Model of servo drive system to increase performance of machine tools

Ioan Ghimbaseanu

Abstract: In the application of mechatronic positioning systems (also known as CNC axes) it is desirable to predict the servo performance. By using computer simulation techniques it is possible to construct a very accurate model of the servo drive. It is possible to observe the effect on performance by changing the drive parameters such as load inertia, backlash, stick-slip, viscous friction, load thrust, servo loop gains, velocity and acceleration. This paper deals with a model of a numerically controlled mechatronic system which consists of a motion control system driven by a D.C. motor. Both position and velocity feedback loops are present in the structure of the system. By means of MATLAB & Simulink software, simulation diagrams were built in order to test the behavior of the system in positioning regime. An analytic tuning of the position controller was performed prior to the simulation. Two velocity profile were used for testing the position behavior of the system: the trapezoidal profile and the parabolic one. The results in the case of the parabolic velocity profile were satisfactory: even if the positioning error has quite a high value it was constant and easy to compensate by introducing it as an offset value in the memory of the NC equipment. The analytic tuning, however, fails in the case of the trapezoidal profile. The system oscillates a lot and the positioning error was inconstant and subsequently cannot be compensated. In order to compensate the behavior of the system two solutions were adopted: re-tuning of the position controller using the “continuous emulation tuning method” and using a 16 bit DAC instead of a 14 bit one. The compensated system performed well during the simulation in both cases, the trapezoidal profile being usable in this case with satisfactory results.

Keywords: CNC axes, compensation, mechatronic system, positioning regime, velocity profile

I. INTRODUCTION

Servo-driven contouring machine tools are capable of producing high quality precise parts. However, the accuracy of such machines is often limited by tracking (positioning) and contouring errors.

Tracking (positioning) and contouring errors result from various kinds of sources, including mechanical hardware deficiencies, cutting process effects and drive dynamics.

A detailed discussion of this performance limiting factors can be found in [1] and [4], where they are categorized as dynamic constraints, uncertainties and non-linearity.

In the design and synthesis of machine tool feed drives, it is desirable to predict the effects of the non-liniarities in the machine system on servo performance [5].

Ioan Ghimbaseanu is with the Transylvania University of Brasov, Romania, Faculty of Materials Science and Engineering as Associate Professor (corresponding author to provide e-mail: ghimbasani@unitbv.ro)
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It is possible to use computer simulation techniques in analyzing and synthesizing servo drives. Thus, non-linearity of the system can be treated much more realistically.

It is the purpose of this paper to show how simulation can be used to accurately represent and predict performance of a machine tool feed servo drive (mechatronic system), specifically performance in the positioning regime.

II. THE NC FEED DRIVE MODEL

A simplified block diagram for a NC feed drive is shown in Fig. 1, where: K_g – gear ratio [m/rad]; K_e – encoder gain [pulses/rad], K_p – position gain; K_c – DAC gain [V/bit]; x_r – reference position [μm]; U_c- voltage command [V]; ω_M – motor velocity [rad/s], θ_M – motor angular position [rad]; x – actual position [μm].

Fig. 2 shows a detailed diagram of the behaviour of the mechanical system.

The diagram includes one predominant mechanical spring mass system, which has a known resonance, a load inertia, viscous damping, backlash, and stick-slip, where: K_t – motor torque constant [Nm/A]; J_m – motor inertia [kgm^2]; B_m – motor viscous friction [Nms/rad]; K_x – mechanical stiffness [Nm/rad]; J_l – load inertia [kgm^2]; B_l – load viscous friction [Nms/rad]; F_a – cutting force [N], m – mass of the load [kg]; $\omega_{m,\,l}$ – motor/load velocity [rad/s], $\theta_{m,\,l}$ – motor/load angular position [rad];

