ANNALS of Faculty Engineering Hunedoara – International Journal of Engineering
ISSN: 1584 - 2665

LabVIEW BASED POSITION CONTROL FOR A PNEUMATIC CYLINDER

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ABSTRACT:
Design and application of the robust and accurate position control for a pneumatic cylinder based on the sliding-mode technique is presented. The controller is implemented on a LabVIEW based controller. The advantage of the proposed algorithm for position control is the accuracy better than 10 μm. The paper describes the nonlinear model of the pneumatic cylinder and the three main design steps of the proposed control method; sliding surface design, control law selection and chattering free implementation. Experimental investigations on the system with proposed sliding-mode algorithm confirmed high accuracy and robustness of the horizontal pneumatic cylinder position control.
KEYWORDS:
Sliding-mode control, pneumatic positioning

1. INTRODUCTION

As an important driver element, the pneumatic cylinder is widely used in industrial applications for many automation purposes thanks to their variety of advantages, such as: simple, clean, low cost, high speed, high power to weight ratio, easy maintenance and inherent compliance. Traditionally, they are used for motion between two hard stop. The design of a stable robust position controller for a pneumatic servo-system is difficult since it is a very nonlinear time-variant controlled plant because of the compressibility of air, the friction force between the piston and the cylinder, air mass flow rate through the servo-valve, etc. By the advent of PCs with high computation power, the accurate and robust control of pneumatic actuators has become possible.

A good background of the pneumatic servo systems research can be found in [1].The early applications based on linear PID controllers proposed by Burrows and Web, 1966; Vaughan, 1965 had limited operation area. A gain scheduling PID control is proposed by Pu et al., 1993 [2] to extend the operation area. Several papers proposed automatically tuned PID controller for pneumatic servo-systems at the end of last century. Fok and Ong, 1999 [3] reached ion of ± 0.3 mm. Another solution is to employ the advanced nonlinear control strategies developed in recent years (soft computing) [4][5]. Fujiwara et al., 1995 [6]; Matsukuma et al., 1997 [7] proposed artificial neural network and Jeon et al., 1998 [8] proposed genetic algorithm to tune the PID controller. The accuracy was ± 0.1 mm in the best case.

Nonlinear adaptive controllers were proposed by Wikander, 1988 [9]; Miyata, 1989 [10]; Bobrow és Jabbari, 1991 [11]; Oyama et al., 1990 [12]; McDonell, Bobrow, 1993 [13]; Tanaka et al. 1994 [14]; Li et al. 1997 and Soong et al., The best accuracy (0.01 mm) was reached by Wikander, 1988 [9]; Nakano et al., 1993 [15] proposed a piezo-electric method with accuracy of 2μm.

Sliding-mode control was proposed by Noritsugu and Wada, 1989 [16]; Tang and Walker, 1995 [17]; Pandian et al., 1997 [18]; Hamerlain, 1995 [19]; Bouri et al., 1996; Surgenor and Vaughan, 1997 [20]; Paul et al., 1994 [21]; Song and Ishida, 1997 [22] but the accuracy was limited. The goal of this paper is to improve the accuracy of the existing sliding-mode type controllers (e.g. relay type).

Sliding-mode control was introduced in the late 1970's [23, 24] as a control design approach for the control of robotic manipulators. In the early 1980's, sliding-mode was further introduced for the control of induction motor drives [25]. These initial works were followed by a large number of research papers in robotic manipulator control [26], in motor drive control and power electronics [27]. However, despite the theoretical predictions of superb closed-loop system performance of sliding-mode, some of the experimental work indicated that sliding-mode has limitations in practice, due to the need for a high sampling frequency to reduce the high-frequency oscillation phenomenon about the sliding-mode manifold - collectively referred to as „chattering”. In most of the experimental work involving sliding-mode, the effort spent on understanding the theoretical basis of sliding-mode control is generally minimized, while a great deal of energy was invested in empirical techniques to reduce chattering. Among these experimental studies, a few succeeded in showing closed-loop system behavior which was predicted by the theory [28]. Those who failed to realize, the experimental designs successfully, concluded that chattering is a major problem in realizing sliding-mode control in practice.

