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Hybrid control with on/off electropneumatic standard valve for tracking positioning.

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ABSTRACT

This paper presents a new method of control applied in electropneumatic field. This strategy is issued from hybrid control theory recently applied in control of asynchronous or synchronous electrical motor (1, 2). The interest of these procedure concerns the possibility of controls the position of an electropneumatic piston all along the cylinder stroke with standard on/off valve. Nowadays the industrial electropneumatic process used on/off valve for point to point aim with displacement from one extremity of cylinder to the other one. When different desired positions are required the constructors used specific components issued from proportional technology: servovalve or servodistributor for example. The evolution in the automation process is moving towards a need of obtaining greater versatility and increased precision in compressed air driven equipment. This means obtaining proportional operation of the power element as a function of an electric control signal. Nevertheless, when the desired precision is near the millimetre and not very good performances are need during dynamic stage, the useful of proportional technology can be debatable. Indeed the system cost and its complexity to tune can be two drawbacks that on/off technology with the proposed algorithm, can be concurrence.

Based on both the models of cylinder and valves, the hybrid control presented here determines the best state of valves by tracking reference values of the cylinder states in the state space. Then a simplified model of electropneumatic system is presented and used to synthesised hybrid control algorithm. Experimental results are presented and discussed.

Keywords: Hybrid Control, standard valve, electropneumatic, experimental results.

NOMENCLATURE

b viscous coefficient (N/m/s)

F force (N)

k polytropic constant

M moving load (kg)

p pressure (Pa)

q_m mass flow rate (kg/s)

r perfect gas constant (J/kg/K)

S area of the piston cylinder (m²)

t time (s)

T temperature (K)

U valve input voltage (V)

v velocity (m/s)

V volume (m³)

x state vector

y position (m)

ΔT sample time

Subscripts and superscripts

d desired

e equilibrium

ext external

E exhaust

N chamber N

P chamber P

PNEU pneumatic

S supply

t0 time value at each sample time

1 INTRODUCTION

At the present time, the majority of the pneumatic cylinders controlled in position use servodistributors to control the mass flow rates delivered into the cylinder chambers in spite of the fact that distributors are less expensive than standard valve. The first control laws applied in Fluid Power concerned classical state feedback and were proposed by Shearer et al (3) and Burrows (4). The manufacturers of the first electropneumatic positioning systems such as Martonair (5) and GAS initially adopted it. Since then new sophisticated algorithms have been applied to electropneumatic systems: adaptive (6, 7), sliding mode control (8, 9, 10, 11), H_∞ (12), fuzzy control (13), neural control (14), flatness (15), backstepping (16)... All of them lead to more or less good results in position tracking but none of them is applicable to systems with on/off electropneumatic valves. The aim of this paper is to offer a cheaper alternative to classic pneumatic systems controlled in position which need a static precision around millimetre but for whose the position tracking error can be high in dynamic. The control must also lead to weak overshoot (around five percent of the total increase) and have few oscillations.

Using on/off electropneumatic valves for tracking positioning leads to a system with a discrete control and a continuous process: the on/off valves can take two or three discrete states even though the state variables of the cylinder are continuous. Such systems, which combine a discrete control and a continuous process, are called in this paper 'hybrid systems' and take many shapes.

Hybrid control is an efficient approach to control this kind of systems. For example, in electrical engineering, hybrid control has been developed to control electrical synchronous and asynchronous machines (1,2). In this case, synchronous and asynchronous machines modelled by continuous state equations are controlled by inverters, which can take eight discrete states. This approach can be expanded to a lot of hybrid systems.

The idea of the hybrid control is to choose at each sample time the best state of the discrete control to follow several variables of interest. Using a control model of the system, the algorithm must calculate, for each possible state of the control, the values of the variables of interest at the next sample time. Then the control state that leads to follow quickly the variables of interest will be chosen. This control state will be applied during one sample time.

