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► **To cite this version:**

Mohamed Smaoui, Xavier Brun, Daniel Thomasset. Robust position control of an electropneumatic system using second order sliding mode. IEEE ISIE, May 2004, Ajaccio, France. pp.429-434 vol. 1, 10.1109/ISIE.2004.1571846 . hal-02064132

HAL Id: hal-02064132

<https://hal.science/hal-02064132>

Submitted on 20 Mar 2019

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Robust Position Control of an Electropneumatic System Using Second Order Sliding Mode

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Abstract— First order sliding mode control is an important robust control approach. For the class of systems to which it applies, first order sliding mode controller provides a systematic approach to the problem of maintaining stability and consistent performance in the face of modeling imprecision. On the other hand, the chattering phenomenon is one of the major problems in first order sliding mode control.

This paper introduces a second order sliding mode control approach in the context of position control of an electropneumatic system. The main objective is to demonstrated through this approach the undesired chattering phenomenon can be avoided while retaining the same robustness of first order sliding mode control.

Index Terms—2-Sliding, Tracking, Electropneumatic system, Experiment

Nomenclature

b	viscous friction coefficient (N/m/s)
k	polytropic constant
M	total load mass (kg)
p	pressure in the cylinder chamber (Pa)
q_m	mass flow rate provided from servodistributor to cylinder chamber (kg/s)
r	perfect gas constant related to unit mass (J/kg/K)
S	area of the piston cylinder (m ²)
T	temperature (K)
V	volume (m ³)
y, v, a	position (m), velocity (m/s) acceleration(m/s ²)
$\varphi(\cdot)$	leakage polynomial function (kg/s)
$\psi(\cdot)$	polynomial function (kg/s/V)
l	length of stroke (m)
Subscript	
ext	external
D	dead volume
S	supply
N	chamber N
P	chamber P
d	desired

I. INTRODUCTION

The sliding-mode controller (SMC) design is an attractive approach. The primary characteristic of SMC is that the feedback signal is discontinuous, switching on one or several manifolds in the state-space. When the state trajectory crosses each discontinuity surfaces in the state-

space, the structure of the feedback system and hence its dynamics will be altered. Under certain circumstances, all motions in the neighborhood of a manifold are directed towards the manifold and, thus, a sliding motion on a predefined subspace of the state-space is established in which the system state repeatedly crosses the switching surfaces. This mode has useful invariance properties even in the presence of uncertainties in the plant model and, therefore, is a good candidate for robust tracking control of uncertain nonlinear systems [1][2] [3][4].

Specific drawback presented by the classical sliding mode techniques is the chattering phenomenon [5]. The chattering phenomenon is generally perceived as motion, which oscillates about 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 second order sliding mode control algorithms [6] [7] In literature, many algorithms have been presented [8] [9][10].

The relative degree of the sliding variable is one with respect to the input, which in the case of position control of electropneumatic system, contains position error, velocity error and acceleration error. The use of the derivative of the sliding variable information in control synthesis, who is the case for several second order sliding mode algorithms, implies the need of jerk information, so it is necessary to differentiate the acceleration. In the other hand, in order to reduce the number of sensors used, the velocity cylinder is determined by analog differentiating and low-pass filtering the output of the position and the acceleration information is obtained by differentiating numerically the velocity. It is clear that the use of an algorithm which does not need information about the derivative of the sliding variable is a good solution to find the best performances since one will not differentiate the position three times.

The super twisting algorithm [9][11][12] has been shown to be robust and stable. Moreover it does not requires the time derivative of the sliding variable which in the case of position control of electropneumatic system would involve jerk (derivative of acceleration). For this, the super twisting algorithm is chosen to develop a robust control law for electropneumatic system.

The remainder of the paper is organized as follows: Section 2 presents super twisting algorithm in an informal

setting. In section 3, a nonlinear model describing the electropneumatic system is presented. The design of a second super twisting controller for this system is derived. Section 4 is devoted to the experimental results. The effectiveness of the proposed super twisting control is verified for the electropneumatic system. Section 5 concludes the paper.

II. SUPER TWISTING ALGORITHM

The effective application of sliding mode control to pneumatic systems needs to face the problem related to the chattering phenomenon and the switching control signals. Higher order sliding modes appear to be suitable to counteract this problems[13].

The main idea behind higher order sliding modes is to act on the higher order derivatives of the sliding variable s rather than the first standard sliding modes. Keeping the main advantage of the standard sliding modes, it has the additional advantage that it removes the chattering effect.

