Proceedings of the
44th IEEE Conference on Decision and Control, and
the European Control Conference 2005
Seville, Spain, December 12-15, 2005

Improvement of the Performance in Machine Tools by Means of State Space Control Strategies

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Abstract— **This paper describes new control techniques based on state space observers that have been developed to improve the precision and dynamic behaviour of machine tools. To improve precision, the position of the tool centre point (TCP) is estimated and used as position feedback instead of the position of the slides. Dynamic behaviour is improved by estimating the acceleration of the TCP and feeding this noiseless signal into the control system.**

These techniques provide a simple way to achieve considerable improvements without using complex and expensive TCP position measurement devices. Tests on actual machines have shown that errors produced by deflections between the slide and the TCP can be reduced by 70%.

I. Introduction

THE high speed and acceleration rates of today's machines give rise to major errors in trajectory that are translated into a lack of precision in the part machined. The use of feed-forward techniques improves matters, but these techniques do not correct errors due to deformation in drive systems or in structures.

A. Control architecture in CNCs

Today's CNCs use the classic cascade control structure (current, velocity and position loop) [1]. As an alternative or supplement to this technique, new control strategies have appeared in recent years to improve performance in high-speed drives. Tomizuka [2] has developed a zero phase error tracking controller (ZPETC) by cancelling the stable dynamics of the servo drive in a feed-forward fashion. With the same purpose, using filtering to eliminate the high-speed components that the ZPETC controller generates, Weck and Ye [3] present the IKF control strategy. Wang, Ramond, Dumur et al. [4], [5], [6], [7] demonstrate the validity of predictive control in its numerous variants in the field of CNCs for machine-tools: generalized predictive control (GPC), direct adaptive generalized predictive control (DAGPC), constrained receding horizon predictive control (CRHPC), etc.

Van Brussel and Van den Braembussche [8] propose robust control schemes for drives with linear motors, based on $H\infty$ synthesis and on the sliding mode control technique. Altintas, Slotine, Zhu et al. [9], [10], [11] also propose the sliding mode control technique, which permits a robust control system against uncertainties in the drive's parameters, and compensates for external disturbances such as friction and cutting force. In this way, the main drawback of the classical controller design techniques, i.e. the sensitivity to modelling errors, as addressed by Yao et al. [12] is minimized. These same characteristics are obtained by Van Brussel et al. [13] by combining a non-causal filter in the feed-forward path in an extended pole placement method with the use of a disturbance observer in the feedback path. A common characteristic of machine-tools is that the natural frequencies of the resonances vary according to the relative positions of the machine's slides. W. Symens et al. [14] propose the gain-scheduling control approach, a system of control robust to these variations.

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B. Precision attained at the TCP

Independently of the performance, robustness characteristics and offsetting of disturbances that may be attained with the aforesaid control techniques, the position signal measured and fed back to the controller is usually provided by the linear scale, which is generally located at the base of the slide to be moved, i.e. far from the point whose position is to be controlled, from the tool (TCP) and from the workpiece. This means that even when drive controllers are optimally adjusted, the deformation caused by the flexible elements between the scale and the point to be controlled is not offset.

This flexible behaviour of the machine structures located beyond the scale gives rise to unwanted movements in the tool, which lie outside the control loops and therefore cause machining precision to decrease. Heisel and Feinauer [15] describe the dynamic causes that may influence the surface quality of machined workpieces and which are or may be located beyond the scale. Such causes include imbalance due to eccentricity of masses and to assembly tolerances, forces produced by the cutting process itself and vibrations arising during the acceleration phases. The loss of precision involved is considerable, especially in machines with slim-line moving parts such as the mobile rams on high-speed milling machines.

C. Solution proposed

To improve the precision and dynamic behaviour of machine tools, the use of an accelerometer located as close as possible to the tool is proposed. The accelerometer signal, combined with the measurement of the linear scale and the current fed to the motor, provides the means to develop a state space observer that estimates the position of the tool and its acceleration.

First, the observer is used to estimate the position of the TCP at each moment. That estimation is fed back to the position loop rather than that indicated by the linear scale. The dynamic behaviour of the machine beyond the location of the scale is thus included in the control loops.

