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Higher-order sliding modes for an electropneumatic system: differentiation and output-feedback control

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ABSTRACT

This paper develops and uses a robust differentiator via sliding modes applied to velocity and acceleration measurements. From the only measure of the position, we are being able to accurately estimate the velocity and the acceleration of a servo drive system. Initially developed by Levant, this differentiator is based on high-order sliding modes. The goal of this work is to show the importance of the choice of the differentiator design in the control of an electropneumatic system. A comparative study is made between the 2nd-order robust differentiator and a classic digital differentiation algorithm, in order to show the influence of the structure differentiation algorithm on the control of the electropneumatic system.

Keywords: high order sliding modes, robust differentiator, electropneumatic system.

1. 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 types of industry actuators (1). 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. In recent years, research efforts have been directed toward meeting this requirement. Most of them have been in the field of feedback linearization (2) (3). However, 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 (4), adaptive control (5), backstepping control (6), classical sliding mode control (7) (8) and high order sliding mode control (HOSM) (9) (10). All of the previous mentioned feedback controllers require generally measurements of acceleration for feedback. However, accelerometers are seldom used in practical drive systems. Indeed, the use of accelerometers adds cost, energy consumption, increases the complexity of the overall system (the accelerometer is mounted to the load in displacement), and reduces its reliability. Many schemes for the estimation of states

variables have been proposed in recent years. Some of these methods are based on nonlinear observer theory such as high gain observer (11), sliding mode observer (12) and backstepping observer (13). However, nonlinear state observers are difficult to implement when poor knowledge on the system dynamics is available. Moreover, in some of these cases the exact differentiation is provided only when some differentiator parameters tend to inadmissible value, like high values. With the same idea of constructing differentiator based on an observer, a recurrent proposition found in the literature is to use an extended Kalman filter (14). This is a reason that the construction of a differentiator is inevitable. Indeed, differentiators are very useful tools to determine and estimate signals. For instance, using differentiators, the velocity and acceleration can be computed from the position measurements. However, the design of an ideal differentiator is a hard and challenging task. In (15) the author has presented a comparative study among some differentiation algorithms in real time. In (16) some discussion has been done on the properties and the limitations of two different structures of linear differentiation system. Other works (17) are reposed on the arbitrary-order robust exact differentiators with finite-time convergence based on a high-order sliding modes. This high-order sliding algorithm presents a simple form and easy design, so it may be use in real-time control system.

In order to avoid measurements velocity and acceleration from sensors which aims to minimize the number of sensors implemented on the system, the comparative study of a two order differentiator allows obtaining the first and the second derivatives of a measured position. The importance choice of the differentiator design in the control of an electropneumatic system is the main subject of this article. In this work, velocity and acceleration of the actuator will be made via the 2nd-order robust differentiator. In the first section of this paper, we recall some basic concepts of higher order sliding mode. In the second part, a 2nd-order robust differentiator via a third sliding mode is presented. Section 3 describes the model of the electropneumatic actuator and equations governing the motion of this plant have been put in a nonlinear affine form. Then the design of a 2nd-order sliding mode controller is presented. Part 5, will be devoted to the experimental result. Last section is consecrated to conclusions.

2. HIGH-ORDER SLIDING MODES

Sliding Mode Control (SMC) is a robust control scheme based on the concept of changing the structure of the controller in response to the alteration state of the system in order to obtain a desired response.
The aim of a SMC device is asymptotically to bring the state of the system starting from an unspecified initial condition $x(0) = x0$ towards the origin. The sliding mode technique rests on the use of a discontinuous feedback signal having for goal to maintain the evolution of the system on a judiciously selected switching function s. A high speed switching control action is used to force the trajectory of the system to move along a chosen switching manifold in the state space. The dynamic of the closed loop system is thus fixed by the sliding surface $s = 0$. So the system motion on the surface $s = 0$ is called the sliding mode. The main feature of this approach is its insensitivity to variation in system parameters, external disturbances and modelling errors. A specific problem associated with implementation of SMC is the chattering phenomenon, which is essentially a high frequency switching of the control. Firstly, chattering was reduced by smoothing out the control discontinuity in a thin boundary layer neighbouring the switching surface (18), (19).