In the case of an on/off electropneumatic standard valve for tracking positioning, the variables of interest are p_N and p_P . Indeed the control of the two pressures means the control of pneumatic force. Once the pneumatic force is controlled, a speed feedback and a position feedback will permit to obtain good performances in term of position tracking.

The study began with the system modelling: a control model will be defined from the knowledge model. The approximations used to define this model concern the sample time and the variables of interest chosen. Notice that the problem is very different from the electrical machine control due to a great difference of the time constants, which is equal to few microseconds in electrical field and equal to few milliseconds in pneumatic area. In next section the control algorithm will be developed. Finally, two series of experimental tests will be presented and analysed.

2 FROM KNOWLEDGE MODEL TO CONTROL MODEL

2.1 Knowledge model

The studied process is an in-line electropneumatic servodrive using a simple rod double acting linear pneumatic cylinder (Fig. 1). Two 3/3 standard valves delivered the mass flow rates into the cylinder

chambers. The rod of the actuator is connected to one side of a carriage. The aim is to control its position.

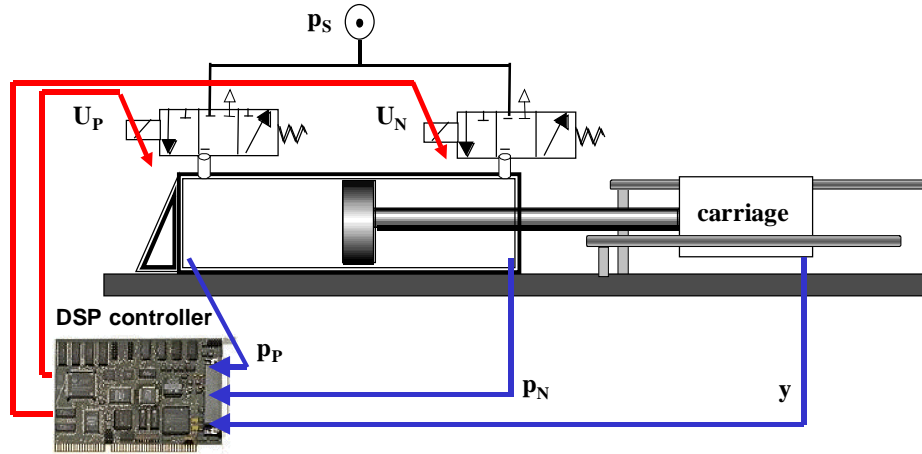


Figure 1: Electropneumatic system

Following classical assumptions (3), the model can be described by equation (1), in which:

- the dynamics of the valves are neglected,
- the evolutions in each cylinder chamber are supposed to be polytropic,
- the temperature variation in each chamber is neglected,
- the dry frictions are neglected,

$$\left. \begin{aligned} \frac{dp_P}{dt} &= \frac{krT}{V_P(y)} \left[q_{mP}(U_P, p_P) - \frac{S_P}{rT} p_P v \right] \\ \frac{dp_N}{dt} &= \frac{krT}{V_N(y)} \left[q_{mN}(U_N, p_N) + \frac{S_N}{rT} p_N v \right] \end{aligned} \right\} \text{pneumatic part} \quad (1)$$

$$\left. \begin{aligned} \frac{dv}{dt} &= \frac{1}{M} [S_P p_P - S_N p_N - f_v v - F_{ext}] \\ \frac{dy}{dt} &= v \end{aligned} \right\} \text{mechanical part}$$

The valve input can take three discrete values: -10V (exhaust pressure); 0V (closed); 10V (input pressure). For energetic reasons, the two input voltages are never set to +10V simultaneously. So eight combinations of control values will be considered:

control index	c1	c2	c3	c4	c5	c6	c7	c8
U_P	+10V	-10V	-10V	+10V	0V	-10V	0V	0V
U_N	-10V	+10V	-10V	0V	+10V	0V	-10V	0V

Table 1: Eight discrete possible states of control

To obtain position equilibrium, the valves need to be closed: $U_P^e = U_N^e = 0$. Indeed if a chamber is connected to the supply or exhaust pressure, the pneumatic force F_{PNEU} defined by equation (2) leads to a displacement to an extremity of the cylinder stroke.