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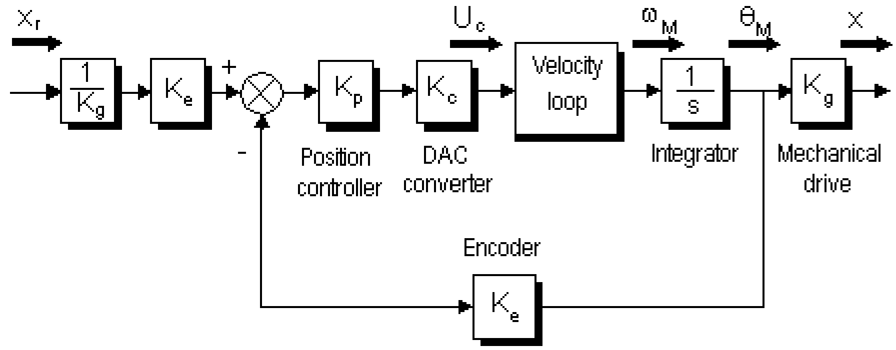

Fig.1 a simplified block diagram for a NC feed drive.

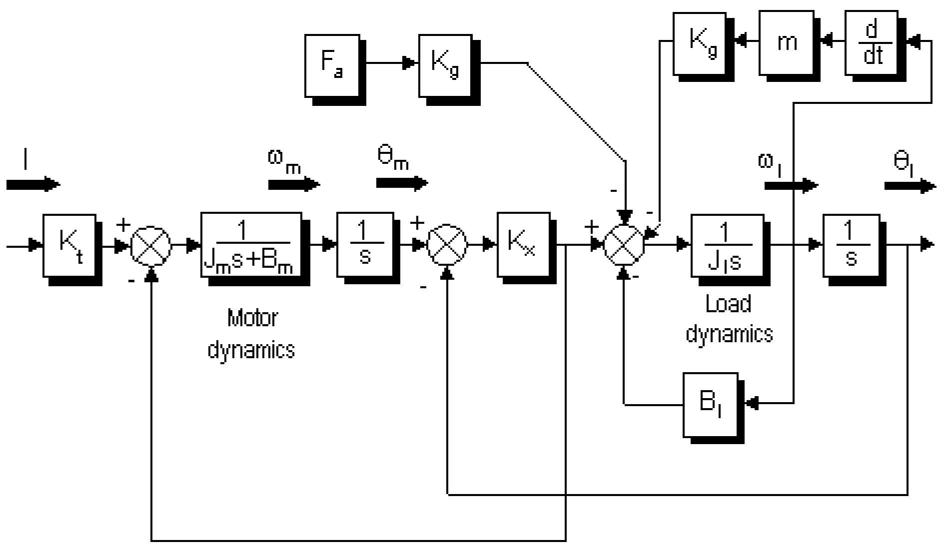

Fig.2 A detailed diagram of the behavior of the mechanical system.

III. SIMULATING THE MECHATRONIC SYSTEM

In order to run the simulation, the MATLAB & Simulink software package was used. Prior to the simulation, two MATLAB function had to be written, in order to include the effects of backlash and stick-slip in the system.

For a machine tool slide that is not moving, the friction is static. As soon as the machine tool slide breaks away (starts to move), the friction drops to a lower Coulomb friction. It was assumed that the Coulomb friction was about 20% of the rated motor torque, and the static friction was some factor larger than the Coulomb friction. The ratio of static friction to Coulomb friction was an input variable. More information about the models used for these non-linearity may be found in [1].

By means of an analytic process with imposed and determined values [1], the mechatronic system analyzed can be described by the characteristics presented in Table 1.

Table 1. Characteristics of the mechatronic system analysed.

Parameter	Value
Mechanical transmission gain K_g [m/rad]	0.0013
Incremental encoder gain K_e [imp/rad]	4096/2
Position controller gain K_e	23.0477
Digital to analog converter gain K_c [V/bit]	0.00097
Power amplifier gain K_a	3.00
Rezisten a la borne R [Ω]	0.27
Motor torque constant K_t [Nm/A]	0.235
Motor velocity constant K_v [Vs/rad]	0.2291
Tachometer gain K_{th} [Vs/rad]	0.0954
Mechanical stiffness Kx [Nm/rad]	10.8
Backlash [mm]	0.005
Motor inertia J_m [kgm^2]	$28 \cdot 10^{-4}$
Motor viscous friction B_m [Nms/rad]	0.0082
Load inertia J_l [kgm^2]	$54.1 \cdot 10^{-4}$
Load viscous friction B_l [Nms/rad]	0.0082
Motor Coulomb friction M_{fcm} [Nm]	0.25
Motor staic friction M_{fsm} [Nm]	0.275
Load Coulomb friction M_{fcl} [Nm]	1.4
Load static friction M_{fsl} [Nm]	1.54
Limit velocity (static friction changes to Coulomb friction) v_e [rad/s]	0.001
Sampling period T_s [s]	0.01

