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A device to reduce positioning errors due to the machine tool compliance

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Abstract: One of the main problems that large machine tool designers must face is the onset of deflection and vibration due to the forces involved during the process: the own weight of the elements of the machine, the cutting forces and the inertial forces. The last ones are especially relevant in the case of large and heavy machine tools with cantilever elements as rams or columns. To counteract these forces, the present work proposes the use of an inertial device, based on the rotation of two eccentric masses around the same axis. Hence, a simplified dynamic model of a column is first developed, and tested during a standard face milling operation. Then, the dynamic model of the device is also developed, and a Computed Torque Control is proposed to govern the action of the device. The simulations show that the use of this device has the potential to reduce drastically the effect of the inertial forces and even the lower frequency components of the cutting forces.

Keywords: Positioning error, Compliance, Machine Tools, Mechatronics

1. Introduction

The development of the oil&gas and renewable energies sectors has led to a strong demand of large machine tools. The design of these machines involves positioning heavy and slender cantilever columns and rams that suffer positional errors from deflection and vibration due to several factors: their own weight, the high inertial forces involved during the motion, and the cutting forces [1,2]. The problem of the weight is usually solved using balancing masses and compensated guides. However, the problems from the inertial and cutting forces are frequently solved by reducing the productivity. That is, to reduce the effect of the inertial forces involved the maximum acceleration and jerk are limited during the motion planning, so the machine becomes less agile and productive than what it could be [3]. On the other hand, to reduce the deflection and vibration from the cutting forces, the material removal is reduced. An effective solution to overcome the problem of vibration, especially chatter vibration, is to use passive or active dampers whose effect is seen at frequencies around the most limiting modes of the machine. The present work analyses the viability of an inertial device with two independent eccentric masses whose rotation introduces inertial forces that try to overcome the lower frequency vibrations due to the inertial and cutting forces in these machines.

The objectives of this paper are two. The first one is to estimate, based on simulations, the positioning errors that can occur in a typical large machine tool drive and to evaluate the reduction that would be achieved if it were possible to incorporate a control force on the column. The second objective is to propose a controlled mechanical device that can exert forces and control torques of such a value that the positioning errors are reduced to a practically negligible level.

2. Methodology and results

We are going to use a simplified dynamics 2D model of the linear drive, assuming the whole column as a flexible element as shown in the scheme of figure 1. The tip of the tool is located at the B end of the massless bar AB, concentrating the mass of the column in that point. In order to model the compliant column, it is considered that it can rotate with respect to the machine carriage at point A and a spring and a damper are added to it to condense the rigidity and damping of the whole structure.

The values of the mass, stiffness and damping have been selected from the modal parameters obtained from the frequency response functions measured at the spindle nose of two large machine tools: one is a moving column machine and the other is a moving column with a box-in-box structure [4]. In both machines the most limiting modes when milling in the X-Y directions came from the combined deflection of the column and the ram. After analyzing those main modes, a mode with a frequency of 15 Hz, stiffness of $4.04 \cdot 10^7$ N/m, modal mass of 4550 kg and damping ratio of 0.05 was selected for testing in this work.

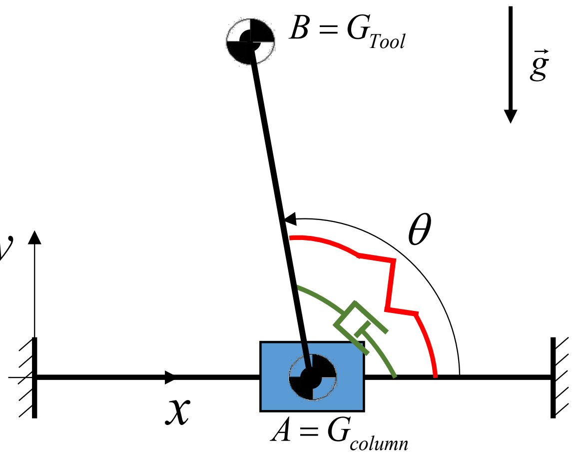

Figure 1. Scheme of the original drive.