Secondly, the observer is used to obtain a noiseless estimation of the acceleration of the TCP. Pritschow et al. [16] show how acceleration feedback improves drive dynamics by making the system more rigid in the face of external disturbances. Unlike these authors, who measure the acceleration signal via a Ferraris sensor and develop a combined current and acceleration control loop, we suggest estimating the acceleration signal via the aforementioned state observer and feeding back the acceleration signal as a correction within the velocity loop. The attraction of using the estimated TCP acceleration signal rather than the measured signal lies in the fact that a considerably lower noise level is involved, so the gain of the acceleration loop can be increased further without destabilising the system. Using this technique not only is the system made more rigid in the face of disturbances but also in some cases the level of damping is increased.

II. Estimating the Position of the TCP

The dynamic behavior of the mechanical elements between the slide and the TCP can be approximated using a simple model of two masses linked to each other by a damping spring assembly. In simplified form, M_1 represents the mass of the slide, M_2 the mass of the headstock and K and C are the stiffness and damping of the flexible element. u and y are the positions occupied by each mass, F is the force exerted by the motor on M_1 and P is the external disturbance on the system.

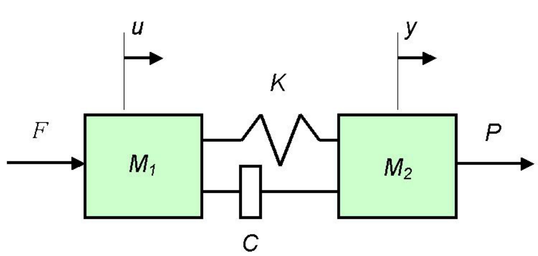

Fig. 1. Model with 2 degrees of freedom.

Setting up the equations for the balance of forces and converting to a state space formula the following equation of state is obtained:

$$\begin{Bmatrix}\dot{y}\\ \ddot{y}\end{Bmatrix} = \begin{bmatrix}0 & 1\\ -\omega_n^2 & -2\xi\omega_n\end{bmatrix}\begin{Bmatrix}y\\ \dot{y}\end{Bmatrix} + \begin{bmatrix}0 & 0 & 0\\ \omega_n^2 & 2\xi\omega_n & \dfrac{1}{M_2}\end{bmatrix}\begin{Bmatrix}u\\ \dot{u}\\ P\end{Bmatrix} \qquad (1)$$

where ω_n,and ξ are the natural frequency and damping ratio of the dominant mode of the transfer function $Y(j\omega)/U(j\omega)$.

Using this equation we can, provided that the initial conditions and the inputs are known, estimate the levels of the state variables at each moment. Thus, the position of the TCP is obtained directly while its acceleration is obtained via the following output equation:

$$\{\ddot{y}\} = \begin{bmatrix}-\omega_n^2 & -2\xi\cdot\omega_n\end{bmatrix}\begin{Bmatrix}y\\ \dot{y}\end{Bmatrix} + \begin{bmatrix}\omega_n^2 & 2\xi\cdot\omega_n & \dfrac{1}{M_2}\end{bmatrix}\begin{Bmatrix}u\\ \dot{u}\\ P\end{Bmatrix} \qquad (2)$$

The observer compares this estimated acceleration signal with the signal measured by the piezoelectric accelerometer on the TCP to make corrections in states via a Kalman gain (Fig. 2).

If this observer approach is to be used, the level of disturbance P must be known, but this is not generally possible. The influence of this factor on the approach must therefore be eliminated, and it must be hoped that correction via measured acceleration will offset the error made. However it has been found that there is a delay in the reaction of the observer in this case, and moreover the deformation due to constant forces cannot be estimated. We have therefore opted for an observer approach that seeks to correct this delay by taking the measured acceleration as an input (rather than entering it only as a correction) and takes the force of the motor as an additional input to correct for static deformation.

$$\begin{Bmatrix}\dot{u}\\ \dot{y}\\ \ddot{u}\\ \ddot{y}\end{Bmatrix} = \begin{bmatrix}0 & 0 & 1 & 0\\ 0 & 0 & 0 & 1\\ -\omega_n^2\mu & \omega_n^2\mu & -2\xi\omega_n\mu & 2\xi\omega_n\mu\\ 0 & 0 & 0 & 0\end{bmatrix}\begin{Bmatrix}u\\ y\\ \dot{u}\\ \dot{y}\end{Bmatrix} + \begin{bmatrix}0 & 0\\ 0 & 0\\ \dfrac{1}{M_1} & 0\\ 0 & 1\end{bmatrix}\begin{Bmatrix}F\\ \ddot{y}\end{Bmatrix}$$

$$\begin{Bmatrix}u\\ \ddot{y}\end{Bmatrix} = \begin{bmatrix}1 & 0 & 0 & 0\\ \omega_n^2 & -\omega_n^2 & 2\xi\omega_n & -2\xi\omega_n\end{bmatrix}\begin{Bmatrix}u\\ y\\ \dot{u}\\ \dot{y}\end{Bmatrix} \qquad (3)$$

where $\mu = M_2/M_1$.