The connection of sliding-mode control to model reference adaptive control introduced some excitement in the research community. In addition, the design of sliding-mode observers [29, 30], provided additional capabilities to a sliding-mode based feedback control loop. Finally, the issue of discrete-time sliding-mode was raised from the theoretical perspective, resulting in a number of different definitions of discrete-time sliding-mode, [31, 32].

The structure of this paper is as follows. The Section II describes the nonlinear equations of the applied pneumatic system. In Section III, a short introduction of sliding-mode control is presented and it is applied for the actual servo-pneumatic system. The main steps of the design of the proposed sliding-mode controller are explained. The Section IV describes the experimental set up and presents the measurement results.

2. MODEL OF THE SERVOPNEUMATIC SYSTEM

In order to design a robust controller and predict the control performance for the pneumatic test rig, a theoretical and practical modelling of the rig is needed (Fig. 1). The equations derived are based upon Burrows [32], see Fig 1. The dynamic of the piston is modelled by the mass “*m*”, the damping “*d*” and the spring “*k*”. The friction force is denoted by “F_f”. The piston can be moved by the pressure difference between the two sides of the piston. The pressures p_a and p_b can be influenced by the input and output air flow rates, which can be controlled by the input and output valves. Of course, the role of input and output are exchanged as the direction of the motion is changed. Since the input and output valves can be tuned simultaneously in the actual pneumatic cylinder, it is a single input system, which can be described by a second ordered nonlinear motion equation

$$m\ddot{x} = p_a(u)A_a - p_b(u)A_b - d\dot{x} - kx - F_f \tag{1}$$

where *x* is the position, *u* is the control signal measured as a percentage value of the input and output spool valves. The percentage value of 0% means that the spool valves are closed and 100% means that they are open totally. The dynamics of the spool valves are ignored. The other parameters and variables *T*, *V*, *A*, *Q* and *c* are the temperature, volume, area, heat energy and specific heat respectively. The subscription refers to the location of actual variable.

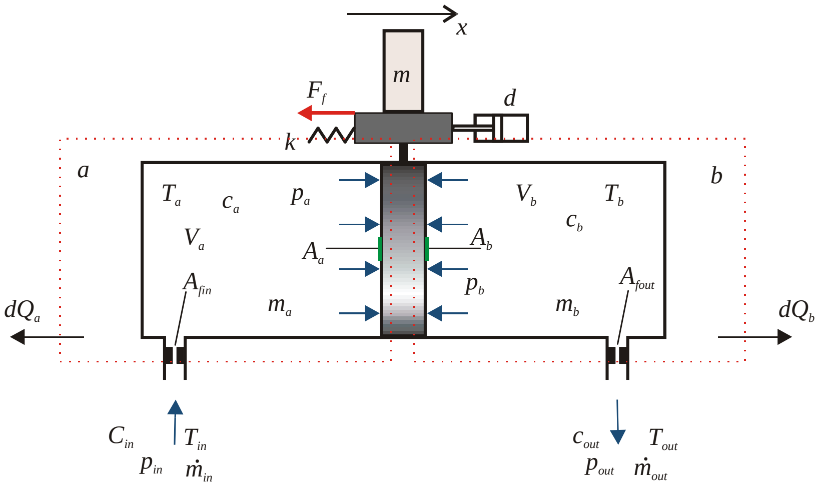

Figure 1. Structure of the pneumatic cylinder

The calculation of p_a and p_b is based on two main laws:

- Balance of the input, output and inner energies
- Balance of the input, output and inner masses

A. Energy balance

Denoting the inner energy of the air by U_{in}, and the mechanical work made the air by W, the energy balance equation for the chamber a is

$$\Delta U_a = \Delta U_{in} + \Delta Q_a + \Delta W_a \tag{2}$$

Assuming adiabatic behaviour $\Delta Q_a = 0$ and ignoring the kinetic energy of the input air, the rate of the energy change is

$$\frac{c_v(p_a\dot{V}_a + \dot{p}_a V_a)}{R} = c_p T_{in}\dot{m}_{in} - p_a A_a \dot{x} \tag{3}$$