In addition, a horizontal force representing the machining force will be applied on the tip of the tool in B. This force has been calculated using a mechanistic model [5], where the cutting force coefficients were experimentally obtained from face milling tests on AISI 1045 steel with a Mitsubishi B09 tool of 125 mm diameter and 9 cutting inserts with 45º of lead angle, being the tangential, radial and axial shear cutting coefficients of 1685 N/mm^2, 763 N/mm^2 and 129 N/mm^2, respectively. The cutting conditions simulated were a face milling with a depth of cut of 4 mm, a radial immersion of 100mm in down-milling, 3000 rpm of spindle speed and a feed per tooth of 0.22 mm. The cutting force in the feed direction has an average value of -180 N with peaks up to 712 N.

The simplified model has been done in natural and mixed coordinates [6] and a formulation of the dynamics based on [7] has been used, which can be seen in equations (1) and (2):

$$M \cdot \ddot{q} + \Phi_q^T \cdot \lambda = Q \tag{1}$$

$$\Phi = 0 \tag{2}$$

where

M is the mass matrix

q is the generalized coordinates vector
Φ is the vector of kinematic constraint equations
λ is the Lagrange's multipliers vector
Q is the generalized forces vector

The indicated formulation uses a projection of positions and velocities [7] combined with a Newmark-beta numerical integrator (in particular, a trapezoidal rule) to generate a robust, efficient and stable algorithm for general purpose simulations. This formulation has been programmed in MATLAB R2019b with a 10^{-5} s integration time step.

As an additional condition, a typical motion of a large machine tool drive has been applied to the carriage, x_A coordinate. Here, it is assumed that the position control of the drive is precise enough to perform the programmed motion with precision. The motion has the following features. First, it is a square sine velocity profile, where there is an initial acceleration stage with the square sine, followed by a motion with constant feed speed, followed by a deceleration with a symmetrical square sine. The displacement goes from zero position to 0.01 m. The feed speed is 6 m/min, that is 0.1 m/s, and the acceleration/deceleration is limited to 0.5G. The same parameters have been tested also using another jerk limited strategy in the acceleration/deceleration stages, a trapezoidal acceleration profile.

2.1. Simulation of the original drive

The simulation results are shown in figure 2. There, the positioning error varies over time, showing a vibration of the column with a maximum value up to 0.6 mm. This vibration is mainly due to the inertia force associated with the column mass. This vibration depends on the type of acceleration profile, since it has been observed that the trapezoidal acceleration profile generates an error with a similar shape but a slightly higher error.

Comparing the main forces involved, see figure 3, the inertial ones and the cutting forces, the milling force does not seem to have a significant influence on the positioning error, given the cutting conditions tested and the mass of the machine. However, it has been found that the introduction of the serrated profile of the machining force generates many difficulties for the integrator, since the discontinuities make the problem to be solved become stiff. For this reason, the cutting force has been simplified assuming a constant value, where the mean and maximum values have been tested.

2.2. Simulation of the drive with a controlled force to reduce positioning errors

In this case, the model includes a control force which is introduced on the x_B coordinate, for simplicity and without loss of generality. Then, equation (1) include an additional vector of controlled forces, as shown in equation (3).

$$M \cdot \ddot{q} + \Phi_q^T \cdot \lambda = Q + Q_a = Q + B \cdot U \tag{3}$$

The vector of control forces is constructed with a boolean projection matrix $\boldsymbol{B}$ of the control forces and with the vector $\boldsymbol{U}$ of control forces (in this case, it only contains the force applied on the x_B coordinate). The value of the control force is calculated using a computed torque control (CTC) scheme [8], as shown in equation (4):

$$U = (H \cdot M^{-1} \cdot B)^{-1} \cdot (\ddot{y}_{des} + K_D \cdot (\dot{y}_{des} - H \cdot \dot{q}) + K_P \cdot (y_{des} - H \cdot q) - H \cdot M^{-1} \cdot Q) \tag{4}$$

where

H is a boolean matrix of projection of the desired motion.
y_{des} , $\dot{y}_{des}$ and $\ddot{y}_{des}$ are the vectors with the coordinates on which the desired motion is applied and their derivatives with respect to time.
K_D and K_P are diagonal matrices with the gains associated with the error in each coordinate on which the desired movement is imposed.