The positioning regime is mainly determined by the velocity profile used. The most used profiles are the trapezoidal and the parabolic one. In Figures 3, 4 and 5 simulated space, velocity and acceleration are presented for both parabolic and trapezoidal profiles for a distance of 520 mm.

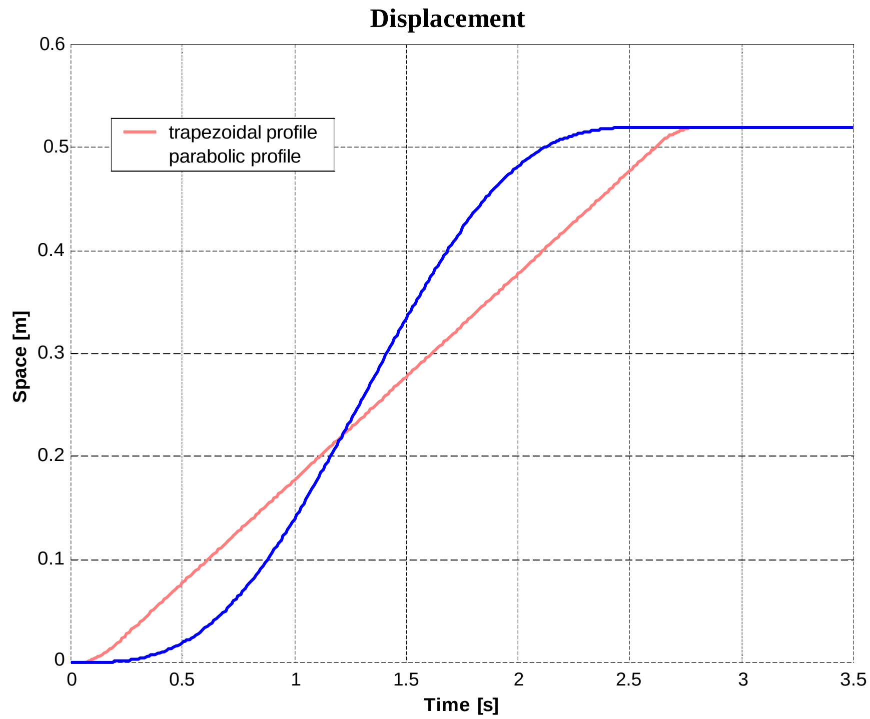

Fig.3 Diagram of simulated space.

