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A Combined First and Second Order Sliding Mode Approach for Position and Pressure Control of an Electropneumatic System

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Abstract— **The aim of this paper is to propose efficient multivariables controllers for an electropneumatic system. The considered system use two three-way proportional servodistributors. Generally, it is supposed that these two servodistributors are equivalent to one five-way proportional servodistributor when they are controlled with inputs of opposite signs. In this case, a monovariable control law can be established. However, with the system of two three-way servodistributors, it is possible to control two different trajectories. For example, it seems useful to control position and pressure without a degradation of the desired specifications.**
Due to uncertainties appearing in the modelization, robust controllers are necessary to ensure position and pressure tracking with high precision. For this, two control laws based on a combination of a first and second order sliding mode are proposed. The experiment results are presented and discussed.

NOMENCLATURE

y, v, a, j position (m), velocity (m/s), acceleration (m/s^2), jerk (m/s^3)
q_m mass flow rate provided from servodistributor to cylinder chamber (kg/s)
r perfect gas constant related to unit mass (J/kg/K)
l length of stroke (m)
b viscous friction coeff. *k* polytropic constant
M total load mass (kg) *S* area of the piston cylinder (m^2)
p pressure (Pa) *T* temperature (K)
V volume (m^3) *U* input voltage (V)
Subscript
ext external *D* dead volume
S supply *N* chamber N
P chamber P d desired

I. INTRODUCTION

Pneumatic cylinder systems have the potential to provide high output power to weight and size ratios at a relatively low cost. Adding to their simple structure, easy maintenance, and low component cost, pneumatic actuators are one of the most common type of industry actuators. However the complexity of the electropneumatic systems and the important range of control laws are a real industrial problem where the target is to choose the best control strategy for a given application.

The traditional and widely used approach to the control of electropneumatic systems is a fixed gain linear controller, based on the local linearization of the nonlinear dynamics about a nominal operating point [1]. This method relies on the key assumption of small range operation for the linear model to be valid. When the required operation range is large, the linear controller is likely to perform very poorly or to be unstable. The harmful effect is due to the limitation of the linear feedback controller tolerance for the adverse effect of the nonlinearities or parameters variations.

When a fixed gain linear controller cannot satisfy the control requirement, it is natural to investigate other controllers. In recent years, research efforts have been directed toward meeting this requirement. Most of them are feedback linearization [2]. However, a reasonably accurate mathematical models for the pneumatic system are required by the feedback linearization. A number of investigations have been conducted on fuzzy control algorithms [3], adaptive control [4] and robust linear control [5].

Another rather theoretically attractive robust approach is the standard sliding mode control [6]. It is believed that a robust controller can be derived based on rather little information of the system. This approach has been used in several works [7], [8]. The standard sliding mode features are high accuracy and robustness with respect to various internal and external disturbances. Specific drawback presented by the classical sliding mode techniques is the chattering phenomenon [9]. The chattering phenomenon is generally perceived as motion, which oscillates around the sliding manifold. In order to overcome this drawback, a research activity aimed at finding a continuous control action, robust against uncertainties, guaranteeing the attainment of the same control objective of the standard sliding mode approach has been carried out in recent years. The results algorithms, turned out to belong to the class of high order sliding mode control [10], [11].

In [12], a second order sliding mode controller for an electropneumatic system is presented. The system use two three-way proportional servodistributors. It is supposed that these two servodistributors are equivalent to one five-way proportional servodistributor when they are controlled with input of opposite signs. In this case**,** a monovariable control law can be established. However, The validity of the control law depends on the stability of the unobservable subsystem, which is one-dimensional. It is very difficult to obtain results about the global stability of the zero dynamics.

With the system of two three-way servodistributors, it is possible to control two different trajectories. For example, it seems useful to control position and pressure without a degradation of the desired specifications (tracking position). As is shown in [13], this strategy can allows to a minimum energy consumption. In this case, the control law is based on feedback linearization and flatness theory. However, this method of designing controller may not be effective when the mathematical model of the plant is unknown.