This kind of solution allows reducing the chattering, but it remains sensitive to input signal. To avoid this drawback some approaches were proposed. The most important approach is a high order sliding mode control (HOSM). In effect the technique of a higher order sliding mode control constitutes a generalization of the concept of standard SMC. Such a technique preserves the main properties of the standard sliding mode and removes the above restriction. They are characterized by discontinuous control acting on the higher order time derivatives of the sliding variable, instead of influencing the first time derivative as happens in standard SMC. Hence the rth order sliding mode is determined by the equalities $s = \dot{s} = \ddot{s} = s^{(r-1)} = 0$. Knowing that the order of the sliding mode is the order of the first discontinuous total time derivative of the sliding variable. Thus by moving the switching to the higher derivatives of the control, chattering in the control is totally eliminated. The HOSM is applicable to control uncertain systems with arbitrary relative degree p. The r-sliding controllers require actually only the knowledge of the system relative degree (18).

3. 2ND-ORDER ROBUST DIFFERENTIATOR

Let the input signal $f(t)$ be a function defined on $[0,\infty[$ consisting of a bounded Lebesgue-measurable noise with unknown features and an unknown base signal $f_0(t)$ with the 2th derivative having a known Lipschitz constant $C > 0$.
The diagram recursive of differentiator proposed by (17) aims to obtain a robust estimate in real time of $\dot{f}_0(t), \ddot{f}_0(t),..., f_0^{(n)}(t)$ and to be exact in absence of the noise. The 2nd-order differentiator is represented by the following equations:

$$\begin{cases} \dot{z}_0 = v_0, \\ v_0 = -\lambda_0 |z_0 - f|^{\frac{2}{3}} sign(z_0 - f) + z_1, \\ \dot{z}_1 = v_1, \\ v_1 = -\lambda_1 |z_1 - v_0|^{\frac{1}{2}} sign(z_1 - v_0) + z_2, \\ z_2 = -\lambda_2 sign(z_2 - v_1) = -\lambda_2 sign(z_1 - v_0), \end{cases}$$

Where $\lambda_0, \lambda_1, \lambda_2$ are positive gains depending on the constant Lipschitz C , as $|\ddot{f}| < C$. Here v_0 , v_1 are the outputs of the differentiator. This differentiator bases himself on the three order sliding mode.
At time $t = 0$, the initial values $z_0(0) = f(0)$ $z_1(0) = z_2(0) = 0$ were taken.
Let us define the sliding surface by: $s = z_0 - f$
After a finite time, we obtain
$s = z_0 - f = 0$

$$\dot{s} = \dot{z}_0 - \dot{f} = -\lambda_0 |z_0 - f|^{\frac{2}{3}} sign(z_0 - f) + z_1 - \dot{f} = 0,$$

$$\ddot{s} = \ddot{z}_0 - \ddot{f} = v_1 - \ddot{f} = 0$$

Then

$$-\lambda_1 |z_1 - v_0|^{\frac{1}{2}} sign(z_1 - v_0) + z_2 - \ddot{f} = 0,$$

So after a finite time, the following relations can be written: $z_1 = v_0$ is the estimation of $\dot{f}(t)$, while $z_2 = v_1$ is the estimation of $\ddot{f}(t)$ .The accuracy of the reconstruction depends on the choice of the parameters $\lambda_0, \lambda_1, \lambda_2$ in the differentiator.

4. ELECTROPNEUMATIC SYSTEM MODEL

The considered system in figure 1 is a linear inline double acting electropneumatic servo-drive using a single rod controlled by two three-way servodistributors, with a stroke equal to 500 mm. The actuator rod is connected to one side of the carriage and drives an inertial load on guiding rails. The total mass (piston, rod and carriage) is equal to 17 kg.

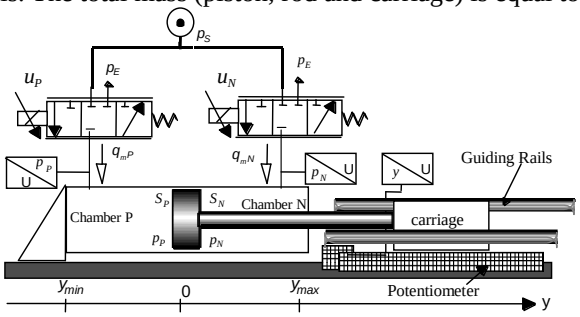


Figure 1: The electropneumatic system.

