Sliding Mode Control Applied to Pneumatic Servo Drive

Željko Šitum, Joško Petrić and Mladen Crneković

***Abstract*—This paper describes the position control of a cylindrical pneumatic actuator using sliding mode control approach. It is difficult task to obtain an accurate mathematical model of pneumatic drives due to significant system nonlinearities, caused by air compressibility, friction effects and variations of process parameters with time. This control strategy can be used to realize the control system requirements achieving a reasonable positioning accuracy. The effectiveness of the proposed control strategy is confirmed by experiments using the existent electro-pneumatic servo system that consists of a rodless cylinder controlled by a proportional servo valve.**

***Index terms*—sliding mode controller, position control, pneumatic actuator, proportional valve**

I. INTRODUCTION

Pneumatically actuated drives are used in widespread applications of modern automation systems mainly for pick-and-place positioning problems. They can be a cheaper alternative to electric and hydraulic systems, especially for light load applications. Development of proportional control valves has provided opportunity for servo application of pneumatic drives. However, the possibility of application of pneumatic drives is limited in practice by the problems regarding to control these plants. The nonlinear effects of pneumatic systems caused by the phenomena associated with air compressibility, significant friction effects, wide range of air supply pressure, load variations, etc., make them difficult to control pneumatic axis. It is very challenging task to get an accurate mathematical model for describing pneumatic servo drives behavior. Classical approach to position control of pneumatic drives is based on the linearization of the nonlinear system dynamics around the desired set point, and such models are valid only for small deviations around an operating point [1, 2].

In this work a sliding mode controller (SLMC) for position control of a pneumatic servo drive has been applied. This control strategy is able to deal with unmodeled dynamics and bounded disturbances. It achieves robust control by using a discontinuous control signal when the system states are within predetermined sliding surface. The mayor drawback that prevents SLMC from being a universal solution to control problems is the necessity for an instantaneous switching device. Since the implementation of real components in practice is necessarily imperfect, and switching is not instantaneous, it causes a phenomenon known as chattering. Chattering involves high control activity and high frequency, finite amplitude oscillations of the control variable [3]. This phenomenon is mainly undesirable in practice due to negative effect on switching control components, and may cause a premature wearing out and damage to system components.

The proposed control strategy is verified by a computer simulation and experimentally on the laboratory model of the pneumatic rodless cylinder controlled by the proportional servo valve.

II. SLIDING MODE CONTROL DESIGN

In the automatic control research community, sliding mode control has been proposed as a robust control technique, able to mach the high performance of a control system in the presence of variable operating conditions or system nonlinearities. SLMC belong to variable structure control (VSC) algorithms, which can make a nonlinear system behave as a linear one, when the states of the system reach and track the sliding surface [4]. The systems with VSC structure are the form of nonlinear systems in which the control law, or control structure is qualitatively changed during the control process in order to obtain desired requirements. Such systems are characterized by a control structure which is switched as the system state crosses specified surfaces in the state-space, and sliding regime or sliding mode is the case when the system state is forced to move within the surfaces [3]. The gains of SLMC are constant but discontinuous and are switched about a sliding surface in the state space. The continuous form of SLMC is simple to design and practically does not require a model of the controlled process.

For pneumatic servo drive control the SLMC approach could be used to overcome the significant system nonlinearities during the control process. The pneumatic

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servo drive is considered as a single-input, single-output (SISO) system defined by the general state-space equation:

$$\dot{\mathbf{x}} = f(\mathbf{x}, t) + b(\mathbf{x}, t)u \tag{1}$$

where $\mathbf{x} = [x \; \dot{x} \; .... \; x^{(n-1)}]^T$ is the state vector and u is the input vector. The nonlinear function $f(\mathbf{x}, t)$ is in general not exactly known and its linearized form in an operating point is usually used. The expression $b(\mathbf{x}, t)$ is the control gain. The task of control process is to get the state vector $\mathbf{x}$ to track a desired state of system $\mathbf{x_R} = [x_R \; \dot{x}_R \; .... \; x_R^{(n-1)}]^T$ in spite of model imprecision. The negative tracking error vector is introduced in the following form:

$$\tilde{\mathbf{e}} = \mathbf{x} \text{-} \mathbf{x_R} = [\tilde{e} \; \dot{\tilde{e}} \; .... \; \tilde{e}^{(n-1)}]^T . \tag{2}$$

The time-varying sliding surface $\sigma(t)$ is a linear combination of the states such that for σ=0 the desired system behavior is achieved. The sliding surface is defined in the state-space R^n by the scalar equation, given by:

$$\sigma(\tilde{\mathbf{e}}, t) = \left(\frac{d}{dt} + \lambda \right)^{n-1} \tilde{e} = 0 \tag{3}$$

where n is the system order and λ is a strictly positive constant, that can be explained as a slope of the sliding surface in the phase plane.