Due to uncertainties appearing in the modelization, robust controllers are necessary to ensure position and pressure tracking with high precision. In [14], a robust multivariable control using backstepping design is proposed. In this paper the second order sliding mode approach is employed to develop a robust position and pressure tracking controller for electropneumatic actuator. The paper is organized as follows. Section 2 describes the model of the electropneumatic actuator. Section 3 deals with the design of two second order sliding mode controllers for this system. Section 4 discusses the implementation results of the proposed control schemes on an experimental set-up.

II. Electropneumatic System Modeling

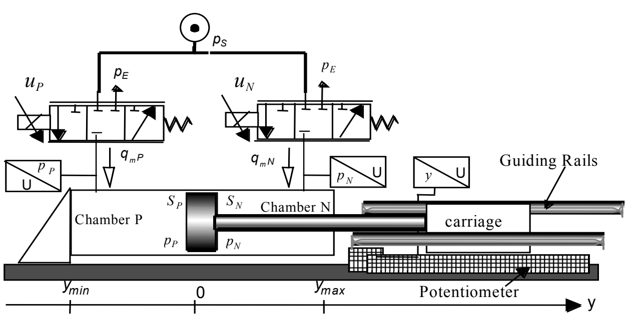

Fig. 1. The electropneumatic system

The considered system (Fig. 1) is a linear inline double acting electropneumatic servo-drive using a single rod controlled by two three-way servo-distributors. The actuator rod is connected to one side of the carriage and drives an inertial load on guiding rails. The total moving mass is 17 kg.

The electropneumatic system model can be obtained using three physical laws: the mass flow rate through a restriction, the pressure behavior in a chamber with variable volume and the fundamental mechanical equation.

The pressure evolution law in a chamber with variable volume is obtained assuming the following assumptions [15]: air is a perfect gas and its kinetic energy is negligible. The pressure and the temperature are supposed to be homogeneous in each chamber. The process is polytropic and characterized by coefficient *k*. Moreover, the electropneumatic system model is obtained by combining all the previous relations and assuming that the temperature variation is negligible with respect to average and equal to the supply temperature. The dynamics of the servodistributors may be neglected [1]. So, the model can be reduced to a static one described by two relationships $q_{mP}(u_P, p_P)$ and $q_{mP}(u_N, p_N)$ between the mass flow rates q_{mP} and q_{mN}, the input voltages u_P and u_N, and the output pressures. The mechanical equation include pressure force, viscous friction and an external constant force due to atmospheric pressure. So the following equation gives the model of the above system:

$$\begin{cases} \dfrac{dy}{dt} = v \\ \dfrac{dv}{dt} = \dfrac{1}{M}\left[S_P p_P - S_N p_N - bv - F_{ext}\right] \\ \dfrac{dp_P}{dt} = \dfrac{krT}{V_P(y)}\left[q_{mP}(u_P, p_P) - \dfrac{S_P}{rT} p_P v\right] \\ \dfrac{dp_N}{dt} = \dfrac{krT}{V_N(y)}\left[q_{mN}(u_N, p_N) + \dfrac{S_N}{rT} p_N v\right] \end{cases} \tag{1}$$

Where:

$$\begin{cases} V_P(y) = V_P(0) + S_P y \\ V_N(y) = V_N(0) - S_N y \end{cases} \text{ with: } \begin{cases} V_p(0) = V_{DP} + S_P \dfrac{l}{2} \\ V_N(0) = V_{DN} + S_N \dfrac{l}{2} \end{cases}$$

are the piping volumes of the chambers for the zero position and $V_{D(P\,or\,N)}$ are dead volumes present on each extremities of the cylinder.