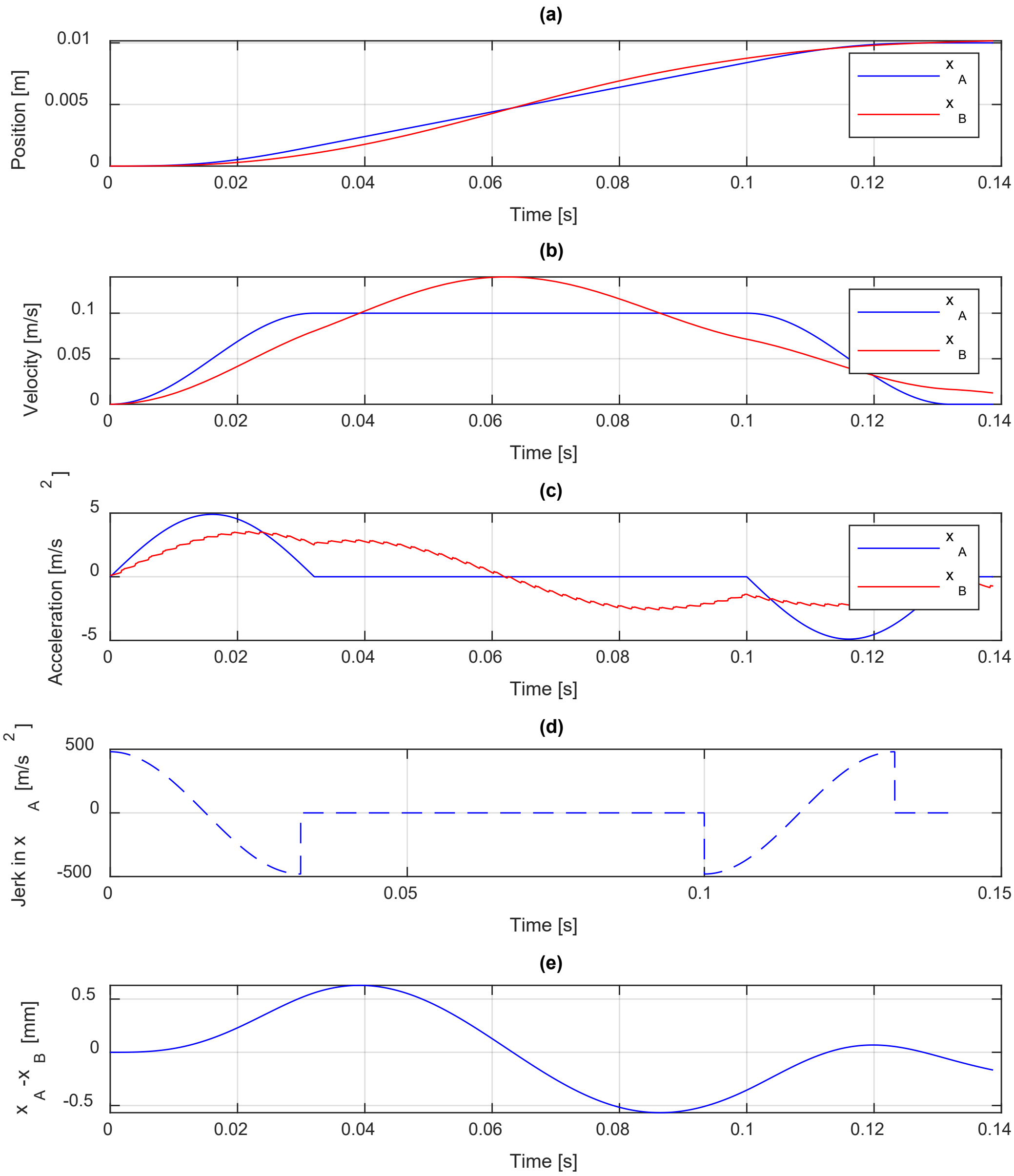

Figure 2. Movement of A (carriage) and B (tool) in the original drive. **(a)** Position. **(b)** Velocity. **(c)** Acceleration. **(d)** Jerk. **(e)** Position error x_A-x_B.

The coordinates on which the desired motion is imposed must be related to the bending of the column. For simplicity, the coordinate θ has been chosen since it can be estimated with some precision by means of strain gauges placed on the column. But it can be applied to any other coordinate easily measurable on the column. In order to eliminate the error due to bending, the θ coordinate should maintain a constant value of $\pi/2$ rad.