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

Super twisting is a 2-sliding algorithm, where the control law does not need any information on the time derivatives of the sliding variable. The super twisting algorithm has been developed and analyzed by Levant in [9].

Consider a system with relative degree one with respect to the input as in (1)

$$\dot{s} = \phi(s, t) + \psi(s, t)u \quad (1)$$

In this case, one can write:

$$\ddot{s} = a(s, u, t) + b(s, t)\dot{u} \quad (2)$$

where,

$$0 < |a(\cdot)| \leq c_0 \text{ and } 0 < k_m \leq b(\cdot) \leq k_M \quad (3)$$

The super twisting algorithm defines the control law, $u(t)$, as the combination of two terms:

$$u = u_1 + u_2$$

$$u_1 = \begin{cases} -u & |u| > I \\ -\alpha \text{sign}(s) & |u| \leq I \end{cases} \quad (4)$$

$$\dot{u}_2 = \begin{cases} -\lambda |s_0|^\rho \text{sign}(s) & |s| > s_0 \\ -\lambda |s|^\rho \text{sign}(s) & |s| \leq s_0 \end{cases}$$

where $|s| \leq s_0$ is called the linearity region and in this set conditions (3) are satisfied. The super twisting algorithm converges in finite time and the corresponding sufficient conditions for finite time convergence are as follows[16]:

$$\alpha > \frac{c_0}{k_m} > 0 \quad (5)$$

$$\rho(\lambda k_m)^\rho > (k_M \alpha + c_0)(2k_M)^\rho \quad (6)$$

$$0 < \rho \leq 0.5 \quad (7)$$

Such an algorithm may be efficiently implemented because of its simplicity.

III. CONTROL DESIGN FOR AN ELECTROPNEUMATIC SYSTEM

Electropneumatic systems are encountered in a wide range of modern industrial applications because of their relatively small size, light weight, high speed and easy power translation [14].

In literature, the position control has received by far the most attention. However, most of the solutions have been based on the use of PID or the linearization of the nonlinear dynamics, despite the fact that the underlying dynamics are nonlinear with inevitable modeling uncertainties.

Robust control design of the system is another approach that has been followed in previous work. In [15], the sliding mode control (SMC) strategy has been adopted for the control of electropneumatic system. However, chattering in the control action, which is inherent in first order sliding mode, can easily excite high frequency modes. For this a second order sliding mode algorithm is proposed to control the electropneumatic system.

The considered system (Fig. 1) is a linear 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 guide rails. The total moving mass is 17 kg.

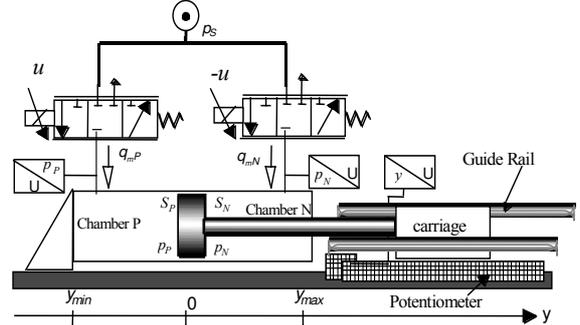


Fig. 1. The electropneumatic system

Table 1 shows the specifications of the pneumatic actuator.

Diameter of piston	$3.2 \cdot 10^{-2}$ (m)
Diameter of rod	$2 \cdot 10^{-2}$ (m)
Length of stroke	0.5 (m)
Dead volume V_{DP}	10^{-5} (m ³)
Dead volume V_{DN}	$1.2 \cdot 10^{-5}$ (m ³)

Table 1 Specifications of the pneumatic actuator

a. Model for control

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.

In our case, the bandwidth of the Servotronic Joucomatic servodistributor and the actuator are respectively about 200 Hz and 2,4 Hz. Using the singular perturbation theory, the dynamic of the servodistributors are neglected and his model can be reduced to a static one described by two relationships $q_{mP}(u, p_P)$ and $q_{mN}(-u, p_N)$ between the mass flow rates q_{mP} and q_{mN} , the input voltage u , and the output pressures p_P and p_N . The pressure evolution law in a chamber with variable volume is obtained assuming the following assumptions [16]: air is a perfect gas and its kinetic energy is negligible. The pressure and the temperature are 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 only friction regarded was viscous friction