The main difficulty for model (1) is to know the mass flow rates q_{mP} and q_{mN}. This model is issue of experimental measurement [16] and therefore a mathematical model for a static flow stage has been obtained from a polynomial approximation [17] affine in control (2) and the nonlinear affine model is then given by (3).

$$q_m(u, p) = \varphi(p) + \psi(p, sgn(u)) \times u \tag{2}$$

$$\begin{cases} \dfrac{dy}{dt} = v \\ \dfrac{dv}{dt} = \dfrac{1}{M}\left[S_P p_P - S_N p_N - bv - F_{ext}\right] \\ \dfrac{dp_P}{dt} = \dfrac{krT}{V_P(y)}\left[\varphi(p_P) - \dfrac{S_P}{rT} p_P v\right] + \dfrac{krT}{V_P(y)}\psi(p_P, sgn(u_P)) \times u_P \\ \dfrac{dp_N}{dt} = \dfrac{krT}{V_N(y)}\left[\varphi(p_N) + \dfrac{S_N}{rT} p_N v\right] + \dfrac{krT}{V_N(y)}\psi(p_N, sgn(u_N)) \times u_N \end{cases} \tag{3}$$

$\psi(.) > 0$ over the physical domain. With two inputs u_P and u_N, the nonlinear model of the system in Fig. 1 has the following form:

$$\underline{\dot{x}} = f(\underline{x}) + g(\underline{x}) \times \underline{U} \quad (4)$$

with

$$\underline{x}^T = (y, v, p_P, p_N)$$

where:

$$f(\underline{x}) = \begin{pmatrix} v \\ \frac{1}{M}\left[S_P p_P - S_N p_N - bv - F_{ext}\right] \\ \frac{krT}{V_P(y)}\left[\varphi(p_P) - \frac{S_P}{rT} p_P v\right] \\ \frac{krT}{V_N(y)}\left[\varphi(p_N) + \frac{S_N}{rT} p_N v\right] \end{pmatrix}$$

$$g(\underline{x}) = (g_1(\underline{x}), g_2(\underline{x})) = \begin{pmatrix} 0 & 0 \\ 0 & 0 \\ \frac{krT}{V_P(y)}\psi(p_P, sgn(u_P)) & 0 \\ 0 & \frac{krT}{V_N(y)}\psi(p_N, sgn(u_N)) \end{pmatrix}$$

$$\underline{U}^T = (u_P, u_N)$$

Using two servodistributors leads to a system with two degrees of freedom according to the control and this opportunity is exploited to achieve two different control objectives. Taking advantage of the supplementary degree of freedom issued from this new design, it is possible to control another output other than the position control.

Let's define $h(\underline{x})$ the vector constituted of the two chosen outputs: position and pressure in chamber P

$$h(\underline{x}) = \begin{pmatrix} h_1(\underline{x}) \\ h_2(\underline{x}) \end{pmatrix} = \begin{pmatrix} y \\ p_P \end{pmatrix} \quad (5)$$

In order to use a sliding mode technique, a coordinate transformation is proposed with diffeomorphism given by (6). The nonlinear affine model is then given by (7).

$$\underline{z} = \phi(\underline{x}) = \begin{bmatrix} h_1(\underline{x}) = y \\ L_f h_1(\underline{x}) = v \\ L^2{}_f h_1(\underline{x}) = a \\ h_2(\underline{x}) = p_P \end{bmatrix} \quad (6)$$

$$\begin{cases} \frac{dy}{dt} = v \\ \frac{dv}{dt} = a \\ \frac{da}{dt} = L^3{}_f h_1(\phi^{-1}(\underline{z})) + L_{g1} L_f{}^2 h_1(\phi^{-1}(\underline{z})) u_p + L_{g2} L_f{}^2 h_1(\phi^{-1}(\underline{z})) u_N \\ \frac{dp_p}{dt} = L_f h_2(\phi^{-1}(\underline{z})) + L_{g1} h_2(\phi^{-1}(\underline{z})) u_P + L_{g_2} h_2(\phi^{-1}(\underline{z})) u_N \end{cases} \quad (7)$$