The electropneumatic system model can be obtained using three physical laws, says the mass flow rate through a restriction, the pressure behaviour in a chamber with variable volume and the fundamental mechanical equation. The pressure evolution law in a chamber with variable volume is obtained via the following assumptions: i) air is a perfect gas and its kinetic energy is negligible; ii) the pressure and the temperature are supposed to be homogeneous in each chamber; iii) the process is polytropic and characterized by coefficient k. 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. Moreover, we neglect the dynamics of the servodistributors. In such case, the servodistributors model can be reduced to two static relations between the mass flow rates $q_m^P(u_P, p_P)$ and $q_m^N(u_N, p_N)$, where u_P , u_N are the input voltages and p_P , p_N are the output pressures. The mechanical equation includes pressure force, friction and an external constant force due to atmospheric pressure. The following equation gives the model of the above system:

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

where: $\begin{cases} V_P(y) = V_P(0) + S_P y \\ V_N(y) = V_N(0) - S_N y \end{cases}$ 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 effective volumes of the chambers for the zero position and are dead volumes present at each extremities of the cylinder.

The main difficulty for the following model [1] is related to the knowledge of the mass flow rates q_m^P and q_m^N. In this paper, the results of the global experimental method giving the static characteristics of the flow stage (20) have been used. The global characterization has the advantage of obtaining simply, by projection of the characteristic series $q_m(u, p)$ on three planes: (p, q_m), (u, q_m) and (u, p). The flow stage characteristics were approximated characteristics by polynomial functions affine in control such that: $q_m(u, p) = \varphi(p) + \psi(p, sign(u))$, [2]

where $\psi(.) > 0$ over the physical domain. In the case of monocriteria, it seems more interesting to choose a system with one five-way servo-valve. In fact this structure is more attractive in terms of cost, regarding the equipment. However, in our work, we use the two three-way servo-valves which seem equivalent to one five-way when they are controlled with input of opposite signs $(u_P = u)$ and $(u_N = -u)$. So we assumed that the two servovalves are identical and symmetrical.

5. SECOND ORDER SLIDING MODE CONTROLLER

The fixed goal in our work is synthesized a control law respecting an excellent accuracy in term of position tracking for a desired position. The relative degree of the position is three. This means that the electropneumatic system can only track position trajectory at least three times differentiable.

The desired trajectory has been carefully chosen in order to respect the differentiability required (see Figure 2).

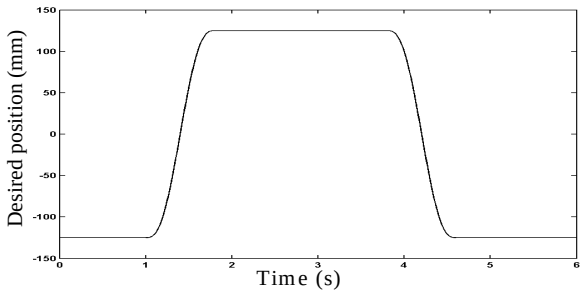


Figure2: Desired position (mm).

The choice of the order sliding mode control depends on the choice of the sliding surface and the relative degree compared to the sliding function.
Let define the sliding surface by:

$$s = \eta(y - y_d) + (v - v_d) = \eta.e_y + e_v.$$

Where η is a positive parameter, e_y and e_v are respectively a position error and velocity error. The relative degree of the position is equal two compared to the chosen variable sliding. So the order of the control law is be equal two, this mean that it should explicitly appear at least the command in the second derivative of the variable sliding s.
By using the model [1], the successive time derivatives of s are given by the expressions below:

$$\dot{s} = \eta.e_v + e_a,$$

$$\ddot{s} = \Delta\alpha(x) + \alpha(x) + \beta(x) \times u,$$

with

$$\alpha(x) = \eta.e_a - \frac{kv}{M}\left[\frac{S_N^2 p_N}{V_N(y)} + \frac{S_P^2 p_P}{V_P(y)}\right] - \frac{b}{M} \times \frac{dv}{dt} - j^d;$$

$$\beta(x) = \frac{krT}{M}\left[\frac{S_P}{V_P(y)}\psi(p_P, \text{sgn}(u)) + \frac{S_N}{V_N(y)}\psi(p_N, \text{sgn}(-u))\right];$$

$$\Delta\alpha(x) = \frac{krT}{M}\left[\frac{S_P}{V_P(y)}\varphi(p_P) - \frac{S_N}{V_N(y)}\varphi(p_N)\right].$$