Assuming that the c_v can be estimated by the specific heat value of air beside constant volume, and the c_p can be estimated by the specific heat value of air beside constant pressure

$$R = c_p - c_v \tag{4}$$

then the changing rate of the pressure can be expressed as:

$$\dot{p}_a = RT_{in}\frac{c_p\dot{m}_{in}}{c_v V_a} - p_a A_a \frac{c_p}{c_v V_a}\dot{x} \tag{5}$$

Similarly, for the chamber b, the changing rate of the pressure is

$$\dot{p}_b = RT_{out}\frac{c_p\dot{m}_{out}}{c_v V_b} + p_b A_b \frac{c_p}{c_v V_b}\dot{x} \tag{6}$$

B. Mass flow rate

On the basis of Bernoulli equation, the mass flow rate can be expressed by a nonlinear function

$$\dot{m}_{in} = \mu_{in} A_{in} p_{in}\sqrt{\frac{2}{R\cdot T_{in}}}\Psi_{in} \tag{7}$$

where μ_{in} is a constant depending on the type of valve and Ψ_{in} is a nonlinear term based upon pressure ratio

$$\Psi_{in} = \sqrt{\frac{\chi}{\chi-1}\left[\left(\frac{p_a}{p_{in}}\right)^{\frac{2}{\chi}} - \left(\frac{p_a}{p_{in}}\right)^{\frac{\chi+1}{\chi}}\right]} \tag{8}$$

Here χ is the specific heat ratio. Note that (8) is valid only if $p_a/p_{in} > 0.528$. If $p_a/p_{in} \le 0.528$, the speed of the air will be equal to the actual sonic speed and $\Psi_{in} = 0.484$.

The mass flow rate of the exhausted air can be expressed similarly but the roles of the "source" and "drain" must be exchange. According to that and based on (7), yields

$$\dot{m}_{out} = \mu_{out} A_{out} p_{out}\sqrt{\frac{\chi}{R\cdot T_{out}}}\Psi_{out} \tag{9}$$

where Ψ_{out} is defined as

$$\Psi_{out} = \sqrt{\frac{2}{\chi-1}\left(\frac{p_b}{p_{out}}\right)^{\frac{\chi-1}{\chi}}\left[\left(\frac{p_b}{p_{out}}\right)^{\frac{\chi-1}{\chi}} - 1\right]} \tag{10}$$

Note that (10) is valid only if $p_b/p_{out} < 1.885$. If $p_b/p_{out} \ge 1.885$ then

$$\Psi_{out} = 0.578\frac{p_b}{p_{out}} \tag{11}$$

A MATLAB and SIMULINK model based on the above equation is presented in paper [33] in order to investigate the basic properties of pneumatic actuators

3. DESIGN OF A SLIDING-MODE CONTROLLER

A good introduction into sliding-mode control can be found in [34]. The design of a sliding-mode controller consists of three main steps. First step is the design of the sliding surface, the second one is the design of the control which holds the system trajectory on the sliding surface, and the third and key step is the chattering-free implementation. The purpose of the switching control law is to force the nonlinear plant's state trajectory to this surface and keep on it. The control has discontinuity on this surface that is why some authors call it switching surface. When the plant state trajectory is „above" the surface, a feedback path has one gain; but if the trajectory drops „below" the surface, it has a different gain.

Consider a single-input, single-output second-order nonlinear dynamic system

$$\ddot{x} = f(x, \dot{x}, u) \tag{12}$$

where x is the output signal (position) of the controlled plant and u is the control signal. If x_d denotes the desired value, then the error between the reference and system states may be defined as

$$e = x_d - x \tag{13}$$

A. Sliding surface design

Classically, a scalar variable s is calculated as a linear combination of the error and its derivative.

$$s = e + \lambda \cdot \dot{e} \tag{14}$$

Let $s(\dot{e}, e) = 0$ define the sliding surface in the space of the error state. The purpose of the sliding-mode control law is to force the state trajectory of the error to approach the sliding surface and then move along the sliding surface to the origin, Fig.2.