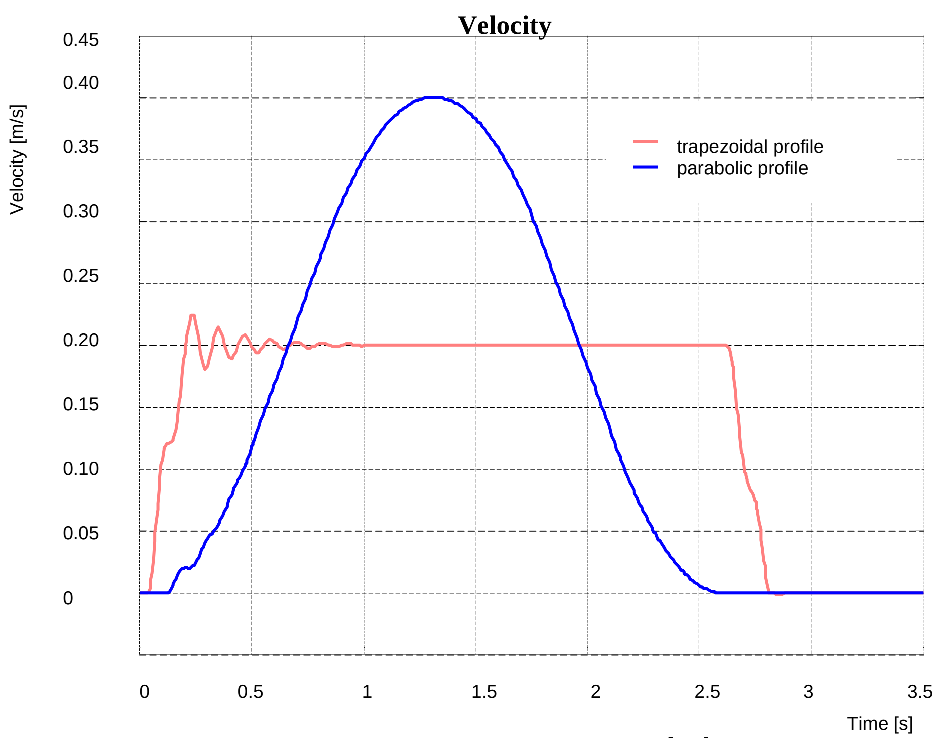

Fig.4 Diagram of velocity.

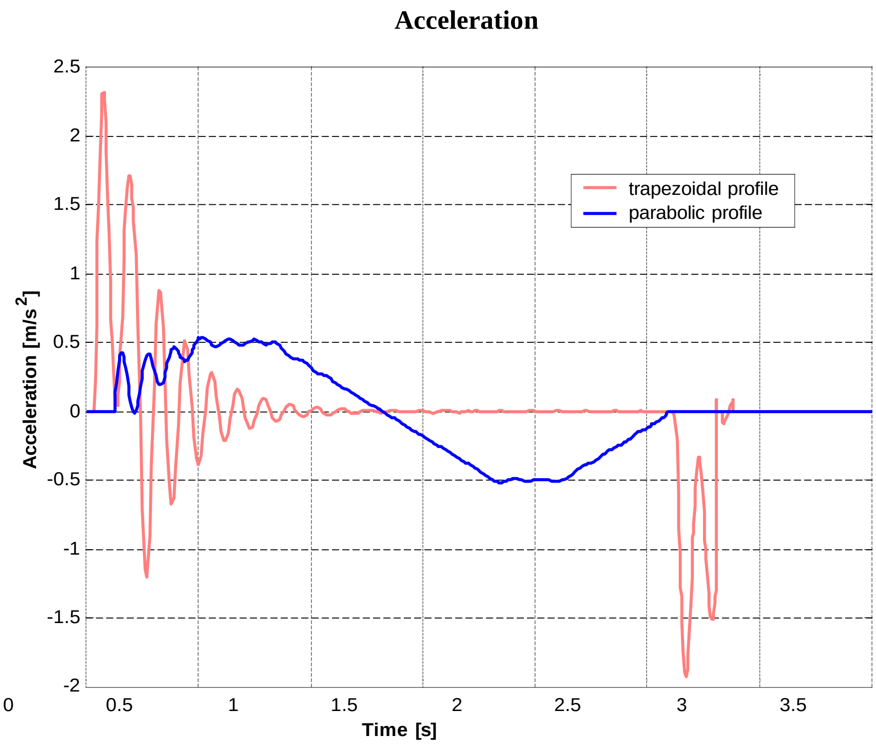

Fig.5 Diagram of acceleration.

It can be observed that the trapezoidal profile introduced oscillations in the system. The positioning error has quite a high value (518μm) for the parabolic velocity profile, but it is extremely constant for a large domain of velocities and/or accelerations. In the case of trapezoidal profile, the positioning error shows a highly inconstant value. The behavior of the positioning error lead us to the following conclusion: even if the error is high for the parabolic profile it's constant value make it easy to compensate by introducing it in the NC equipment's memory as offset value. On the other hand, the trapezoidal velocity profile cannot be used in this case. However, the generation of the behavior of the positioning error lead us to the following conclusion: even if the error is high for the parabolic profile it's constant value make it easy to compensate by introducing it in the NC equipment's memory as offset value.