The gains (in this case, only one) are related as shown in equation (5) to achieve a critical damping. In this way, it is only necessary to define the proportional gains, which are directly related to the admissible positioning errors.

$$K_{Di} = 2 \cdot \sqrt{K_{Pi}} \tag{5}$$

For this case, a value of 10^5 has been set for the proportional gain.

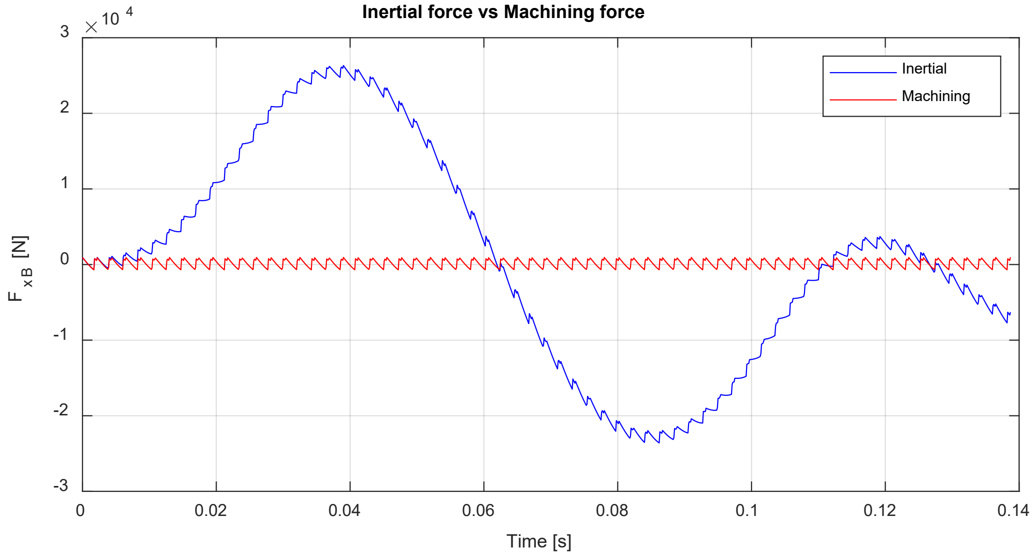

Figure 3. Inertial forces vs. Milling force in feed direction.

Figure 4 shows the results of the simulation of the drive movement with the control force. It can be seen that the magnitude of the positioning error is drastically reduced. A closer look at the contribution of the control force terms shows that the term relative to the desired acceleration in figure 2(c) (and consequently proportional to the inertia force of the column) is the dominant one.