$$F_{PNEU} = S_P p_P - S_N p_N \quad (2)$$

Considering these remarks and relation (1), the set equilibrium imposed relation (3) and so (p_P^e, p_N^e) is imposed by valves but not by cylinder. This result is right exclusively if the pneumatic force is equal to external force (see relation (1) with null equilibrium velocity). This condition is less restrictive if dry friction force is considered, and in a lot of experimental case, this condition is respected.

$$q_{mP}(U_P^e, p_P^e) = q_{mN}(U_N^e, p_N^e) = 0 \quad (3)$$

2.2 Model simplification

Dynamic of mechanical variables (position and velocity) is slower a lot than dynamic of pneumatic variables (the two pressures). Then, in pneumatic equations, position and velocity can be considered constant during a sample time ΔT chosen small enough compared to mechanical time constant:

$$\left\{ \begin{array}{l} \frac{dp_P}{dt} = \frac{krT}{V_P(y_{t0})} \left[q_{mP}(U_P, p_P) - \frac{S_P}{rT} p_P v_{t0} \right] \\ \frac{dp_N}{dt} = \frac{krT}{V_N(y_{t0})} \left[q_{mN}(U_N, p_N) + \frac{S_N}{rT} p_N v_{t0} \right] \end{array} \right\} \quad \text{Pneumatic part} \quad (4)$$

Simulations show that the evolution of the nonlinear system (equation (1)) in the plane (p_P, p_N) is linear all over the physical domain ($1 \text{ bar} < P < 7 \text{ bar}$) excepted when the piston is initially near an extremity. In the following results, the initial position of the piston is considered near of the central position.

The linearity of the evolution of the system in the plane (p_P, p_N) involves that for each initial position in (p_P, p_N) , it is possible to calculate the eight reachable points in (p_P, p_N) corresponding to the eight control values. Indeed considering the evolution of the pressure in a chamber during ΔT as linear and knowing the state of the system at time t_0 , the pressures values can be determined at time $t_0 + \Delta T$ as given by equation (5) where the continuous model is transform in a discrete one.

$$\left\{ \begin{array}{l} p_P(k+1) = \left[\frac{d}{dt}(p_P(t)) \right]_{t=t_0} \times \Delta T + p_P(k) \\ p_N(k+1) = \left[\frac{d}{dt}(p_N(t)) \right]_{t=t_0} \times \Delta T + p_N(k) \end{array} \right. \quad (5)$$

In these expressions $\left[\frac{d}{dt}(p_P(t)) \right]_{t=t_0}$ is a function of p_P, y, v and U_P , and $\left[\frac{d}{dt}(p_N(t)) \right]_{t=t_0}$ is a function of p_N, y, v and U_N . At each sample step p_N, p_P, y are measured, v is estimate by numerical derivation so for each one of eight couples $(U_P, U_N), (p_P(k+1), p_N(k+1))$ can be calculated.

The choice of the sample time ΔT has been made by many simulations using the nonlinear model (mentioned in equation (1)). For each one of eight control values and for different simple times and different initial condition values of the system, the results show that:

- for a simple time greater than 100 milliseconds, pneumatic variables saturate before the end of the simulation.
- for a simple time lower than 10 milliseconds, the nonlinear model is not true because the dynamic of valve is not negligible anymore.

So the sample time has been chosen equal to $\Delta T=10$ ms.

Figure 2 presents the evolution of the nonlinear system during ten milliseconds for the eight control values in (p_P, p_N) plane and the eight points determined with the discrete linear model. These figure has been studied for different initial positions in (p_P, p_N) plane all over the physical domain. The conclusion is that the discrete time linear model is very good in direction and good in amplitude.