On the other hand, the trapezoidal velocity profile cannot be used in this case. However, the generation of the parabolic velocity profile may be cumbersome and it consumes a large amount of computing power from the NC system. It can be helpful, in many cases to find solutions in order use the trapezoidal profile. The tuning of the system controller (the position controller) was done analytical. We have to test the system in order to study the quality of this tuning process. In Figures 6 and 7 the response of the system to ramp, step and pulse inputs are presented.

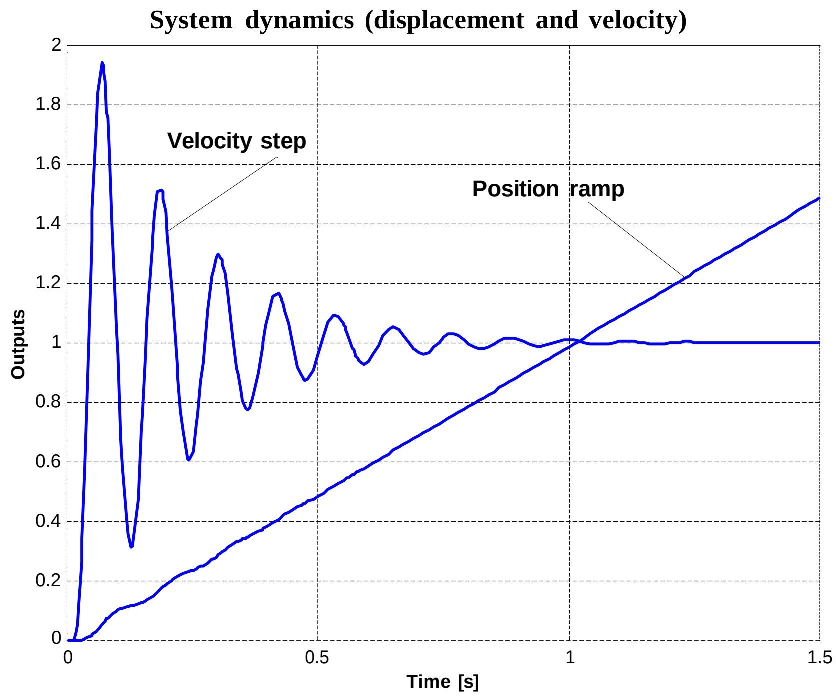

Fig.6 Diagram of system dynamics (displacement and velocity)

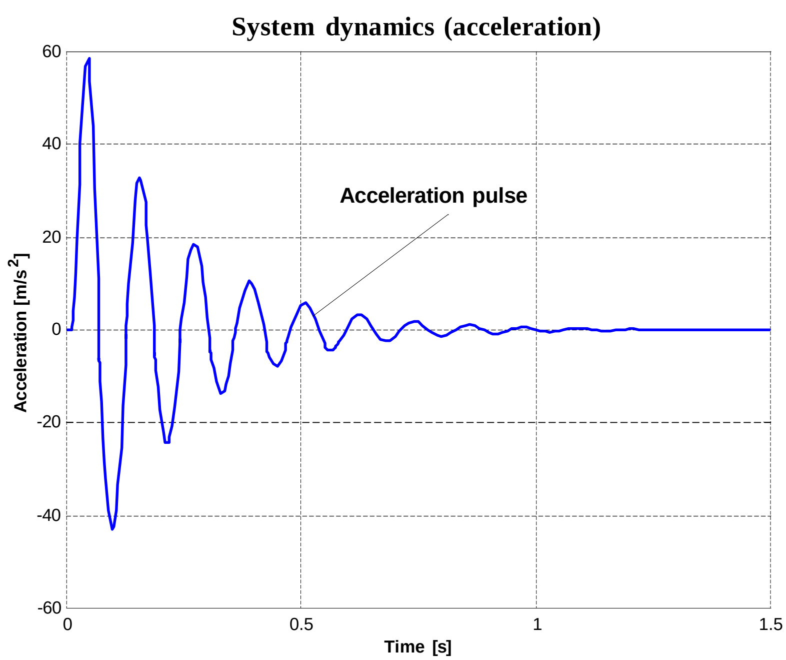

Fig.7 Diagram of system dynamics (acceleration).