A simplified linearized model, which still reflects essential characteristics of pneumatic drive controlled by proportional valve, assuming that the valve dynamics is negligible compared to the cylinder dynamics, can be presented with the following transfer function [5]:

$$G(s) = \frac{x(s)}{u(s)} = \frac{C_0}{s(\omega_0^{-2} s^2 + 2\zeta\omega_0^{-1} + 1)} \tag{4}$$

where:

$x(s)$	cylinder position,
$u(s)$	voltage input to the proportional valve,
ω_0	natural frequency of the cylinder,
ζ	damping ratio,
C_0	velocity gain.

The numerical values of system parameters are listed in Table I and are given in Appendix.

According to the equation (3) for the continuous-time transfer function, the sliding surface for n=3 is defined as follows:

$$\sigma = \frac{d^2}{dt^2}\tilde{e} + 2\lambda \frac{d}{dt}\tilde{e} + \lambda^2 \tilde{e} \tag{5}$$

$$\sigma = \lambda^2 \tilde{e} + 2\lambda \, \dot{\tilde{e}} + \ddot{\tilde{e}} \tag{6}$$

$$\sigma = c_1 \, \tilde{e} + c_2 \, \dot{\tilde{e}} + \ddot{\tilde{e}} \; . \tag{7}$$

Using the expression $\tilde{e} = x \text{-} x_R$ in equation (7) leads to:

$$\sigma = c_1 \, \tilde{e} + c_2 \, \dot{x} + \ddot{x} \tag{8}$$

$$\sigma = -c_1 \, e + c_2 \, \dot{x} + \ddot{x} \; . \tag{9}$$

The necessary condition for the existence of conventional sliding modes of system (1) is given by [3]:

$$\frac{1}{2}\frac{d}{dt}\sigma^2 < 0 \tag{10}$$

or

$$\sigma\dot{\sigma} < 0 \, . \tag{11}$$

The above inequalities have dynamical character because they include the information of the instantaneous state of the sliding surface $\sigma(t)$ as well as its time-derivative $\dot{\sigma}(t)$.

A corresponding block-diagram with SLMC strategy is shown in Fig.1.

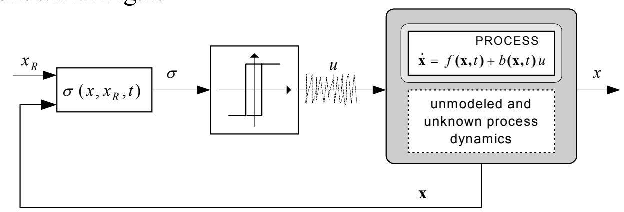

Fig. 1. Block-diagram of discontinuous system using SLMC strategy

The condition (11) is often not feasible in practice because the switching of real components is not instantaneous and this leads to an undesired phenomenon known as chattering in direction of sliding surface σ=0, Fig. 2.

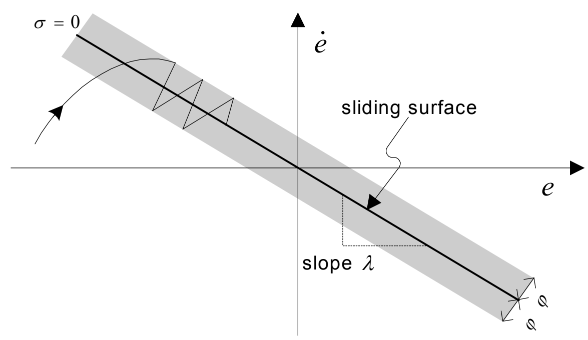

Fig. 2. Graphical interpretation of sliding region in phase plane with chattering as result of non-ideal switching