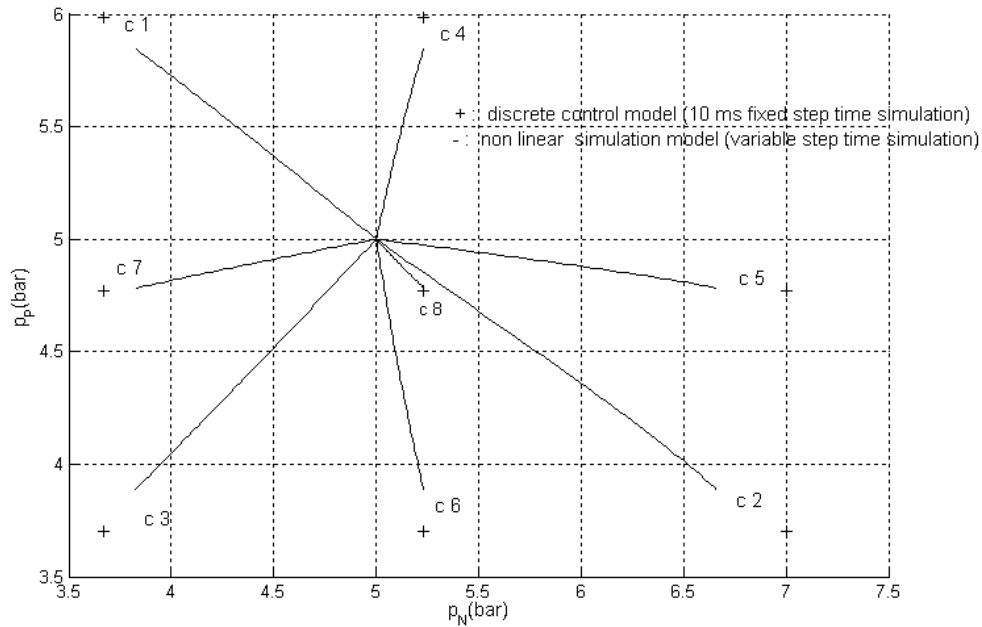


Figure 2 : Validation of linear discrete time model used for control synthesis

3 CONTROL SYNTHESIS

3.1 Hybrid force control

As underlined previously, it is possible to control position if the pneumatic force tracking is correct, adding for example a speed control feedback and a position control feedback. The mechanical equations show that pneumatic force can be following by controlling pressures p_P and p_N . So a desired force generated by the position loop can be translated in a target point in (p_P, p_N) . The hybrid control algorithm consists of choosing the best state for each valve at each sample time in order to reach a target point in (p_P, p_N) .

The target will be fixed at each sample time in view of these two constraints :

- it must corresponds to the desired pneumatic force, so to respect equation (2) it must belong to the line of equation (6) with constant pneumatic force:

$$p_P = \frac{S_N}{S_P} p_N + \frac{F_{PNEU}}{S_P} \quad (6)$$

- it must be as near as possible of the partial pneumatic equilibrium set (p_P^e, p_N^e) in order to reach the equilibrium set in static stage. This constraint is not necessary in dynamic stage and could be harmful to dynamic performances, but as explain in introduction the most important aim concerns static performances.

The target in (p_P, p_N) plane is calculated each sample time to be the nearest point of (p_P^e, p_N^e) belonging on the line of equation (6) (see bold line in figures 4 and 5) and inside the physical domain (see rectangle in figures 3 and 4).