So the following equation gives the model of the above system:

$$\begin{cases} \frac{dy}{dt} = v \\ \frac{dv}{dt} = \frac{1}{M} [S_P p_P - S_N p_N - bv - F_{ext}] \\ \frac{dp_P}{dt} = \frac{krT}{V_P(y)} \left[q_m(u, p_P) - \frac{S_P}{rT} p_P v \right] \\ \frac{dp_N}{dt} = \frac{krT}{V_N(y)} \left[q_m(-u, p_N) + \frac{S_N}{rT} p_N v \right] \end{cases} \quad (8)$$

Where:

$$\begin{cases} V_P(y) = V_P(0) + S_P y \\ V_N(y) = V_N(0) - S_N y \end{cases} \quad (9)$$

With:

$$\begin{cases} V_P(0) = V_{DP} + S_P \frac{l_{Stroke}}{2} \\ V_N(0) = V_{DN} + S_N \frac{l_{Stroke}}{2} \end{cases} \quad (10)$$

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 (8) is to know the mass flow rates q_{mP} and q_{mN} . This model is issue of experimental measurement [17][18] and therefore a mathematical model for a static flow stage has been obtained from a polynomial approximation [19] affine in control (equation (11)) and the nonlinear affine model is then given by equation (12):

$$\begin{cases} q_m(u, p_P) = \varphi(p_P) + \psi(p_P, \text{sgn}(u)) \cdot u \\ q_m(-u, p_N) = \varphi(p_N) - \psi(p_N, \text{sgn}(-u)) \cdot u \end{cases} \quad (11)$$

The nonlinear model of the system has the following form:

$$\dot{\underline{x}} = f(\underline{x}) + g(\underline{x}) \cdot u \quad (12)$$

where \underline{x} , $f(\underline{x})$ and $g(\underline{x})$ are given by relations

$$\underline{x} = \begin{bmatrix} y \\ v \\ p_P \\ p_N \end{bmatrix} \quad (13)$$

$$f(\underline{x}) = \begin{pmatrix} v \\ \frac{1}{M} [S_P p_P - S_N p_N - bv - F_{ext}] \\ \frac{krT}{V_P(y)} \left[\varphi(p_P) - \frac{S_P}{rT} p_P v \right] \\ \frac{krT}{V_N(y)} \left[\varphi(p_N) + \frac{S_N}{rT} p_N v \right] \end{pmatrix} \quad (14)$$

$$g(\underline{x}) = \begin{pmatrix} 0 \\ 0 \\ \frac{k \cdot r \cdot T}{V_P(y)} \cdot \psi(p_P, \text{sgn}(u)) \\ -\frac{k \cdot r \cdot T}{V_N(y)} \cdot \psi(p_N, \text{sgn}(-u)) \end{pmatrix} \quad (15)$$

By using model (12), the aim of this paper is to apply super twisting algorithm (4) to the electropneumatic system while trying to provide a good accuracy in term of position tracking and to avoid the chattering phenomena.

b. Control design

Let's define the sliding variable as follows:

$$s = \lambda^2 (y - y_d) + 2\lambda (\dot{y} - \dot{y}_d) + (\ddot{y} - \ddot{y}_d) \quad (16)$$

where λ is a positive parameter such that $P(x) = \lambda^2 x + 2\lambda \dot{x} + \ddot{x}$ is Hurwitz polynomial.

$$s = \lambda^2 (y - y_d) + 2\lambda (v - \dot{y}_d) + \frac{1}{M} [S_P p_P - S_N p_N - bv - F_{ext}] - \ddot{y}_d \quad (17)$$

$$\dot{s} = \phi + \psi \times u \quad (18)$$

where:

$$\begin{aligned} \phi = \lambda^2 (v - \dot{y}_d) + 2\lambda \left(\frac{1}{M} [S_P p_P - S_N p_N - bv - F_{ext}] \right. \\ \left. - \ddot{y}_d \right) + \frac{krT}{M} \left[\frac{S_P}{V_P(y)} \varphi(p_P) - \frac{S_N}{V_N(y)} \varphi(p_N) \right] \end{aligned} \quad (19)$$

$$- \frac{v}{rT} \left(\frac{S_P^2 p_P}{V_P(y)} + \frac{S_N^2 p_N}{V_N(y)} \right) \left] - \frac{b}{M^2} (S_P p_P - S_N p_N - bv - F_{ext}) - \ddot{y}_d = \phi_0 + \delta \phi$$

$$\psi = \frac{S_P}{M} \frac{krT}{V_P(y)} \psi(p_P, \text{sgn}(u)) + \frac{S_N}{M} \frac{krT}{V_N(y)} \psi(p_N, \text{sgn}(-u)) \quad (20)$$