Where

$$L^3{}_f h_1(\underline{x}) = \frac{krT}{M}\left[\frac{S_P}{V_P(y)}\varphi(p_P) - \frac{S_N}{V_N(y)}\varphi(p_N) - \frac{v}{rT}\left(\frac{S_P^2 p_P}{V_P(y)} + \frac{S_N^2 p_N}{V_N(y)}\right)\right] - \frac{b}{M^2}(S_P p_P - S_N p_N - bv - F_{ext}) \quad (8)$$

$$L_{g_1} L_f{}^2 h_1(\underline{x}) = \frac{krTS_P}{MV_P(y)}\psi(p_P, sign(u_P)) > 0 \quad (9)$$

$$L_{g_2} L_f{}^2 h_1(\underline{x}) = -\frac{krTS_N}{MV_N(y)}\psi(p_N, sgn(u_N)) < 0 \quad (10)$$

$$L_f h_2(\underline{x}) = \frac{krT}{V_P(y)}\left[\varphi(p_P) - \frac{S_P}{rT} p_P v\right] \quad (11)$$

$$L_{g_1} h_2(\underline{x}) = \frac{krT}{V_P(y)}\psi(p_P, sgn(u_P)) > 0 \quad (12)$$

$$L_{g_2} h_2(\phi^{-1}(\underline{z})) = 0 \quad (13)$$

Using the electropneumatic model (7), the control laws are synthesized in the next section using a combination of first and second order sliding mode technique.

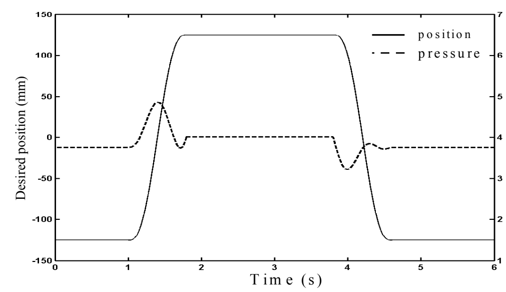

Fig. 2. Desired trajectories

The aim of the control law is to respect a good accuracy in term of position and pressure tracking for a desired trajectories. The relative degree of the position and the pressure are respectively three and one. This means that the electropneumatic system can only track position trajectories at least three times differentiable and pressure trajectories at least one time differentiable. The desired trajectory have been carefully chosen in order to respect the differentiability required. (see Fig. 2).

III. Sliding Mode Controllers For Electropneumatic System

A. Second order sliding mode

The effective application of sliding mode control to pneumatic systems needs to resolve the problem related to the chattering phenomenon and the switching control signals [11]. Higher order sliding modes appear to be suitable to counteract this problems [10], [11].

Let $s(x,t)$ the sliding variable, the r^{th} order sliding mode is determined by the equalities s= $s=\dot{s}=\ddot{s}=...=s^{(r-1)}=0$, which form an r-dimensional condition on the state of the dynamic system. In general, any *r*-sliding controller needs $s,\dot{s},\ddot{s},...,s^{(r-1)}$ to be made available, i.e. *2*-sliding controller needs $s,\dot{s}$ to be made available[10].

Consider the system (14) with relative degree equal to two with respect to the input:

$$\begin{cases}\dot{x}_1 = x_2\\ \dot{x}_2=\phi(x,t)+\psi(x,t)u\end{cases} \tag{14}$$

where $x_1=s$, $x_2=\dot{s}$ and $\phi(x,t)$, $\psi(x,t)$ are uncertain functions with $0<|\phi(x,t)|\le C_0$ and $0<k_m\le\psi(x,t)\le k_M$.

Several 2-sliding algorithms have been presented in the literature [11],[19]. In this paper, twisting algorithm and the algorithm with a prescribed convergence law are used.

The twisting algorithm is defined by the following control law [11]:

$$u=\begin{cases}-\lambda_m\, sgn(s) & if\ s\dot{s}\le 0\\ -\lambda_M\, sgn(s) & if\ s\dot{s}>0\end{cases} \tag{15}$$