In order to eliminate the chattering, the condition (11) is expanded by a boundary layer in which the controller switching action is not demanded and thus the condition (11) can be rewritten as:

$$\sigma\dot{\sigma} \le -V\,|\,\sigma\,| \tag{12}$$

or

$$\dot{\sigma} \le -V \,\mathrm{sgn}\,(\sigma) \, . \tag{13}$$

where V is a strictly positive number, and it is the maximum voltage which is sent to the proportional valve (5 Volts in this work). Existing of the sign function in (13) indicates a discontinuous controller action. Depending on the state point location in the state-space, full voltage for forward or reverse motion is applied. In literature, there are several approaches to make the SLMC concept applicable in practice. In order to smooth out the discontinuous sign function a hyperbolic-tangent function or more often a saturation function is used instead of the sign function. This way the phase trajectory can oscillate within the sliding surfaces without switching controller action. From the previous consideration, the condition (13) can be written as follows:

$$\dot{\sigma} = -V \,\mathrm{sat}\,(\frac{\sigma}{\varphi}) \tag{14}$$

where the saturation function is defined as follows:

$$\mathrm{sat}\left(\frac{\sigma}{\varphi}\right)=\begin{cases}\mathrm{sgn}\left(\frac{\sigma}{\varphi}\right) & \text{when } |\sigma|>\varphi \rightarrow \begin{array}{ll} 1 & \text{for } \sigma>\varphi \\ -1 & \text{for } \sigma<-\varphi \end{array} \\ \frac{\sigma}{\varphi} & \text{when } |\sigma|\le\varphi \end{cases}$$

The control law, which ensured the existing of sliding motion in the state-space, is given by:

$$u=-V\,\mathrm{sat}\left(\frac{\sigma}{\varphi}\right) \tag{15}$$

where $\sigma=-c_1\,e+c_2\,\dot{x}+\ddot{x}$, according to equation (9).

It can be noted when the system enters within the sliding region, SLMC action becomes equivalent to a state feedback controller action (position-velocity-acceleration controller, PVA) of the form:

$$u=-K\,\mathbf{x} \tag{16}$$

with controller gains $K=V/\varphi\,[-c_1 \;\; c_2 \;\; 1]$, and the system state vector defined as $\mathbf{x}=[x_R-x \;\; \dot{x} \;\; \ddot{x}]$.

In practical realization of the control algorithm instead of acceleration feedback gain a pressure feedback gain is used. Namely, from a standpoint of practical application of control algorithm based on SLMC, directional measurement of acceleration is costly. An alternative procedure is double differentiation of measured cylinder position, which however results with a large noise level signal. As a compromise solution the measurement of pressure difference in cylinder chambers can be used [6]. The expression for cylinder acceleration yields from the expression for the balance of forces acting on the cylinder piston assuming that there are no external forces and neglecting the friction force:

$$\ddot{x}(t)=\frac{p_A(t)-p_B(t)}{m}A=\Delta p(t)\frac{A}{m} \tag{17}$$

This way the acceleration signal can be reconstructed from the measured signals of pressures, and this form is used in experimental study.

III. Description Of Experimental Equipment

A photo of the laboratory equipment is given in Fig. 3., while the schematic description of cylinder controlled by proportional control valve is illustrated in Fig. 4. The actuator is a rodless cylinder (*SMC* CDY1S15H-500) with stroke length of l=500 mm and diameter d=15 mm. Piston position is measured by the horizontal linear potentiometer (*FESTO* MLO-POT-500-TLF), which is attached to the actuator. The linear potentiometer has an advertised repeatability of ±0.25 mm and linearity 0.05 % of full scale. The directly actuated proportional control valve (*FESTO* MPYE-5 1/8 HF-010B), which is connected to both cylinder chambers, controls the linear motion of the piston. The bandwidth of the valve is about 100 Hz. Three pressure transducers (*SMC* ISE4-01-26) are added to measure cylinder pressures and the pressure of air supply.