The process of sliding-mode control can be divided into two phases, that is, the approaching phase with $s(\dot{e}, e) \neq 0$ and the sliding phase with $s(\dot{e}, e) = 0$. Here 1 denotes the approaching phase, 2 and 3 denote the sliding phase. If the system is in sliding-mode, the error is decreasing exponentially, where λ is a time constant type parameter. If λ is small, then the system response is slow but accurate. If it is big, the system response is fast but the system might chatter.

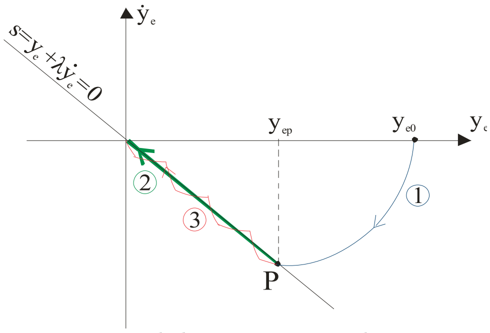

Figure 2. Sliding motion in the state space

B. Selection of the control law

In order to guarantee that the trajectory of the error vector e will translate from approaching phase to sliding phase, the control strategy must satisfy the sliding condition

$$s(\dot{e}, e) \cdot \dot{s}(\ddot{e}, \dot{e}) < 0 \tag{15}$$

This means that system trajectory should be forced to move toward the sliding surface. A proper control should be selected to satisfy the condition (15) in any time instant. Let us assume that the desired value is constant and according to (13) and (14) follows

$$\dot{s} = \dot{e} + \lambda \cdot \ddot{e} = -\dot{x} - \lambda \cdot \ddot{x} = -\dot{x} - \lambda \cdot f(x, \dot{x}, u) \tag{16}$$

If $s > 0$ or $s < 0$ the control law should be selected in a way, which ensures

$$-\dot{x} - \lambda \cdot f(x, \dot{x}, u) < 0 \quad \text{or} \quad -\dot{x} - \lambda \cdot f(x, \dot{x}, u) > 0 \tag{17}$$

The simplest control law that might lead to sliding-mode is the relay

$$u = \delta \cdot \mathrm{sign}(s) \tag{18}$$

The relay-type controller does not ensure the existence of the sliding-mode for the whole state space, and relatively big value of δ is necessary, which might cause a chattering phenomenon. If the sliding-mode exists ($s=0$ and $\dot{s} = 0$), then there is a continuous control, know as equivalent control u_{eq} which can hold the system on the sliding surface.

In practice, there is no perfect knowledge of the whole system and parameters, so, only $\hat{u}_{eq}$, the estimate of u_{eq}, can be calculated. Since $\hat{u}_{eq}$ does not guarantee convergence to the switching surface, in general, a discontinuous term is usually added to $\hat{u}_{eq}$, thus,

$$u = \hat{u}_{eq} + \delta \cdot \mathrm{sign}(s) \tag{19}$$

The role of the discontinuous term in the control law is to compensate the effect of the uncertain perturbations and bounded disturbance. The more knowledge of process is implied in the control law, the smaller discontinuous term is needed.

C. Chattering free implementation

Chattering is the main problem of sliding-mode control and chattering free implementation is the key step in design of a sliding-mode controller. A quite general solution is that the relay (which changes control value suddenly) is replaced by a saturation function. There is a boundary layer around the sliding surface where the control signal is changing continuously. If the system trajectory is close to the sliding surface and the control signal is small, then the system might stick before the goal. To avoid it a modified saturation function shown in Fig.3. is proposed.