The corresponding sufficient conditions for the finite time convergence to the sliding manifold are [10]:

$$\lambda_M>\lambda_m>\frac{C_0}{k_m} \tag{16}$$

$$k_m\lambda_M-C_0>k_M\lambda_m+C_0 \tag{17}$$

In this case, the trajectories are characterized by twisting around the origin on the phase portrait of sliding variable. The convergence of the algorithm with a prescribed convergence law is different [10], [18]. The trajectories converge to a prescribed function and then to the origin. The control law is defined as follows [18]:

$$u=-\alpha\, sgn(\dot{s}-g(s))$$

where $\alpha>0$, and the continuous function $g(s)$ is smooth everywhere except on $s=0$. It is assumed that all the solutions of the equation $\dot{s}=g(s)$ vanish in a finite time and the function $\frac{\partial g(s)}{\partial s}g(s)$ is bounded:

$$\left|\frac{\partial g(s)}{\partial s}g(s)\right|<C_1 \tag{18}$$

The sufficient condition for the finite time convergence to the sliding manifold is defined by the following inequality [18]:

$$\alpha>\frac{C_0+C_1}{K_m} \tag{19}$$

B. Sliding variable and control synthesis

Let us define a vector S of components $s_i\ (i = 1,2)$ by:

$$\begin{aligned}s_1&=\lambda(y-y_d)+(v-v_d)\\ s_2&=(p_P-p_{Pd})\end{aligned} \tag{20}$$

where λ is a positive parameter. Consider the second time derivative of s_1 and the first time derivative of s_2:

$$\begin{aligned}\ddot{s}_1&=\lambda(a-a_d)-j_d-\frac{kv}{M}\left(\frac{S_P^2p_P}{V_P(y)}+\frac{S_N^2p_N}{V_N(y)}\right)\\ &+\alpha(y,v,p_P,p_N)+L_{g_1}L_f{}^2h_1(\phi^{-1}(\underline{z}))u_P\\ &+L_{g_2}L_f{}^2h_1(\phi^{-1}(\underline{z}))u_N\end{aligned} \tag{21}$$

$$\dot{s}_2=-\frac{kS_P}{V_P(y)}p_Pv+\phi(y,p_P)+L_{g1}h_2(\phi^{-1}(\underline{z}))u_P-\dot{p}_{Pd} \tag{22}$$

with

$$\alpha(y,v,p_P,p_N)=\frac{krT}{M}\left(\frac{S_p}{V_p(y)}\varphi(p_p)-\frac{S_N}{V_N(y)}\varphi(p_N)\right) -\frac{b}{M^2}(S_Pp_P-S_Np_N-bv-F_{ext}) \tag{23}$$

and

$$\phi(y,p_P)=\frac{krT}{V_P(y)}\varphi(p_P) \tag{24}$$

Functions $\alpha(y,v,p_P,p_N)$ and $\phi(y,p_P)$ contains all uncertainties, i.e., the leakage polynomial function and friction. Using the static feedback[1]:

$$\underline{U}=B^{-1}\times[A+\underline{V}] \tag{25}$$

with:

$$\underline{U}=[u_P\quad u_N]^T \tag{26}$$

$$B=\begin{bmatrix}Lg_1L_f{}^2h_1(\phi^{-1}(\underline{z}) & L_{g_2}L_f{}^2h_1(\phi^{-1}(\underline{z})\\ L_{g_1}h_2(\phi^{-1}(\underline{z})) & 0\end{bmatrix} \tag{27}$$

$$A=\begin{bmatrix}-\lambda(a-a_d)+j_d+\frac{kv}{M}\left(\frac{S_p^2p_P}{V_P(y)}+\frac{S_N^2p_N}{V_N(y)}\right)\\ \frac{kS_P}{V_P(y)}p_Pv+\dot{p}_{Pd}\end{bmatrix} \tag{28}$$

and $\underline{V}=[v_1\quad v_2]^T$ is the new control vector, one gets

$$\ddot{s}_1=\alpha(y,v,p_P,p_N)+v_1 \tag{29}$$

$$\dot{s}_2=\phi(y,p_P)+v_2 \tag{30}$$

There exist positives constants C_0 and C_1 so that

$$|\alpha(y,v,p_P,p_N)|<C_0 \tag{31}$$

$$|\phi(y,p_P)|<C_1 \tag{32}$$

The system is subdivided into two subsystems. The second subsystem (30) is associated to the pressure and it is stabilized by a first order sliding mode controller:

$$v_2=-(C_1+k_1)\,sgn(s_2) \tag{33}$$

where k_1 is a positive constant.