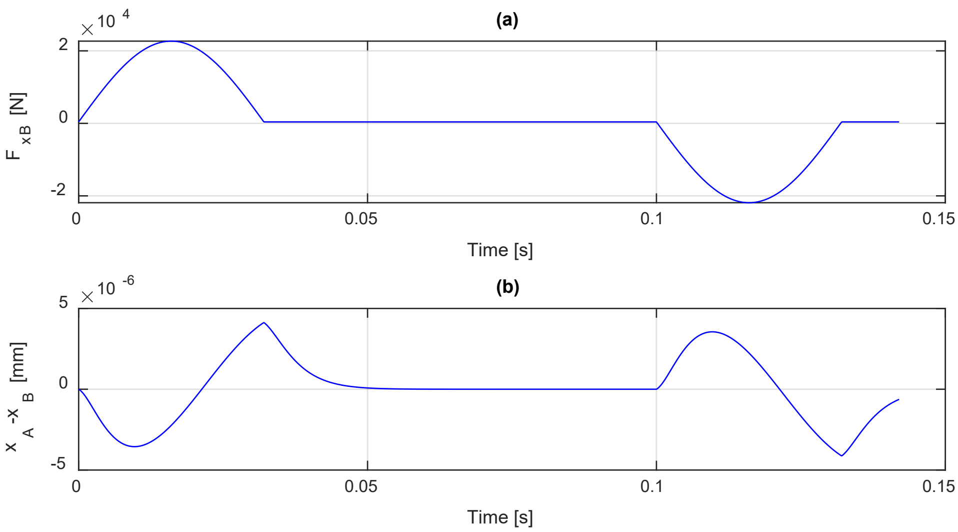

Figure 4. Drive with a controlled force to reduce the error. **(a)** Force in x_B. **(b)** Position error x_A-x_B.

Another data of great interest is the value taken by the control force, which can have a considerable value and its time evolution, which is clearly linked to the acceleration of the drive. This affects the device to be implemented to exert this control force.

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2.3. Simulation of the drive with the device to reduce positioning errors

The device shown in figure 5 is composed of two equal discs that rotate with respect to the same geometric axis (point C) - for simplicity - and whose center of mass is located outside the axis of rotation. Both discs are driven by rotation actuators and can rotate with respect to the column in an independent and controlled manner.

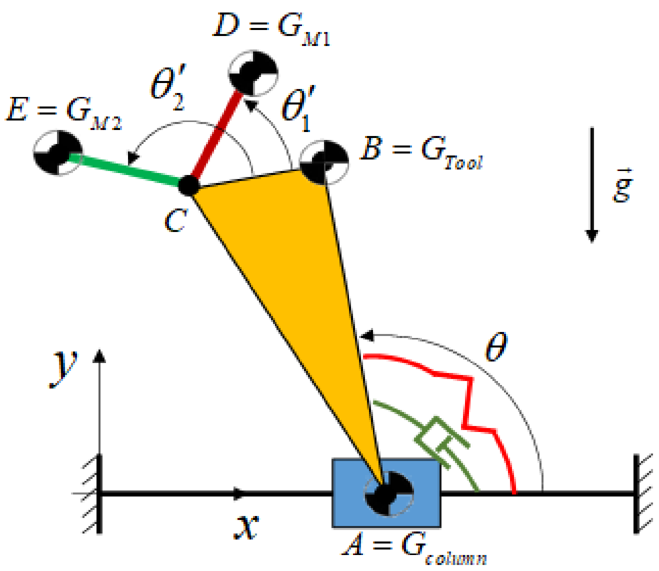

Figure 5. Scheme of the drive with the device.

The formulation used is the one indicated in equation (3) but the CTC scheme is not the same since in this case, the controlled system has 3 degrees of freedom but only two actuators. This change is translated in the use of the pseudoinverse matrix to solve the problem of the control torques [5] in the rotating actuators, as shown in equation (6):

$$U = (H \cdot M^{-1} \cdot B)^{+} \cdot (\ddot{y}_{des} + K_D \cdot (\dot{y}_{des} - H \cdot \dot{q}) + K_P \cdot (y_{des} - H \cdot q) - H \cdot M^{-1} \cdot Q) \tag{6}$$

In this study, two discs of 50 kg and whose centres of mass are separated 0.5 m from the centre of rotation C have been used. The proportional gain is maintained at a value of 10^5.