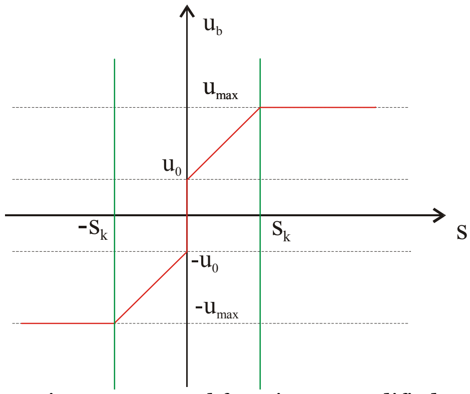

Figure 3. Control function as modified saturation function

4. THE SERVOPNEUMATIC POSITIONING SYSTEM

The system is shown in Fig.4. and Fig.5. (details can be found in [33]). It consists of a double-acting pneumatic rodless cylinder (MECMAN 170 type) with bore of 32 mm, and a stroke of 500 mm, controlled by a five-way servo- distributor (FESTO MPYE-5-M5-010-B type).

A linear encoder (LINIMIK MSA 320 type) gives the position. Velocity and acceleration are obtained by numerical derivation. Pressure sensors (Motorola MPX5999D) are set in each chamber. The controller is implemented in LabVIEW environment.

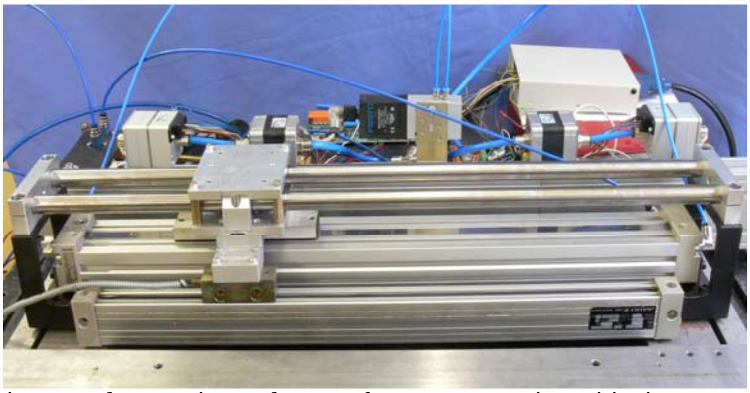
Figure 4. The experimental setup of servo-pneumatic positioning system

The control goal is to move the piston from any initial position to the target position. Using the sliding approach it is possible to minimize the positioning error.

In order to design a robust controller and predict the control performance for the pneumatic test rig, a mathematical and experimental (real world) model of the test rig is needed.

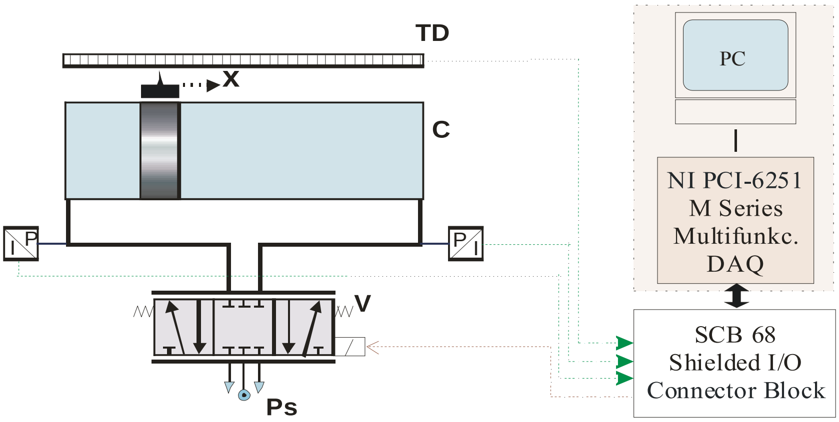

Figure 5. Configuration of pneumatic positioning system

The system pressure is set to be 6 bar, the sampling time is 1 ms. In order to analyze the positioning control methods, a real-time data acquisition program was designed in LabVIEW to capture the system output data through the connector block to the NI PCI-6251 M Series Multifunction DAQ device.