Equation (29) is associated to the position. According to (31) and (32), one can apply the twisting algorithm

[1] According to the physical domain, the determinant of the matrix B is never equal to 0.

previously presented:

$$v_1 = \begin{cases} -\lambda_m \, sgn(s_1) & if \; s_1 \dot{s}_1 \le 0 \\ -\lambda_M \, sgn(s_1) & if \; s_1 \dot{s}_1 > 0 \end{cases} \tag{34}$$

$\lambda_M > \lambda_m > C_0$ and $\lambda_M > \lambda_m + 2C_0$ are the corresponding conditions for the finite time convergence to the sliding manifold.

This algorithm is based on an adequate commutation of the control between two different values so that the trajectories in the phase plan of execute an infinite number of rotations while converging in finite time to the origin.

The convergence of the algorithm with a prescribed convergence law is different. The trajectories converge to a prescribed function and then to the origin (see next section). The control law is defined as follows:

$$v_1 = -\alpha \, sgn(\dot{s}_1 - g(s_1)) \tag{35}$$

The function $g(s_1)$ is chosen as follow[18]:

$$g(s_1) = -k|s_1|^{\frac{1}{2}} \, sgn(s_1) \tag{36}$$

Given the system described in (3), the actual inputs defined in (25), with v_1 and v_2 are given by (34) (33) or (35) (33), are applied to the electropneumatic system. A sliding mode occurs on S leading to desired tracking property for the position and the pressure.

IV. Experimental Results

These controllers were implemented using a dSpace DS1104 controller board with a dedicated digital signal processor. The sensed signals, all analog, were run through the signal conditioning unit before being read by the A/D converter. The velocity cylinder is determined by analog differentiating and low-pass filtering the output of the position given by an analog potentiometer. The acceleration information is obtained by differentiating numerically the velocity.

Some experiment results are provided here to demonstrate the robustness of these controllers. Firstly, the twisting algorithm is applied. Fig. 3 shows the position error and the pressure error.

From the experiments results, a good tracking responses are obtained for the position and pressure in chamber *p*. In steady state, the position error is about *0.23 mm*, which is better than with a feedback linearization control law [13]. The maximum position tracking error is less then *3 mm*, i.e. 1,2 % of the total displacement magnitude. The maximum pressure tracking error is equal to *0.06 bar*. However, the proposed controller is a combination of first and second order sliding mode. Indeed, a first order sliding mode controller is used in order to track the desired pressure. That is why, the chattering phenomena cannot be completely eliminated.

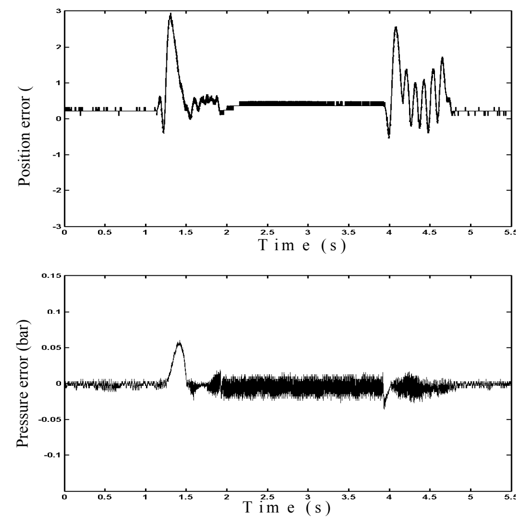

Fig.3. Position and pressure errors

To illustrate the convergence of the twisting algorithm, the initial position is set at 125 mm. The algorithm is employed when the desired position reaches -125mm (see Fig. 4).