with

$$\phi_0 = \lambda^2 (v - \dot{y}_d) + 2\lambda \left(\frac{I}{M} [S_P P_P - S_N P_N - F_{ext}] - \ddot{y}_d \right) - \frac{krT}{rTM} \left[\left(\frac{S_P^2 P_P v}{V_P(y)} + \frac{S_N^2 P_N v}{V_N(y)} \right) \right] - \ddot{y}_d \quad (21)$$

and

$$\delta\phi = -2\lambda \frac{b}{M} v + \frac{krT}{M} \left[\frac{S_P}{V_P(y)} \varphi(p_P) - \frac{S_N}{V_N(y)} \varphi(p_N) \right] - \frac{b}{M^2} (S_P P_P - S_N P_N - bv - F_{ext}) \quad (22)$$

ϕ_0 is the known nominal expression whereas the expressions $\delta\phi$ contain all the uncertainties, i.e., $\varphi(p_P)$, $\varphi(p_N)$ and b . The final controller is composed by a linearizing controller coupled to a super twisting algorithm

Using the static feedback:

$$u = \psi^{-1} [-\phi_0 + v] \quad (23)$$

one obtains:

$$\dot{s} = \delta\phi + v \quad (24)$$

There exist a positive constant c_0 , so that:

$$0 < |\delta\dot{\phi}(\cdot)| \leq c_0 \quad (25)$$

The super twisting algorithm defines the control law, $v(t)$, as the combination of two terms.

$$v = v_1 + v_2 \quad (26)$$

$$v_1 = -\alpha \text{sign}(s) \quad (27)$$

$$\dot{v}_2 = \begin{cases} -\lambda |s_0|^\rho \text{sign}(s) & |s| > s_0 \\ -\lambda |s|^\rho \text{sign}(s) & |s| \leq s_0 \end{cases} \quad (28)$$

with

$$\alpha > c_0 > 0; \quad (29)$$

$$\rho \lambda^\rho > 2^\rho (\alpha + c_0) \quad (30)$$

$$0 < \rho \leq 0.5 \quad (31)$$

The relative order of the position output of the electropneumatic system is three. This means that there are "three integrators" between the control input of the system and the output, which is the cylinder piston position:

- This is why 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, but, in the physical domain, the local stability has been proved [20].
- This means also that the electropneumatic system can only track position trajectories at least three times differentiable. The desired trajectory has been carefully chosen in order to respect the differentiability required.

IV EXPERIMENT RESULTS

This controller was 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.

The velocity cylinder is determined by analog differentiating and low-pass filtering the output of the position. However, the accelerometer sensor is not used because of its cost. The acceleration information is therefore obtained by differentiating numerically the velocity.

The aim of the control law is to respect a good accuracy in term of position tracking for a desired trajectory defined by a fifth order polynomial function (Fig. 2). The amplitude of displacement is equal to 50% of the total stroke around the central position, i.e. the total displacement is equal to 500mm. The maximum desired velocity is equal to 0.60 m/s. The test have been made several times in the aim to see the repeatability of the experimental results. Some experiment results are provided here to demonstrate the effectiveness of the super twisting controller.

The robust control characteristics of this controller can be observed. Indeed, as is shown in Fig. 3, In steady state, the position error is about 70 μm and the maximum position tracking error is about 0.7 mm, so this control law seems more interesting that feedback linearization [21].

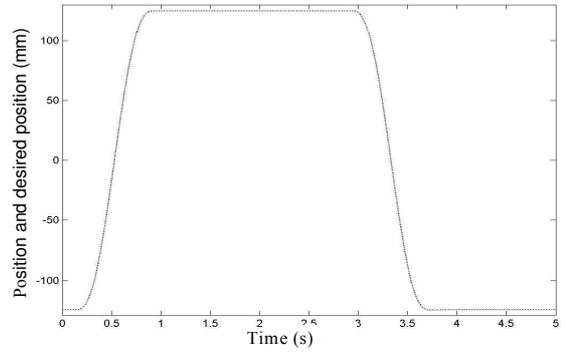


Fig. 2: Real and desired position (mm)

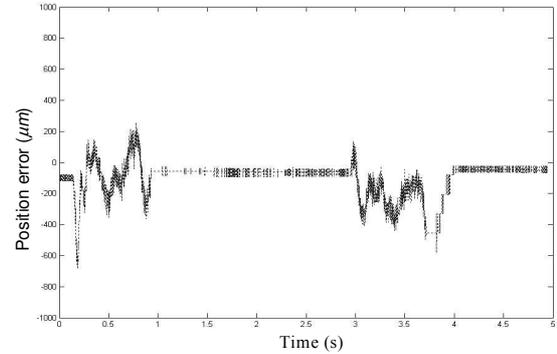


Fig.3: Position error (mm)