In the output equation, the influence of the external disturbance P has been omitted, as it is an unknown variable. The error produced will be corrected by the feedback provided by the Kalman filter.

Again, the position of the TCP is obtained directly from the state, while its acceleration is estimated via the output equation. In this approach, the slide position u is a state variable estimated by integrating the state equation. The measurement from the scale provides a long term correction to the state variables and eliminates the effect of drift from the integration.

Prior to implementing this observer, the first step is to identify the system, i.e. to learn the modal characteristics ωn, ξ, and the masses of M_1, μ. This can be done in three ways:

--Perform a modal analysis of the system.

--Define the system as a parametric model and use an identification algorithm such as Recursive Least Squares (RLS) [17], [18].

--Use the state observer itself, increasing the vector of states with the parameters to be calculated and applying the Extended Kalman Filter (EKF) algorithm [17], [19].

$$\begin{Bmatrix} \dot{u} \\ \dot{y} \\ \ddot{u} \\ \ddot{y} \\ \dot{\omega}_n \\ \dot{\xi} \\ \dot{M}_1 \\ \dot{\mu} \end{Bmatrix} = A \cdot \begin{Bmatrix} \dot{u} \\ y \\ u \\ y \\ \omega_n \\ \xi \\ M_1 \\ \mu \end{Bmatrix} + B \cdot \begin{Bmatrix} F \\ \ddot{y} \\ \omega_n \\ \xi \end{Bmatrix} \quad (4)$$

$$\begin{Bmatrix} u \\ \ddot{y} \\ \omega_n \\ \xi \\ M_1 \\ \mu \end{Bmatrix} = C \cdot \begin{Bmatrix} \dot{u} \\ y \\ u \\ y \\ \omega_n \\ \xi \\ M_1 \\ \mu \end{Bmatrix} \quad (5)$$

where A, B, and C are the extended matrixes of the model described in (3).

If the modal characteristics of the system remain more or less constant while the machine is operating, all three of these methods are valid. If not, real time adjustment of parameters is required, which means that the second or third method must be used. The third is preferable for its compactness.

Once the model is identified, the next step is to develop the state observer described in (3). Denoting the state vector by $X(k)$, the input vector by $U(k)$, the output vector as $Y(k)$, and using G_{obs}, H_{obs}, C_{obs} to refer to the matrices obtained by discretising (3), the dynamics of the observer in state space formulation (Fig. 2), are defined as follows:

$$\begin{aligned} \hat{X}(k+1) &= G_{obs}\hat{X}(k) + H_{obs}U(k) + K_l(Y(k) - \hat{Y}(k)) \\ \hat{Y}(k) &= C_{obs}\hat{X}(k) \end{aligned} \quad (6)$$

where the superscript ^ indicates an estimation. Given that the model identified never achieves a 100% emulation of the behaviour of the system, a correction must be introduced into the states using a Kalman gain, K_l. The Kalman gain can be obtained via the usual techniques for locating poles in closed loops [20], or via Riccati's equation [18], [20].

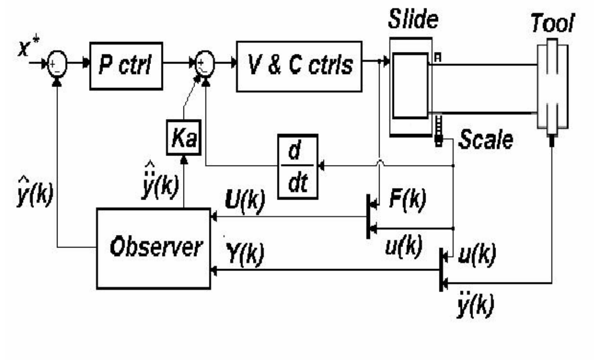

Fig. 2. Observer design layout.