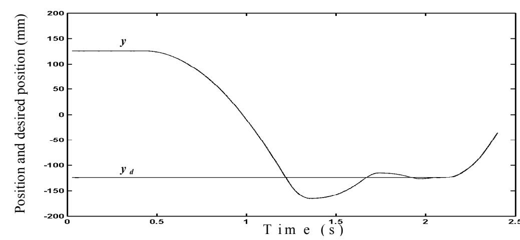

Fig.4. Position and desired position (mm)

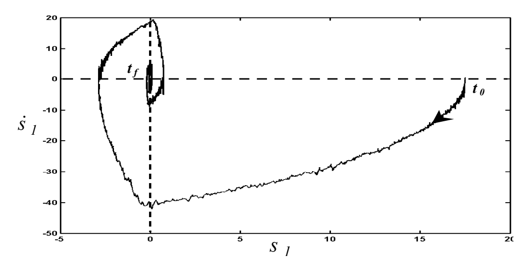

Fig.5. Phase trajectory

Fig. 5 shows the trajectories of the twisting algorithm. The trajectories are characterized by twisting around the origin on the phase portrait of sliding variable.

Fig.6 shows the position and pressure errors obtained with combined first order sliding mode and the algorithm with a prescribed convergence law.

The robust control characteristics of this algorithm controller can be observed. The maximum position tracking error is about *1,6 mm* and the maximum pressure tracking error is about *0.06 bar* at the same time.

To illustrate the convergence of the algorithm with the prescribed convergence law algorithm, it is employed when the desired position reaches –125 mm and the position is equal to 140 mm (see Fig. 7). Fig. 8 shows the trajectories of the algorithm with a prescribed convergence law. Firstly, the trajectory reach the function $\dot{s}_1 = g(s_1)$ and then the origin $s_1 = \dot{s}_1 = 0$ in finite time.

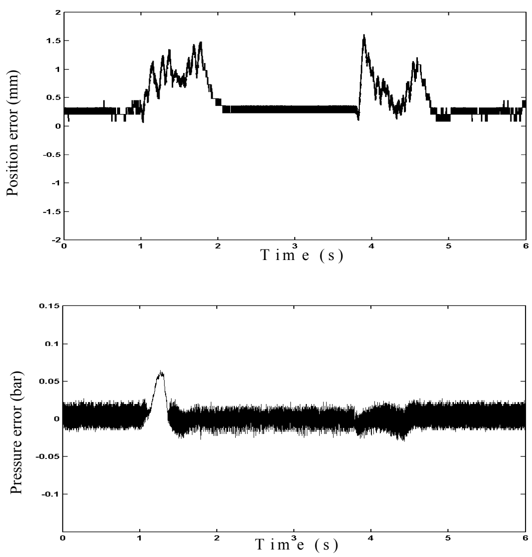

Fig. 6. Position and pressure errors

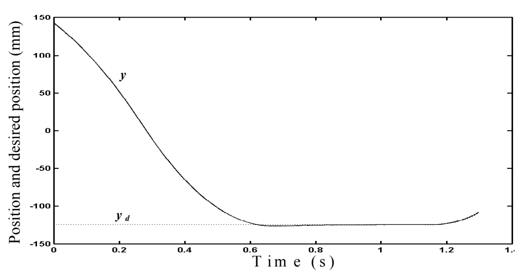

Fig.7. Position and desired position (mm)

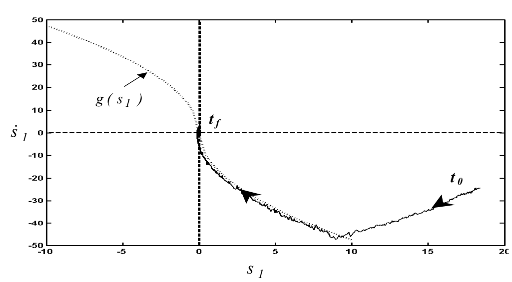

Fig.8. Phase trajectory

V. CONCLUSION

This paper has successfully demonstrated the application of two second order sliding mode controllers to control the position and the pressure of an electropneumatic system. Firstly, the mathematical model of the electropneumatic system was introduced. Then, the theoretical bases of the controllers were described in detail. The robustness vis-a-vis modeling errors has been proved. Then, experimentation was carried out to check the effectiveness of the proposed controllers. The obtained experimental results are satisfactory and are in concordance with the numerical results, the required performance are achieved.

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