Figure 6 shows the results of the simulation of the drive motion with the positioning error reduction device. It can be seen that the magnitude of the positioning error is also drastically reduced with respect to the case without the device, but not as much as in the case of pure control force in figure 4(b), besides having a different variation over time. The reasons for this difference are to be found in the fact that the device uses the inertial forces of the discs to generate an action equivalent to the necessary control force. The modulus of these inertial forces depends on the velocity and acceleration of rotation of the discs while their direction depends on their angular position, making the composition of this equivalent force variable over time. Taking this into account, the capacity of the device to generate these inertial forces is limited, on the one hand, by the possibilities of the actuator (velocity, acceleration and torque that can be achieved) and, on the other hand, by the strength of the mechanical elements to be used, which cannot be massive either.

Finally, it can be observed that the velocity of the two actuators is not zero at the end of the simulated motion. If the simulation period were extended, both actuators would eventually stop but this condition may not be possible to apply during a machining process where machining operations and trajectory changes are continuous. To avoid this situation there are several possibilities (incorporation of a third disc, modification of the control,...) that would require further study in future studies.

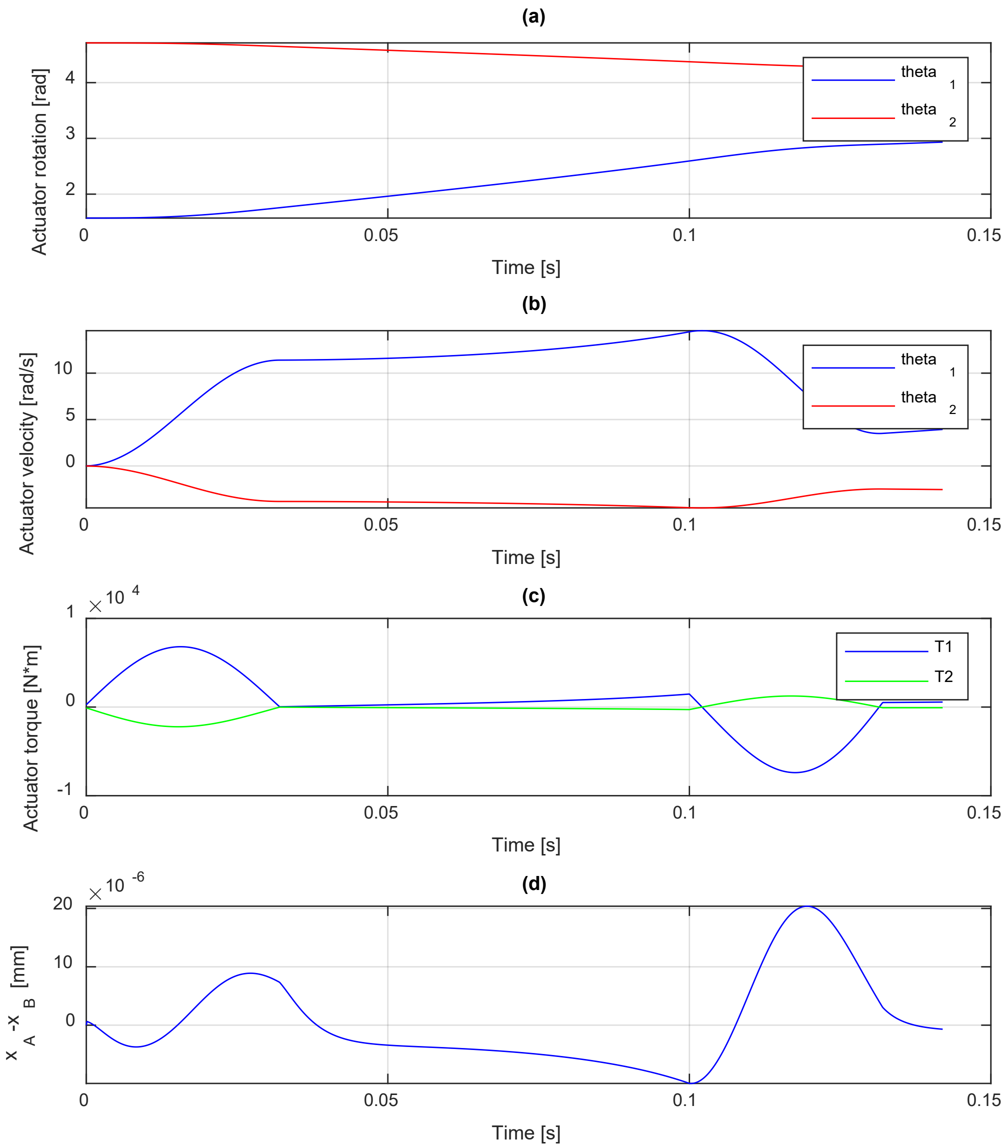

Figure 6. Drive with the device. **(a)** Actuator rotation. **(b)** Actuator velocity. **(c)** Actuator torque. **(b)** Position error x_A-x_B.

3. Conclusions

In this paper, a simplified 2D model has been presented for the estimation of positioning errors due to the bending in the column of large machine tools and its use to study devices for the reduction of such positioning errors. In particular, a device capable of significantly reducing the positioning error at the tool tip is proposed. The adjustment of the control of this device is very simple since it has only one parameter and, in addition, it is very robust, producing acceptable results in a very wide range of this parameter.