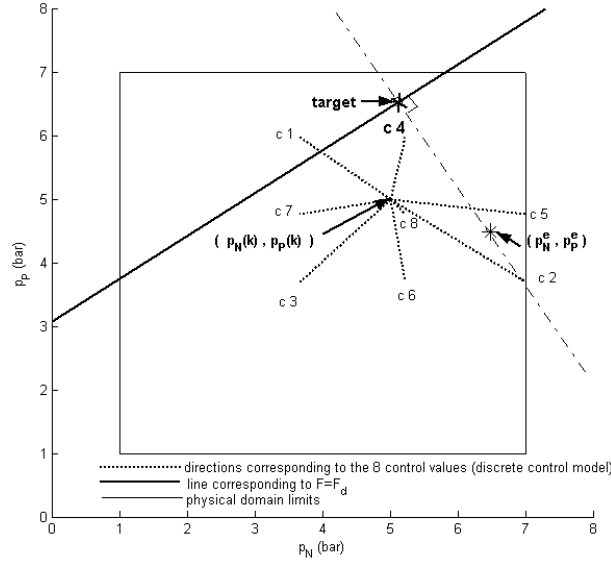


Figure 3 :target in (p_P, p_N) plane when $F_d = 100$ N

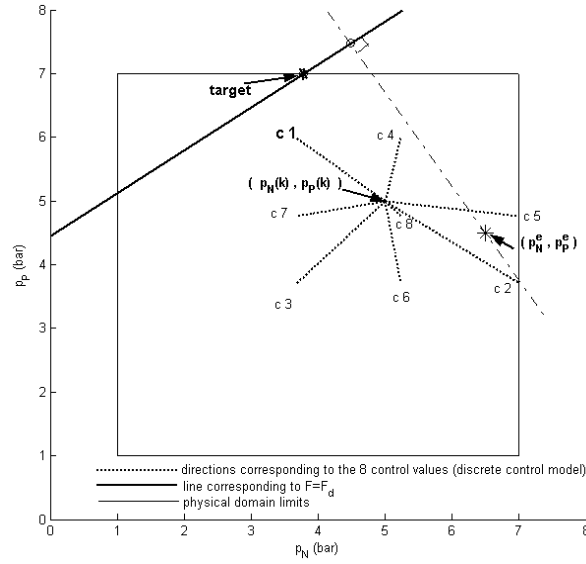


Figure 4 :target in (p_P, p_N) plane when $F_d = 350$ N

For each sample time, the algorithm:

- calculate the eight reachable points in (p_N, p_P) plane corresponding to the eight control values thanks to the discrete model (5),
- calculate the target point corresponding to the desired force,
- calculate the Euclidean distance between the eight reachable points and the target point,
- choose the control that leads to the fewest Euclidean distance. For examples control c4 in figure 3 and control c1 in figure 4.

The chosen control is then applied during ten milliseconds. Thanks to this algorithm, good performances of the force tracking have been obtained.

3.2 Position control

The figure 5 presents the general structure for position control.

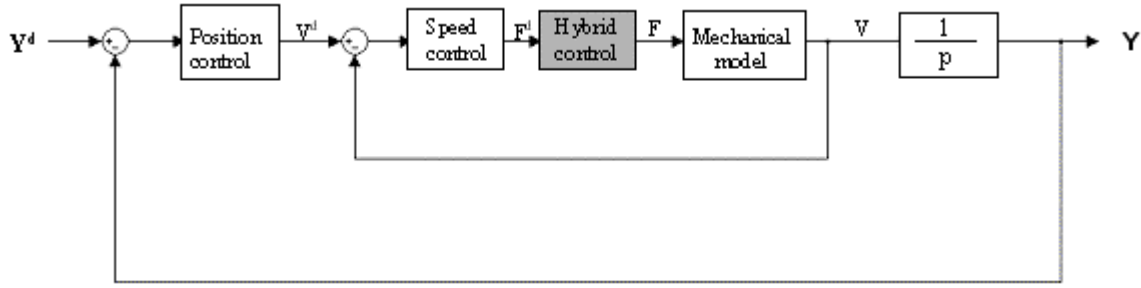


Figure 5: General structure for position control

To simplify the determination of the position and speed controllers, the force response will be considered as perfect so in the following:

$$\frac{F}{F_d} = 1 \quad (7)$$

The figure 6 presents the discrete time position control scheme.