From figures 6 and 7, in order to compensate the behavior of the system, two measures were performed: re-tuning of the position controller using the “continuous emulation tuning method” and using a 16 bit DAC instead of a 14 bit.

The dynamic of the compensated system is presented in Figures 8 and 9.

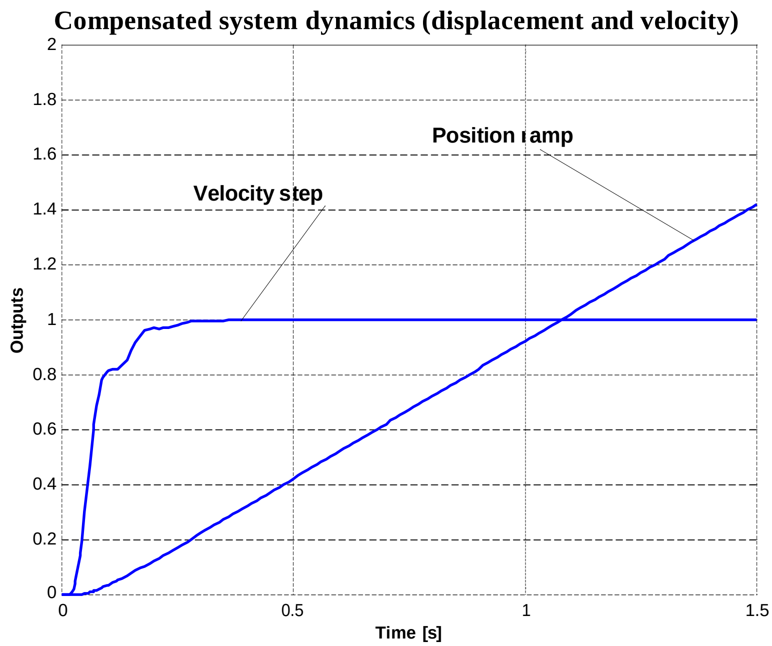

Fig.8 Dynamic response of compensated system- displacement and velocity.

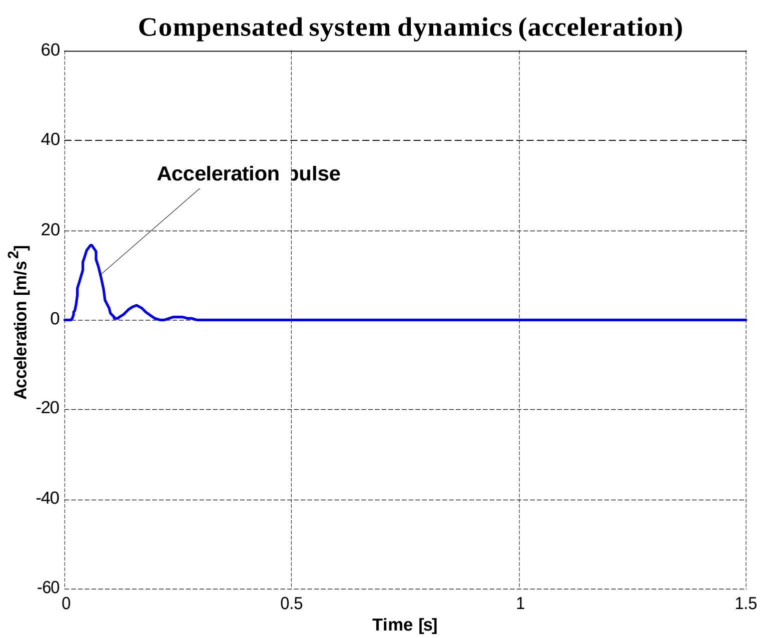

Fig.9 Dynamic response of the compensated system – acceleration.

The behavior of the system seems now to be satisfactory: no overshoot is present (velocity step response) and the oscillations are quite low (acceleration pulse response). The rising and settling time are also low, depicting a fast response of the system.

After the compensation, another set of simulations of the system in positioning regime, using the trapezoidal velocity profile were performed. The results are presented in Figures 10, 11, 12.

