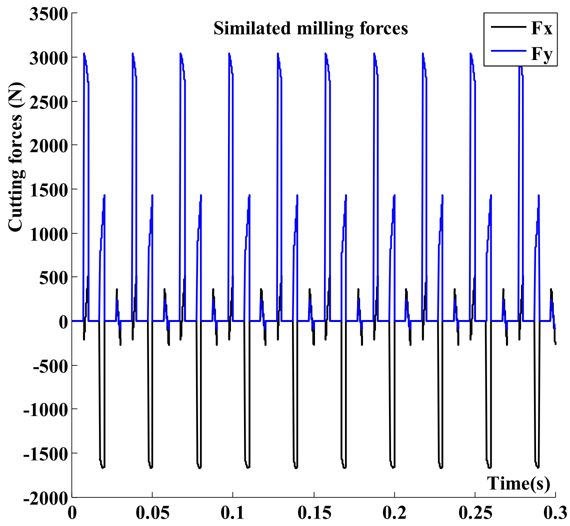

Fig. 7. Simulated milling forces.

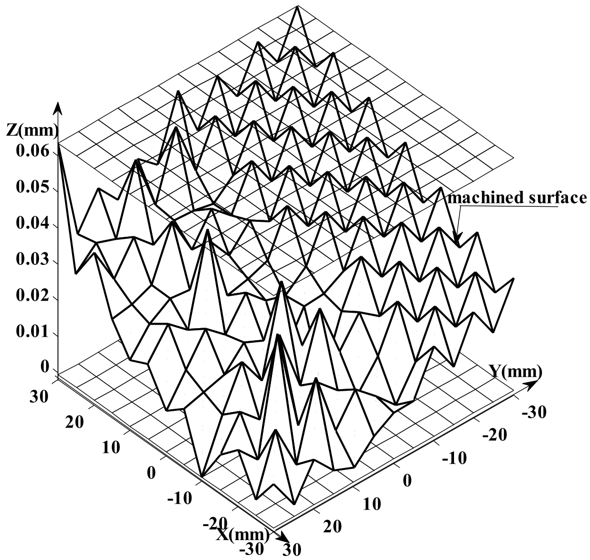

Fig. 8. Orientation and topography of the machined surface in the absence of absorber.

Table 5
Estimation of the flatness of the machined surface.

	Flatness	Roughness (Ra)
Without absorber	0.067 mm	0.0227 mm
With absorber	0.0198 mm	0.00157 mm

5. Conclusion

In this paper, a simulation of the vibratory behaviour of a flexible workpiece is studied when it is attached to a linear absorber in order to highlight its efficiency in improving considerably the flatness and roughness of a machined surface. Indeed, the flatness of the machined surface was quantified based on the linear least squares estimation method and some geometric interpretations. The simulated results show that the use of a linear absorber is an efficient solution when the vibratory behaviour of a flexible workpiece is critical due to an excitation close to the resonance.

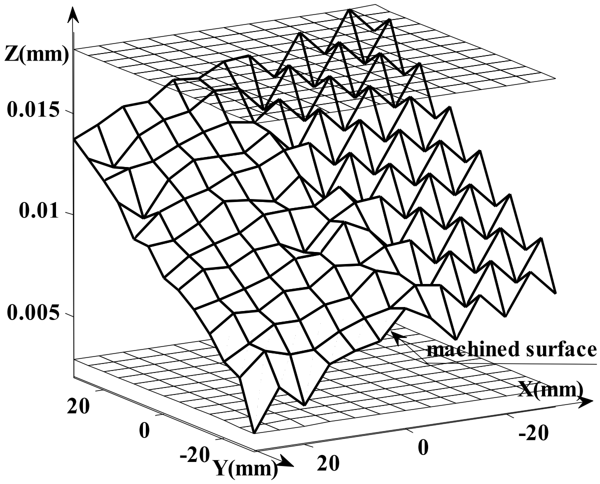

Fig. 9. Orientation and the topography of the machined surface when the absorber is attached to the workpiece.

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