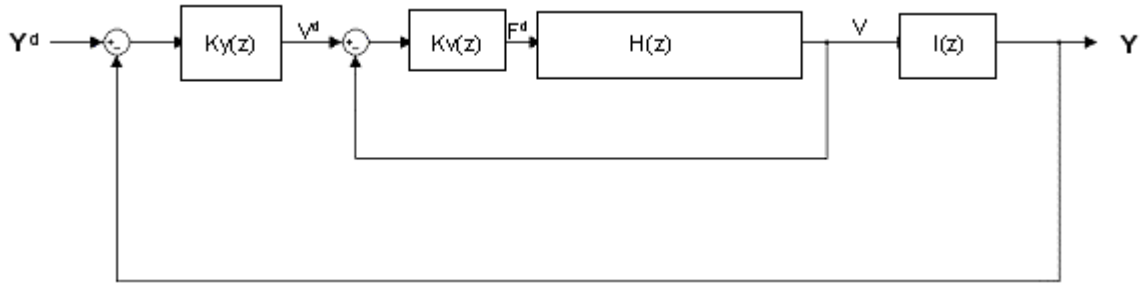


Figure 6: Discrete time position control scheme

A classical approach (18) permits to determine $K_v(z)$ and $K_y(z)$ by chosen the closed-loop dynamic behaviour in speed and position as an equivalent classic second order filter response:

$$\frac{V(p)}{V_d(p)} = \frac{1}{1 + 2 \frac{x_v}{w_v} p + \frac{1}{(w_v)^2} p^2} \quad \text{and} \quad \frac{Y(p)}{Y_d(p)} = \frac{1}{1 + 2 \frac{x_y}{w_y} p + \frac{1}{(w_y)^2} p^2}$$

In the following experimentation, $w_v = 30 \text{ rad/s}$, $x_v = 0.9$, $w_y = 10 \text{ rad/s}$, $x_y = 0.9$.

4 EXPERIMENTAL RESULTS

The following results give an illustration of the overall control performance. The test bench is presented in figure 1. In this case, the two valves are two servodistributors for which the input voltage take discrete values -10V , $+0\text{V}$ or $+10\text{V}$ (see table 1). The exhaust pressure value (1 bar) and the supply pressure (7 bar) limit the pressures. The dry friction force induces $p_p^e = 4.49 \text{ bar}$ and $p_N^e = 6.49 \text{ bar}$.

The desired trajectory has been synthesised to reduce the energy consumption (17) on the same test bench so as it is easy to compare the results with those other control algorithm using control values included between plus or minus ten volt (11, 15, 17).

Two experimental results are shown:

- in the first case (figure 7), the position loop is closed all time and the hybrid algorithm works even in steady state: the aim is to minimise the steady state position error.
- in the second case (figure 8) when the absolute value of position error is less than three millimetres the position loop is open and the control is fixed to c8 (both servovalves closed): the aim is to minimise valves switching occurrences during steady state.