1 - Linear potentiometer, 2 - Pneumatic rodless cylinder, 3 - Load mass, 4 - Rotational potentiometer, 5 - Ref. voltage on potentiometer, 6 - Pressure sensor, 7 - Proportional valve (servovalve), 8 - Proportional pressure valves, 9 - On/off solenoid valves, 10 - Filter-regulator unit, 11 - Air supply valve, 12 - Electronic interface, 13 – Control computer

Fig. 3. Photo of experimental model

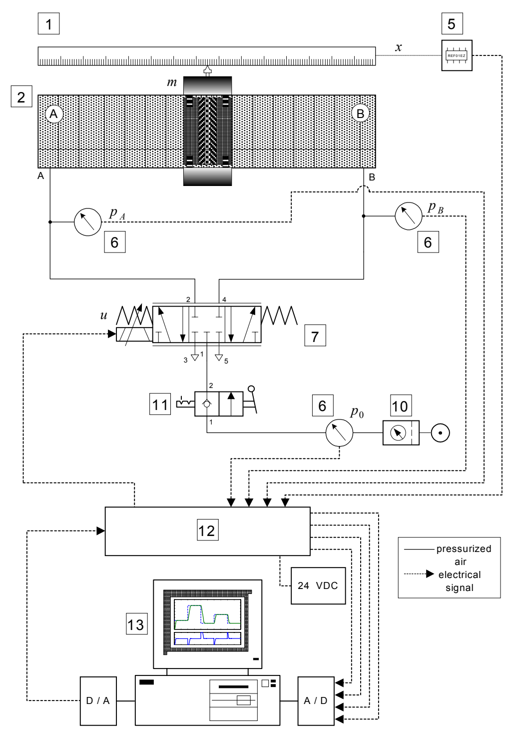

Fig. 4. Schematic diagram of control system

The feedback control algorithms are implemented on a Pentium-based PC using PCL-812PG data acquisition card. All signals from the process are sent to a microcomputer via a 12-bit A/D converter. The calculated control signals from the microcomputer are sent via 12-bit D/A converter to the proportional control valve. The control software is coded in “C” language. The experimental equipment also includes two proportional pressure valves (*SMC* VY1A00-M5) and two on/off solenoid valves (*SMC* EVT307-5D0-01F) [7], which are not used in this work.