$\Delta\alpha(x)$ is an uncertain and a bounded function.
The control input u is the sum of a continuous and discontinuous component, which means that u can be defined by the following feedback:

$$u = \beta^{-1}(x)[-\alpha(x) + u_d];$$

With u_d is the new discontinuous input whose role is to ensure the appearance of the sliding mode, despite the presence of uncertainties $\Delta\alpha(x)$.
In (18), a family of r-order sliding mode control with finite-time convergence is presented in (17). In this paper, an 2^{rd} order sliding mode controller from this family is used.
Indeed, the new input is defined by:

$$u_d = -\gamma_1 \,\text{sgn}(\dot{s} + \gamma_2 |s|^{\frac{1}{2}} \,\text{sgn}(s)). \quad [3]$$

In this case, two scalar parameters γ_1, γ_2 are to be adjusted.

6. EXPERIMENTAL RESULTS

Control law [3] is implemented using a Dspace 1104 controller board with the 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 position is given by an analog potentiometer. In our case, the control law is implemented by using just one sensor. Indeed, the position sensor is a NovoTECHNIK model TLH500, which have a precision and repeatability equal to 10 μm and present linearity equal to 0.05%. The gain controller γ_1 and γ_2 have been respectively tuned as $\gamma_1 = 310$ and $\gamma_2 = 15$.

Experiment results are provided here to demonstrate the effectiveness and the influence of the two different design differentiators on the control strategy. As indicated previously, a comparative study between the robust (R.D) differentiator defined above and the classic numerical derivation algorithm (C.A), studied in (15), is presented below. This classic algorithm is given by the following expression:

$$a(k) = \frac{v(k) - v(k-2)}{2T_e}, \quad [4]$$

With Te is a sampling period. We numerically build velocity from the measured position by the sensor of the electropneumatic bench, and the same thing for estimated acceleration a. Firstly, the classic algorithm [4] is used to recover the velocity and the acceleration.

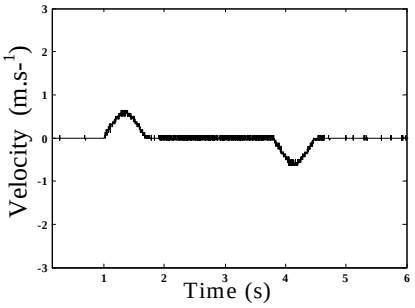


Figure 3: Velocity ($m.s^{-1}$): C.A

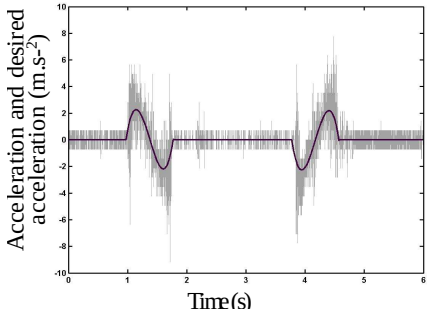


Figure 4: Acceleration and desired acceleration ($m.s^{-2}$):C.A

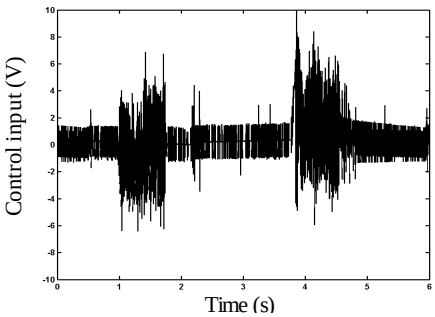


Figure 5: Control input (V): C.A

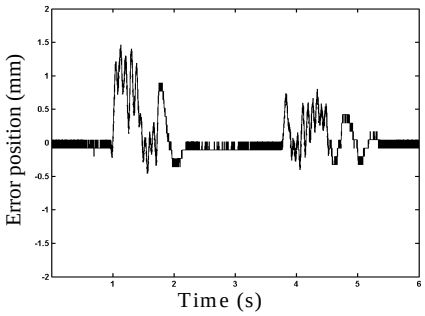


Figure 6: error position (mm): C.A

Figures 3, 4, 5 and 6 show the velocity, the estimated acceleration and the desired acceleration, the control input and the position error. The maximum position error is about 1.5mm, which is about 0.6% of the total displacement magnitude (see figure 6), when we used the classic differentiator. In additionally, it is important to note that the control law behaves well.

However, the control input (see figure 5) is affected by the chattering phenomenon, what is due to the velocity and mostly the acceleration signal. Effectively, the noise level is significant (see figure 3 and figure 4), so the control input which depends on velocity and

acceleration is affected.