The results obtained using the observer described on a test bed are shown below. The test bed comprises a linear motor on which a flexible beam is mounted with a large mass on the end to represent the dynamic behaviour of a flexible element. A scale is included on the slide and an encoder on the mass at the end of the beam to check the estimates made (Fig. 3).

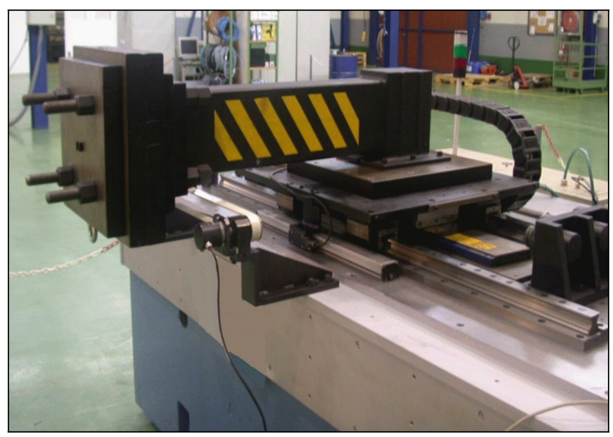

Fig. 3. Test bed.

To analyse the behaviour of the observer, the system was subjected to a position step and then to an impact on the TCP one second after the step. Figure 4 shows how most of the errors due to inertial forces and disturbances can be estimated.

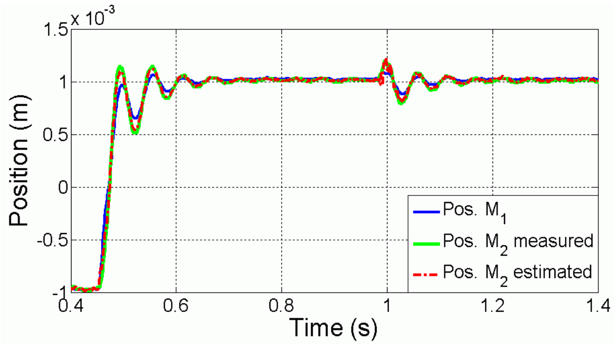

Fig. 4. Estimating the position of the TCP.

Finally, the sensitivity of the observer to errors in the identification of parameters of the 2 d.o.f. mechanical system was studied. Traces shown in Figure 5 are obtained by dividing the standard deviation of the estimation error for the correctly identified system with those obtained for incorrectly identified systems. It can be noticed that even with no precise identification of M_1 and M_2, the observer worked correctly, but it is much more sensitive to the correct identification of ω_n.

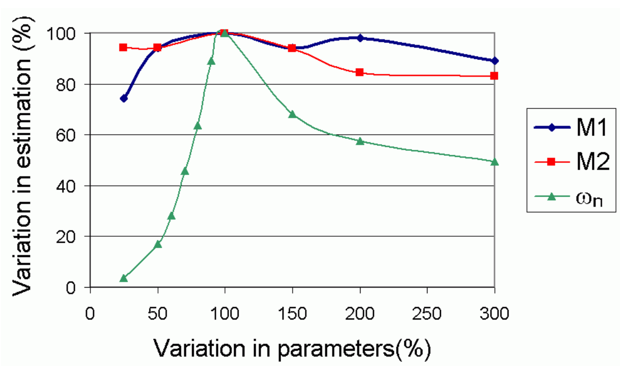

Fig. 5. Sensitivity of the observer to identification errors.

III. ACCELERATION FEEDBACK

The use of an acceleration loop in addition to the classical configuration of three control loops can mean considerable advantages in terms of system stiffness and damping. The acceleration signal fed back can be measured by an accelerometer or estimated by a state observer.

Fig. 6 shows the layout of a classical velocity loop for a two mass model and the layout with acceleration feedback. To enable the effect of acceleration feedback to be analysed, an equivalent system is also depicted with the structure of a classical velocity loop. The analysis takes into account that two configurations are possible depending on whether acceleration is fed back from mass M_1 or from mass M_2.

In Fig. 6, v*, a*, F* = velocity, acceleration and force set points; Kp, Ti = proportional and integral gain of the velocity loop; Mt = total system mass; Ka = gain of acceleration loop; a1, a2, v1, v2 = accelerations and velocities of masses M1, M2; 1/s = integrator. The superscript af means that optimum gains acquire new values when the acceleration loop is included.