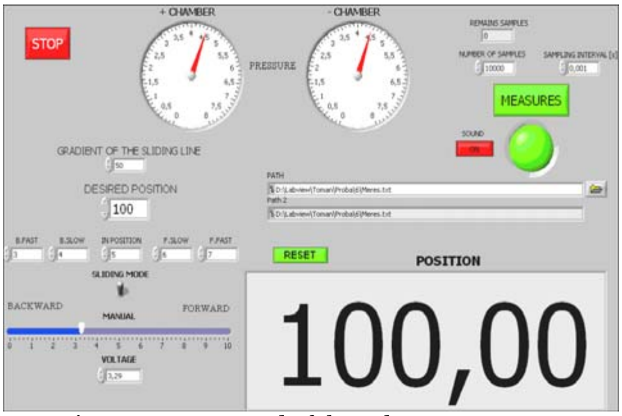

Figure 6. Front Panel of the LabVIEW program

A_a - piston area from chamber side a, m²
A_b - piston area from chamber side b, m²
A_{fin} - area of valve orifice for chamber a, m²
A_{fout} - area of valve orifice for chamber b, m²
T - temperature, K
T_{in} - temepraure of inlet air, K
T_{out} - temepraure of outlet air, K
m - mass of piston and load, kg
m_a - air mass in chamber *a*, kg
m_b - air mass in chamber *b*, kg
m_{in} - inlet air mass, kg
m_{out} - outlet air mass, kg
$\dot{m}_{in}$ - inlet air mass flow, kg/s
$\dot{m}_{out}$ - outlet air mass flow, kg/s
d - damping coefficient, Ns/m
F_f - friction force, Ns/m
k - spring stiffness, N/m
x - piston position, m
δ - discontinuos gain in control signal

A National Instruments data acquisition card (NI 6251/M) reads the signal, pressure sensors and incremental encoder into the PC. National Instruments LabVIEW will be used to collect the data imported through the DAQ card and control the position (Fig. 6.).

Symbols

$\dot{x}$ - piston velocity, m/s
$\ddot{x}$ - piston acceleration, m/s²
V_a - volume in chamber *a*, m³
V_b - volume in chamber *b*, m³
s - sliding surface, sliding line
p_a - pressure in (inlet) camber *a*, Pa
p_b - pressure in (outlet) camber *b*, Pa
p_{in} - supply pressure, Pa
p_{out} - exhaust pressure, Pa
u - control signal, %
u_{eq} - equivalent conrol, %
$\hat{u}_{eq}$ - estimate equivalent control, %
Q_a - heat energy in chamber *a*, J
U_{in} - inner energy in chamber *a*, J
W_a - mechanical work in chamber *a*, J
W - mechanical work, J
R - gas constant, Jkg⁻¹K⁻¹
Ψ_{in} - orifice flow function (inlet)
Ψ_{out} - orifice flow function (outlet)
κ - isentropic constant
λ - sliding line coefficient, time constant, *s*
c_p - constant pressure specific heats of air
c_v - constant volume specific heats of air
μ_{in} - valve flow constant for fill
μ_{out} - valve flow constant for exhaust

5. EXPERIMENTAL RESULT

The transient responses of the piston position and control signal are shown Fig.7. The experiment is a sliding-mode control with classical relay control law. The transient responses of the piston position as well as control signal are shown in Fig. 7. The steady state position error of the system with LabVIEW based relay type sliding-mode control is within ±0.01 mm, as it is shown in the enlarged position transient response in Fig.8. The accuracy of the system is limited by the applied position sensor.