It is clear that if the value of controller parameter γ_1 is decreased or the function sign is replaced by a smooth function, the control input is not affected by the chattering phenomena. But in this case, the position error becomes larger.

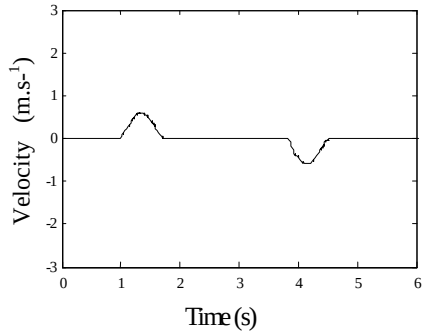


Figure 7: Velocity ($m.s^{-1}$): R.D

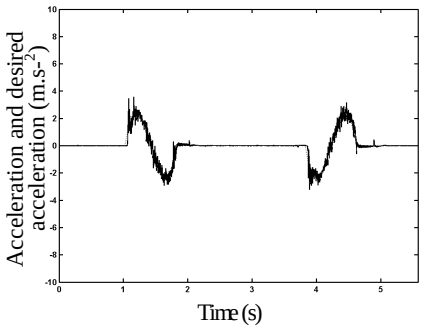


Figure 8: Acceleration and desired acceleration ($m.s^{-2}$):R.D

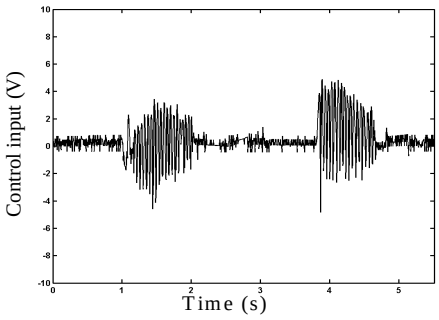


Figure 9: Control input (V): R.D

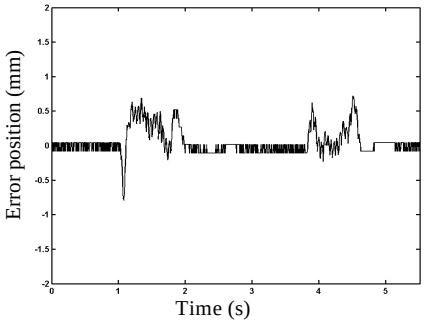


Figure 10: error position (mm): R.D

In the second part, differentiator based on third sliding order is used to recover the velocity and the acceleration signal. From figure 10, the maximum position error is about 0.8mm, so is about 0.32% of the total displacement magnitude. Remark that this value is smaller than the error value determinate with the classic differentiator. Considering figure 9, the control signal is less affected by the chattering phenomenon compared with the figure 5. Therefore the smooth control seems more satisfactory lifetime of components.

This combined controller/differentiator seems more interesting. It is normal that the control input obtained is more good than the controller combined with the classic differentiator, seen that the velocity and acceleration are also more filtered signal (see figure 7 and figure 8). However, the parameters of this robust algorithm depend on the input signal, through the Lipschitz constant of its 2nd order derivative. The Lipschitz constant is usually not known accurately beforehand and also contaminated by the noise. Consequently, the choice of these parameters for this differentiator is a difficult task.

7. CONCLUSION

In this paper, the 2nd order robust differentiator via high order sliding modes has been synthesized. We study, in this work the influence of the design differentiator on the control of an electropneumatic system and the important combination controller/differentiator in the control input. The robust algorithm shows a satisfactory result compared the classic one. The high order siding differentiator attenuates the noise related to differentiation velocity and acceleration signal. This improvement permit jointly to reduce the noise in the control input and cancelled the use of two sensors.
The results founded in this work, encourage testing experimentally the effectiveness of this algorithm design in the other linear or non linear control strategy.

NOTATION

b	viscous friction coefficient	N/m/s
k	polytropic constant	m/s
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
V	volume	m^3
y, v, a, j	Position, velocity, acceleration, jerk	m, m/s, m/s^2, m/s^3
$\varphi(.)$	leakage polynomial function	kg/s
$\psi(.)$	polynomial function	kg/s/V
l	length of stroke	m

Subscript

D	dead volume
S	supply
N	chamber N
P	chamber P
d	desired
C.A	classic algorithm
R.D	robust differentiator

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