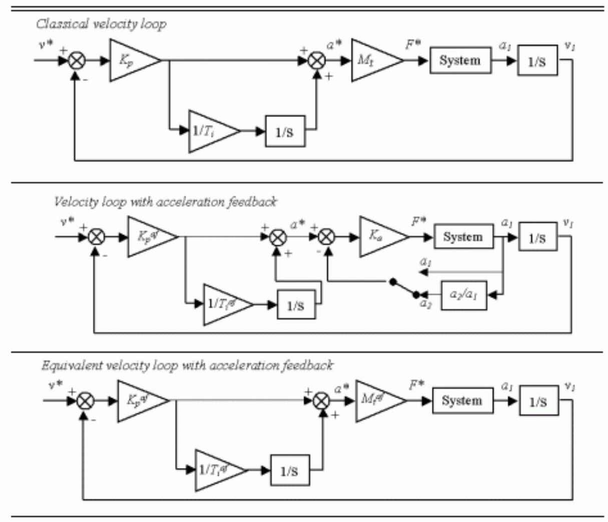

Fig. 6. Layouts of velocity loops with and without acceleration feedback

In flexible mechanical systems, controller adjustment limits are established by the mechanics of the system, and the effect of the current loop is therefore not taken into account. Expressions of optimum gain are defined seeking to obtain maximum damping in a closed velocity loop. A bandwidth of 80% of the drive resonance frequency is a recommendable level. However, in some cases with unfavourable M_2/M_1 mass ratios, this bandwidth can not be achieved.

Proper adjustment of the acceleration loop is equivalent to modifying the M_2/M_1 mass ratio in a flexible system. Thus, in systems with stability problems due to low motor inertia ($M_1 << M_2$), a stable behavior with sufficient phase and gain margins can be achieved by feeding back the acceleration of mass M_1. This way, the use of high inertia motors –specially designed for overcoming this stability problems [21]- can be avoided, allowing economic and energetic savings. On the other hand, in systems with damping problems due to low load inertia ($M_2 << M_1$), feeding back the acceleration of mass M_2 improves damping in the system. Moreover, in both cases, acceleration feedback increases the stiffness of control loops in the face of external disturbances.

The higher the gain in the acceleration loop, K_a, the greater the benefit obtained. In practice, the level of K_a attainable is limited by the noise in the acceleration signal. The design of an acceleration observer that can estimate a noiseless signal is therefore highly attractive.

The table below shows the results for the studied model depending on whether feedback is from the first or second mass.

	Feedback from M_1	Feedback from M_2
Mass ratio M_2/M_1	$\frac{M_2}{M_1^{af}} = \frac{M_2}{K_a + M_1}$	$\frac{M_2}{M_1^{af}} = \frac{K_a + M_2}{M_1}$
Damping attainable in closed velocity loop	Increase for high M_2/M_1 ratios	Increase for low M_2/M_1 ratios
Proportional gain in velocity loop	$K_p^{af}(rad/s) = K_p\left(1 + \frac{M_1 + M_2}{K_a}\right)$	
Integral gain in velocity loop	No influence	
Proportional gain in position loop	Increase for low M_2/M_1 ratios	
System stiffness	$\Delta_{stiffness} = 1 + \frac{K_a}{M_1 + M_2}$	
System speed	No increase in speed (compared to set points)	

Table 1. Influence of the acceleration feedback

It can be noticed that using acceleration feedback it is possible to adjust the dynamic performance of the system, its damping, and to improve the stiffness of control loops against external disturbances. In the test bed mentioned above, acceleration feedback from the TCP (M_2) has been implemented, and the results are shown in the next section.

A. Estimating the acceleration of the TCP

There is a classical method to estimate the acceleration of the TCP by using only the position signals of M_1 and M_2. This method is described by the next equation:

$$\begin{Bmatrix} \dot{y}(t) \\ \ddot{y}(t) \end{Bmatrix} = \begin{bmatrix} 0 & 1 \\ -\omega_n^2 & -2\xi\omega_n \end{bmatrix} \begin{Bmatrix} y(t) \\ \dot{y}(t) \end{Bmatrix} + \begin{bmatrix} 0 & 0 \\ \omega_n^2 & 2\xi\omega_n \end{bmatrix} \begin{Bmatrix} u(t) \\ \dot{u}(t) \end{Bmatrix}$$

$$\dot{y}(t) = \begin{bmatrix} 0 & 1 \end{bmatrix} \begin{Bmatrix} y(t) \\ \dot{y}(t) \end{Bmatrix} \tag{7}$$

However, on the one hand, this model is not able to estimate the acceleration correctly in presence of external perturbations, and on the other hand it requires the measurement of the TCP position. In order to avoid these limitations, the observer described in the previous section (2), (3) was used to estimate the acceleration of the TCP.