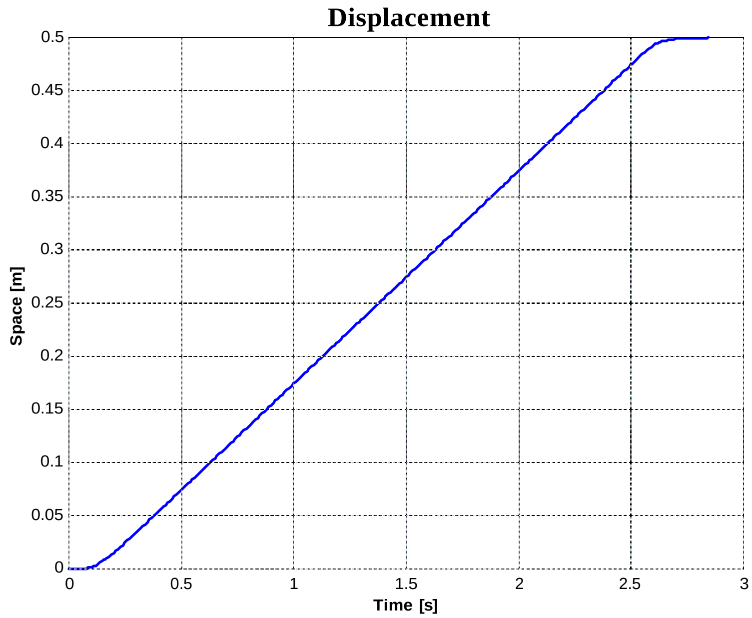

Fig.10 The result of simulation in positioning regime – displacement.

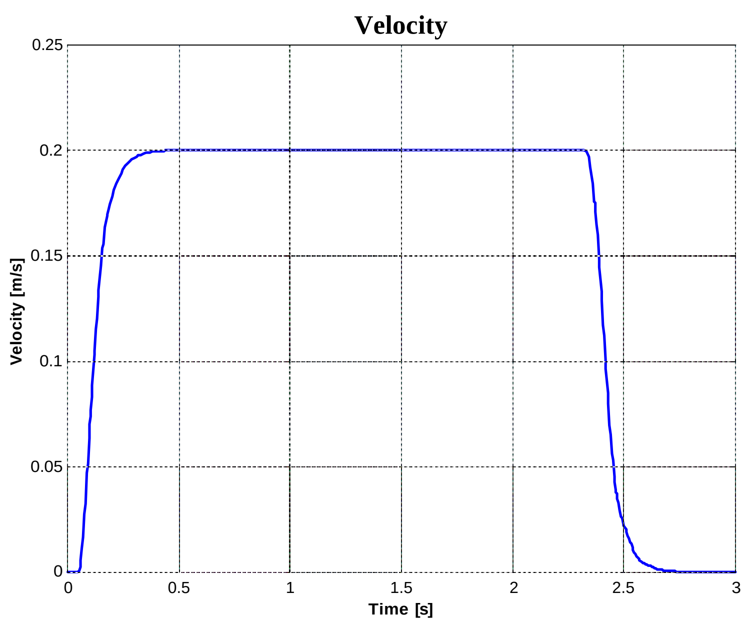

Fig.11 The result of simulation in positioning regime –velocity.