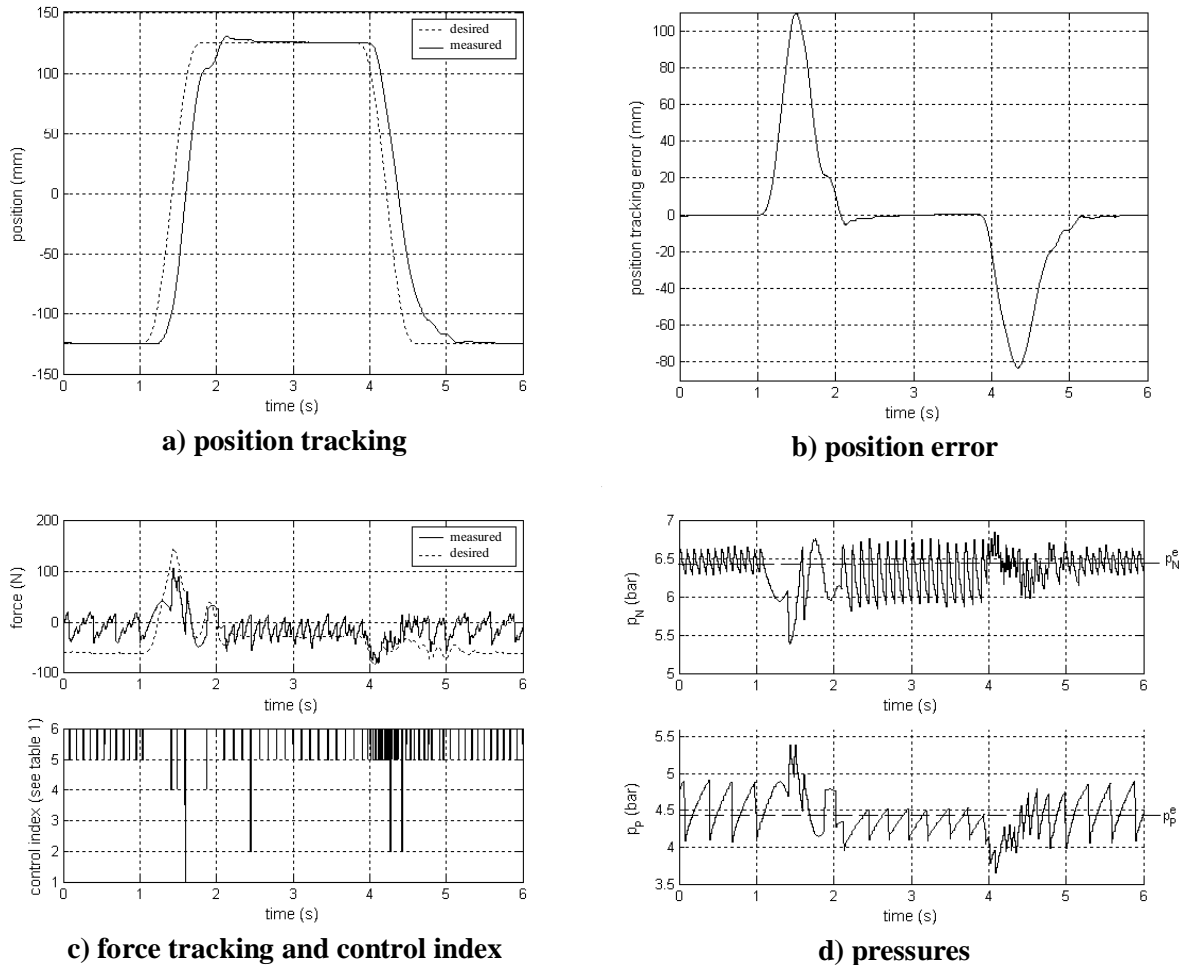


Figure 7: First case of experimental results

In these two cases, the desired position is followed with a tracking error near to 100 millimetres in dynamic stage. The pneumatic force tracking is good, excepted in steady state for second case (see figure 8c) because in steady state the hybrid algorithm is not used.

In the first case the position error in steady state is less than 0.8 millimetres, the input voltage U_P switches 47 times and the input voltage U_N switches 90 times during six seconds. In the second case, the position error in steady state can reach 2 millimetres but input valves voltages switching is limited: 31 switching for U_P and 32 for U_N .

If the application needs a great precision in position tracking, the first case will be chosen whereas the second case is less precise but the valves are less excited and so the energy consumption decreases.

Notice that the figures show two of the many tests that have been done. But we remark generally a good repeatability of experimental results when the correctors are good tuned.