The estimated acceleration was fed back into the velocity loop on the test bench described above, and very good results were obtained. Since the estimated signal is practically noise free, higher gains can be used in the acceleration loop, and systems with better damping and greater stiffness can therefore be obtained. Damping in the closed velocity loop was increased by 40%, while stiffness was increased almost three-fold. The figures below show the estimated acceleration signal and the obtained improvement in system damping:

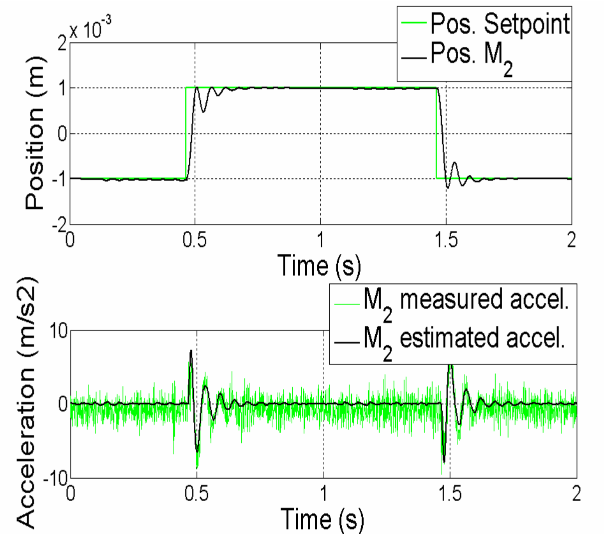

Fig. 7. Response with no feedback of acceleration.

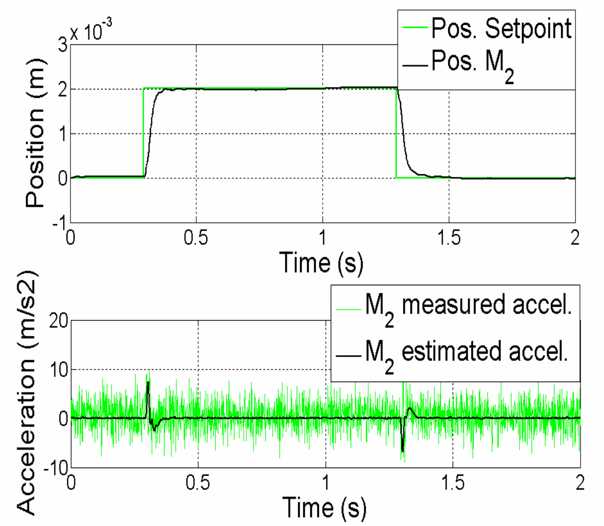

Fig. 8. Response with feedback of estimated acceleration.

IV. CONCLUSION

The state space observer proposed here improves the precision and dynamic behaviour of machine-tools. By feeding back the position of the point of interest, TCP, it is possible to include deformation in machine elements beyond the scale in the drive control loops. On a test bed, most of the errors arising from inertial deformation and disturbances were reduced.

Moreover, the estimated TCP acceleration signal is practically noiseless, so by feeding it back higher gains can be used in the acceleration loop. This in turn enables better damped, more rigid systems to be attained. On the test bed, damping in a closed velocity loop was increased by around 40% and stiffness was increased almost three-fold.

ACKNOWLEDGEMENTS

This study was made possible by financial support from the Spanish Ministry of Science and Technology (MCYT) and the CDTI (Centre for Industrial technology Development) as part of the project “Metodología para el ajuste de las funciones más importantes de los CNCs de última generación en máquinas-herramienta y análisis de futuras estrategias de control” [“Methods for adjusting the major functions of latest generation CNCs in machine-tools and analysis of future control strategies"], ref. FIT-020200-2003-26.

We are also grateful for support from Sociedad para la Promoción y Reconversión Industrial (SPRI) as part of the project “Diseño de una Metodología para el ajuste de CNCs en máquinas-herramienta y mejora de prestaciones” [“Design of a method for CNC adjustment on machine-tools and improvement of performance”], ref. CN02FU02.

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