The development and installation of this new device requires the previous characterization of the rigidity and damping characteristics of the machine tool column and involves the installation of an additional sensor that measures the column deformation in real time, such as strain gauges.

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As future research lines, it is proposed to measure the deformation of the column through the acceleration of a significant point of the column, since this magnitude is easier to measure than the bending of the column. This change would lead to the simplification of the control associated with the rotation actuators. Another interesting research line arises from considering the possibility of making the column lighter and more flexible and placing one or more linear actuators (such as hydraulic cylinders) on the carriage to stiffen the column in a controlled manner when necessary.

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References

[1] Zulaika J J, Campa F J and López de Lacalle L N 2011 An integrated process-machine approach for designing productive and lightweight milling machines *International Journal of Machine Tools and Manufacture* **51** (7) pp 591-604
[2] Uriarte L, Zatarain M, Axinte D, Yagüe-Fabra J, Ihlenfeldt S, Eguia J and Olarra A 2013 Machine tools for large parts *CIRP Annals* **62** (2) pp 731-50
[3] Ansoategui I and Campa F J 2019 Mechatronic model based overshoot prediction and reduction in servodrives with compliant load *Mechanism and Machine Theory* **137** pp 227-236
[4] Zulaika J J 2011 *Metodología para la concepción de fresadoras de gran volumen productivas y ecoeficientes* (University of Basque Country UPV/EHU, Doctoral Thesis)
[5] Altintas Y 2000 *Manufacturing Automation: Metal Cutting Mechanics, Machine Tool Vibrations, and CNC Design* (Cambridge: Cambridge University Press)
[6] García de Jalón J and Bayo E 1994 *Kinematic and Dynamic Simulation of Multibody Systems – The real-time challenge* (New York: Springer)
[7] Von Schwerin R 1999 *Multibody System Simulation, Numerical methods, algorithms and software* (Berlin: Springer-Verlag)
[8] Mouzo F, Lugrís U, Pâmies-Vilà R and Cuadrado J 2018 Skeletal-level control based forward dynamic analysis of acquired healthy and assisted gait motion *Multibody System Dynamics* **44** (1) pp 1-29