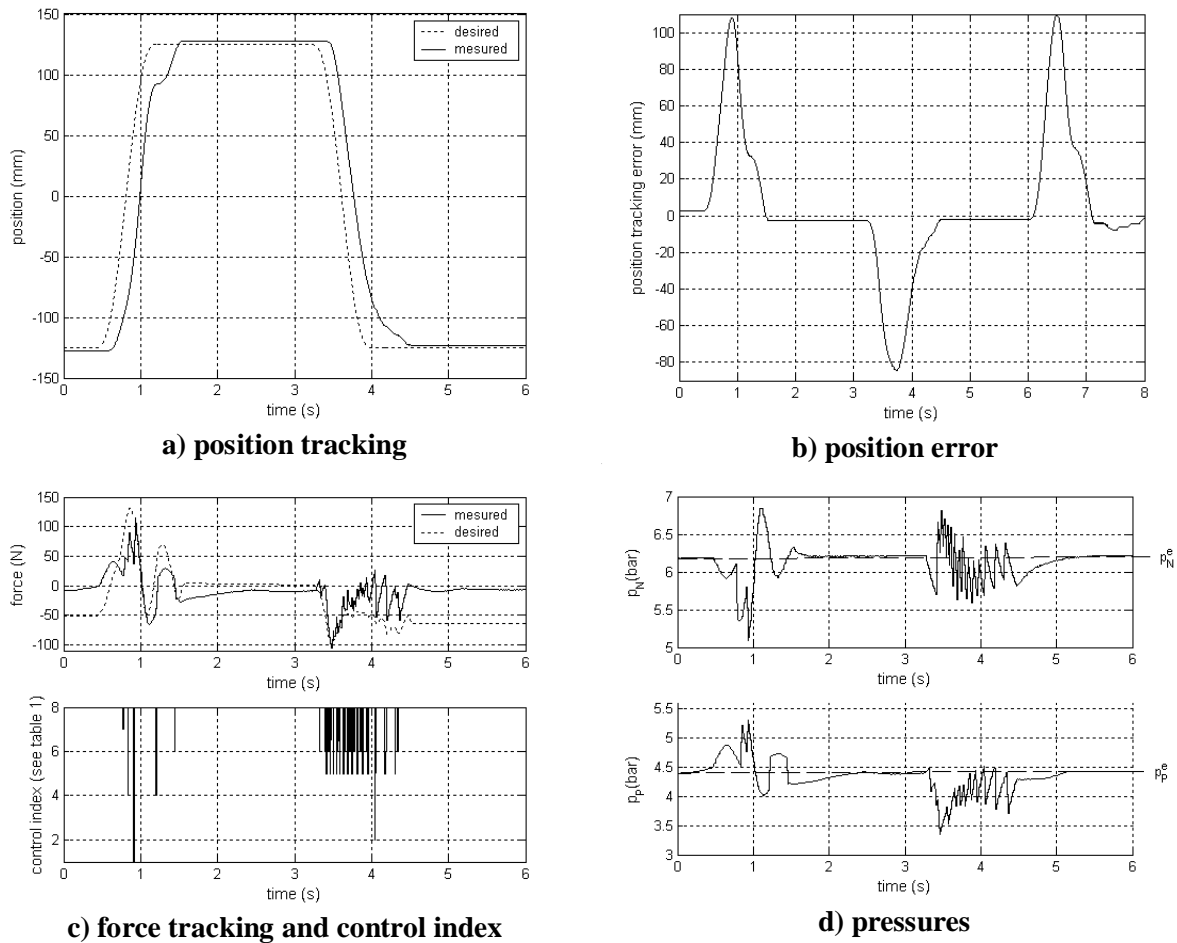


Figure 8: Second case of experimental results

5 CONCLUSION

Firstly, the first contribution of this paper concerns the transfer of the hybrid method developed for control electrical engines (1, 2) to a system composed by a pneumatic cylinder and two on/off valves. Notice that the on/off property of the valves is interesting because on/off valves are less expensive than servo valves used in classical approaches. The algorithm built with the hybrid approach, which uses a very simple control model, allows the tracking of the pneumatic force. Secondly, a speed feedback and a position feedback have been designed to control the position. Finally, the experimental tests lead to encouraging results to continue to explore this new strategy. The tuning of the correctors permits to fit the method to many applications for which the position tracking precision is less than one millimetre and the dynamic behaviour is not critical. If the precision in position tracking is less restrictive, a threshold for position tracking error under which the valves will be closed can permit to limit valve switching.

The main problem with this approach due to important number of the switching control is the valve life. Today the first results permit to solve this problem in steady state, it will be interesting to test specific hybrid control algorithm issued from electrical motor control (19) to increase dynamic performances. Future works will focus also on a more complex design of position and velocity controllers.

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