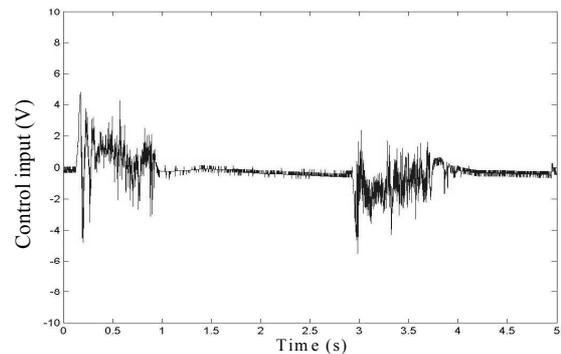


Fig.4: Control input (V)

Fig. 4 shows that the control signal is not influenced by the chattering effect.

A first order sliding mode controller is synthesized and applied to the electropneumatic system. As is shown in Fig. 5, the chattering phenomenon occurs and this reduces the performance of the controller. The chattering phenomenon is a bad property from an energetically efficient point of view. Indeed, when the control signal increases, supply energy is used. And when the control signal decreases, the pneumatic energy approaches the exhaust. So the control signal oscillations lead to inefficiency. Moreover, there is a noise that can be heard during the experiment

So by comparing the super twisting control law with that obtained by first order sliding mode (see also [15] [22]), super twisting controller seems more satisfactory lifetime of components and consumption of energy.

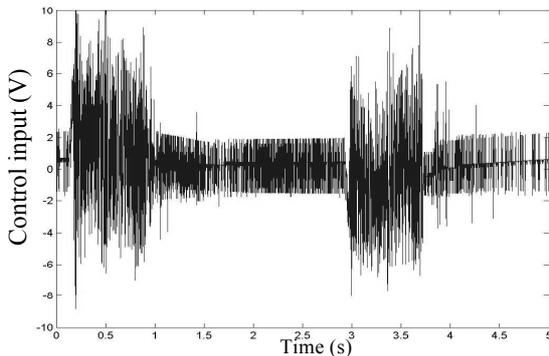


Fig.5: Control input (V)

To illustrate the convergence of the super twisting algorithm, the initial position is set at 75 mm. The algorithm is employed when the desired position reaches 120mm (see Fig. 6).

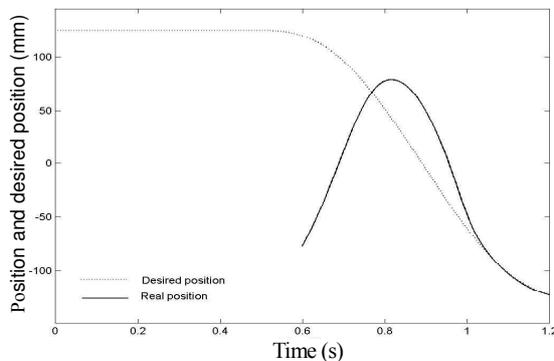


Fig.6: Real and desired position (mm)

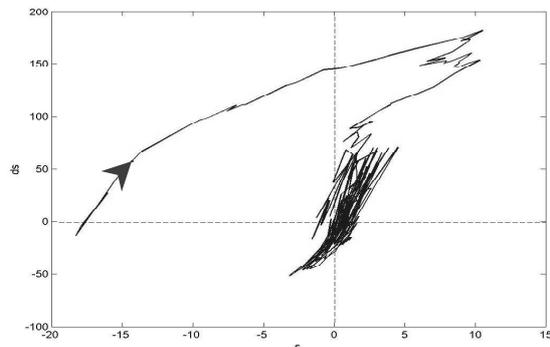


Fig.7: Phase trajectory

Fig. 7 show the trajectories of the super twisting algorithm. The trajectories of the super twisting algorithm are characterized by twisting around the origin on the phase portrait of sliding variable.

V CONCLUSION

The robustness of 2-sliding controller (super twisting algorithm) in the presence of practical engineering constraints and realities is guaranteed. Experimental results show that the trajectory tracking is done with a very good accuracy. Moreover, and contrary to the other algorithms, the super twisting algorithm does not need any information on the time derivative of the sliding variable, i.e., in our case, this algorithm does not need any information on the jerk (derivative of acceleration).

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