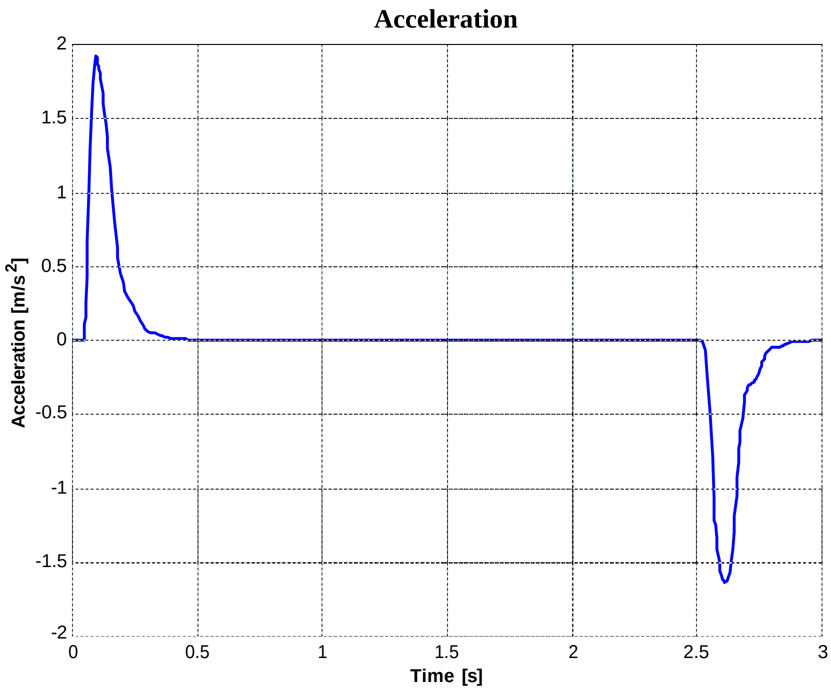

Fig.12 The result of simulation in positioning regime – acceleration.

The positioning error is now quite constant and equal with the one for the parabolic profile (518µm) and it may also be compensated by introducing it in the NC equipment's memory as offset value. In order to test the constancy of the positioning error, a set of computer simulation for a large range of cinematic parameters were performed. Some results regarding the behaviour of the system are presented in Table 2.

Table 2. Results regarding the behavior of the system.

Velocity v [m/s]	Acceleration a [m/s^2]	Acceleration time T_a [s]	Constant velocity time T_{ct} [s]	Positioning cycle time T_c [s]	Positioning error [µm]
0.2	0.5	0.4	2.4	3.2	518
0.2	0.5	0.4	1	1.8	518
0.2	0.5	0.4	0	0.8	517
0.2	1	0.2	2.4	2.8	519
0.2	1	0.2	1	1.4	517
0.2	1	0.2	0	0.4	516
0.2	1.5	0.133	2.4	2.666	519
0.2	1.5	0.133	1	1.266	517
0.2	1.5	0.133	0	0.266	516
0.2	2	0.1	2.4	2.6	519
0.2	2	0.1	1	1.2	517

0.2	2	0.1	0	0.2	516
0.1	0.5	0.2	2.4	2.8	519
0.1	0.5	0.2	0	0.4	516
0.1	1	0.1	2.4	2.6	519
0.1	1	0.1	0	0.2	516
0.1	1.5	0.066	2.4	2.532	519
0.1	1.5	0.066	0	0.132	516
0.1	2	0.05	2.4	2.5	519
0.1	2	0.05	0	0.1	516

IV. CONCLUSIONS

This paper deals with a model of a numerically controlled mechatronic system which consists of a motion control system driven by a D.C. motor. Both position and velocity feedback loops are present in the structure of the system. By means of MATLAB & Simulink software, simulation diagrams were built in order to test the behaviour of the system in positioning regime.

An analytic tuning of the position controller was performed prior to the simulation. Two velocity profile were used for testing the position behaviour of the system: the trapezoidal profile and the parabolic one.

The results in the case of the parabolic velocity profile were satisfactory: even if the positioning error has quite a high value it was constant and easy to compensate by introducing it as an offset value in the memory of the NC equipment. The analytic tuning, however, fails in the case of the trapezoidal profile. The system oscillates a lot and the positioning error was inconstant and subsequently cannot be compensated.

An explanation may be that the analytic tuning has not taken into consideration the hybrid character of the system: both analogue (inner velocity loop) and digital signals (outer positioning loop) are present in the system. In order to compensate the behaviour of the system two solutions were adopted: re-tuning of the position controller using the “continuous emulation tuning method” and using a 16 bit DAC instead of a 14 bit one.

The compensated system performed well during the simulation in both cases, the trapezoidal profile being usable in this case with satisfactory results.

Each of the two profiles used presents some characteristics: the trapezoidal velocity profile is easier to be generated and the minimum programmable positioning distances are lower as in the case of the parabolic velocity profile. However, the parabolic profile is more robust: even if the overall gain of the system changes (due to long use) the behaviour of the system remains satisfactory, only the offset value having to be changed (which can be easy determined experimentally).

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