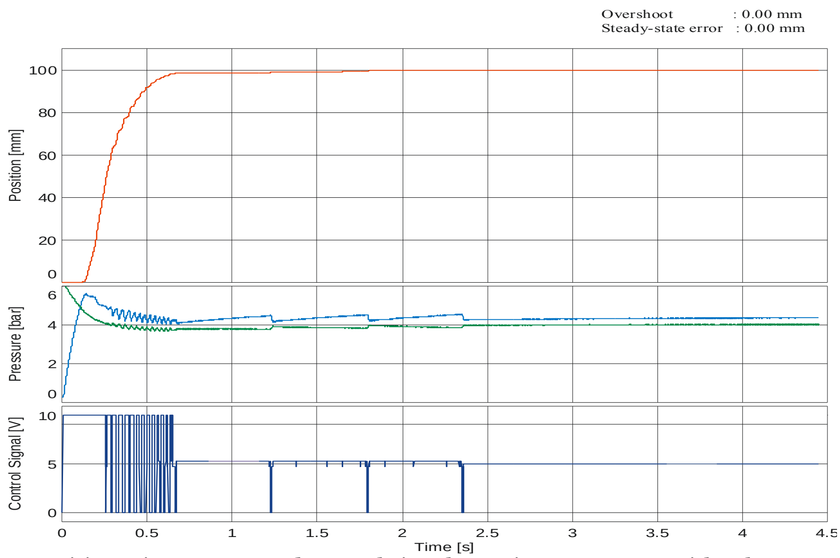

Figure 7. Piston position, air pressure and control signal transient responses with relay type SM controller

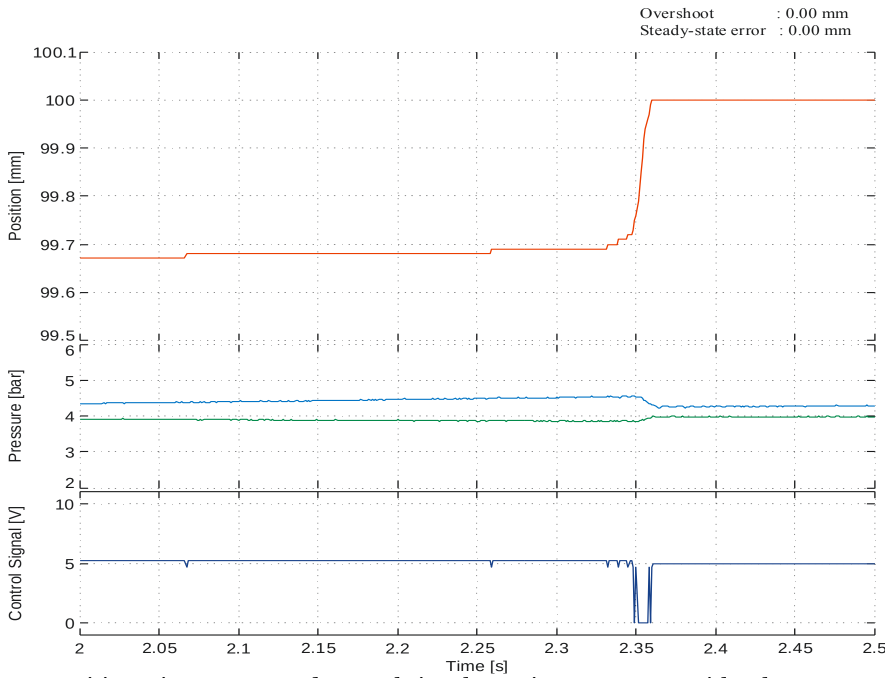

Figure 8. Piston position, air pressure and control signal transient responses with relay type SM controller - enlarged around steady state piston position (100mm)

6. CONCLUSIONS

This paper proved that pneumatic servo systems can be used for the accurate robust position control, not only for the movement between two hard stops. The experimental results showed that proposed sliding-mode controller gives fast response and good transient performance. Furthermore, the controlled system is robust to the variations of the system parameters and external disturbances and they do not require accurate modelling. The positioning system accuracy obtained is better than 10μm and it is limited by position sensor.

The final conclusion is that proposed sliding-mode controller with modified saturation function can eliminate chattering, which is the main problem in the case of sliding-mode control and can be used as a promising tool for accurate control of the servo-pneumatic systems.

Based on the laboratory measurements we can conclude that the pneumatic servo-systems can be used for precise robust position control. The sliding-mode control is a promising tool for controlling such systems. The proposed modified saturation function can eliminate the chattering, which is the main problem in case of sliding-mode control. Further works will be done on the BTL5-S101 type Micropulse Linear Transducer from Balluff with 1 μm resolution and we will be done with applying the input shaping method.

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