IV. Simulation And Experimental Results

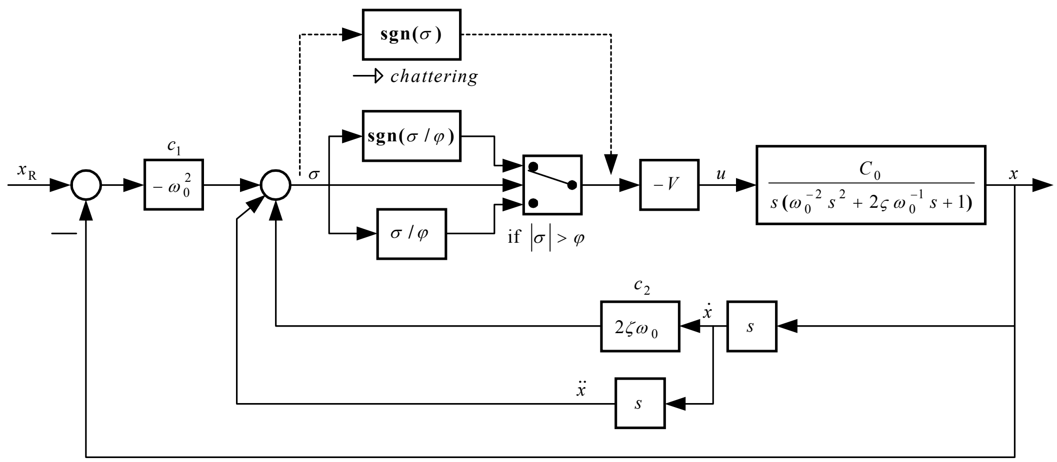

Fig. 5. Block-diagram of pneumatic servo drive position control using SLMC

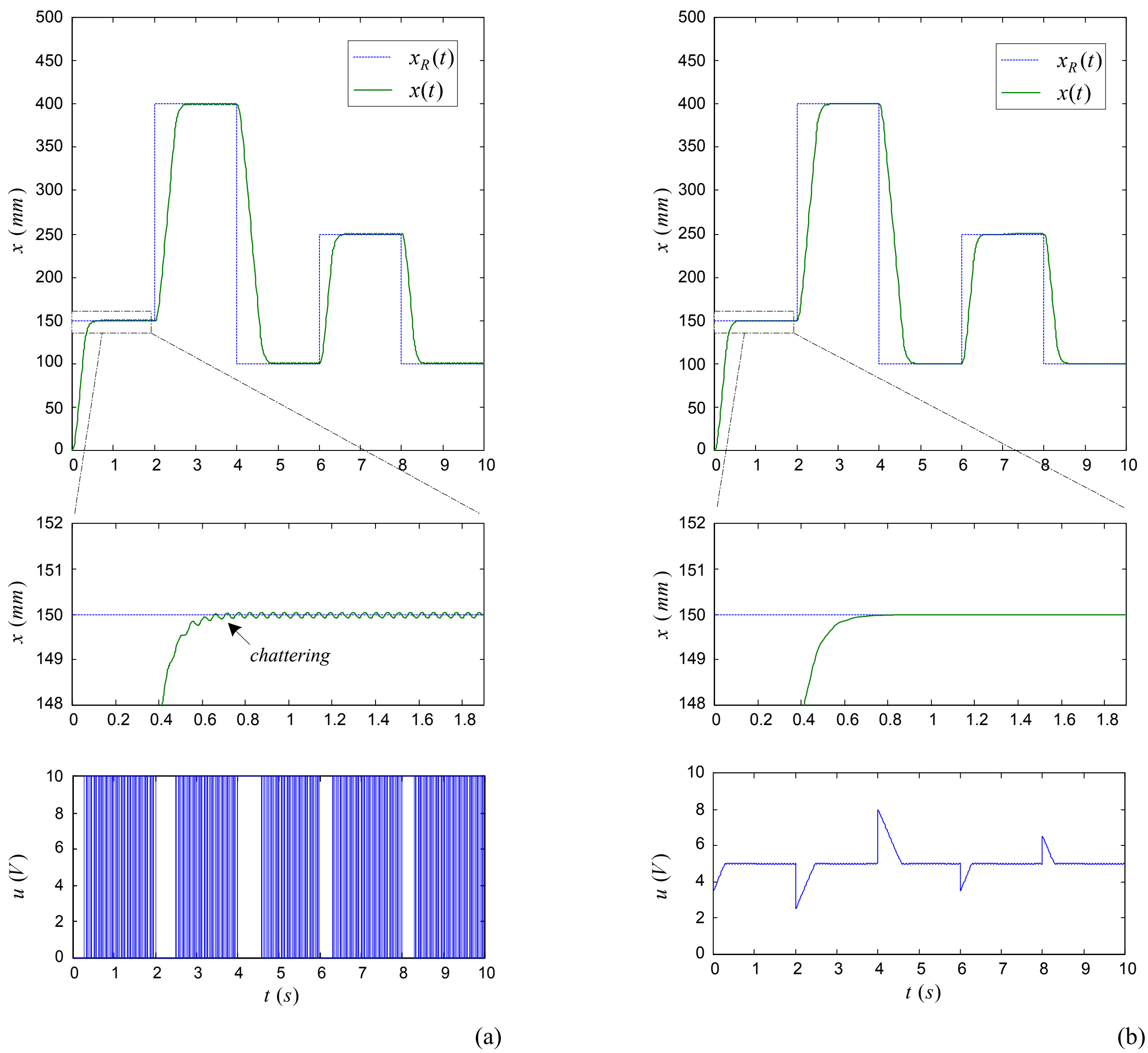

Fig. 6. Simulation results for cylinder position control using: (a) sign-function, (b) saturation-function

The simulation block-diagram of the pneumatic servo drive position control using SLMC is shown in Fig. 5. The controller coefficients are set to the values: $c_1=\omega_0^2=1087$, $c_2=2\zeta\omega_0=72$, according to practical recommendation given in [4], while maximum amplitude of the control signal is $V=5$. By changing the parameter φ, the boundary layer of the sliding mode region is moved, which has an influence on dynamic system behavior.

By introducing the boundary layer, steady-state error is relatively increased. Therefore, the trade-off between the smoothing of control input and positioning accuracy should be considered. By using the sign-function $\mathrm{sgn}(\sigma)$ in simulation process, the existing of chattering in control signal is obvious. This phenomenon has a negative effect on the controlled variable and the quality of control process is substantially deteriorated. However, by using a saturation function with switching logic it is possible to eliminate the high frequency oscillations of the control signal, Fig. 6.

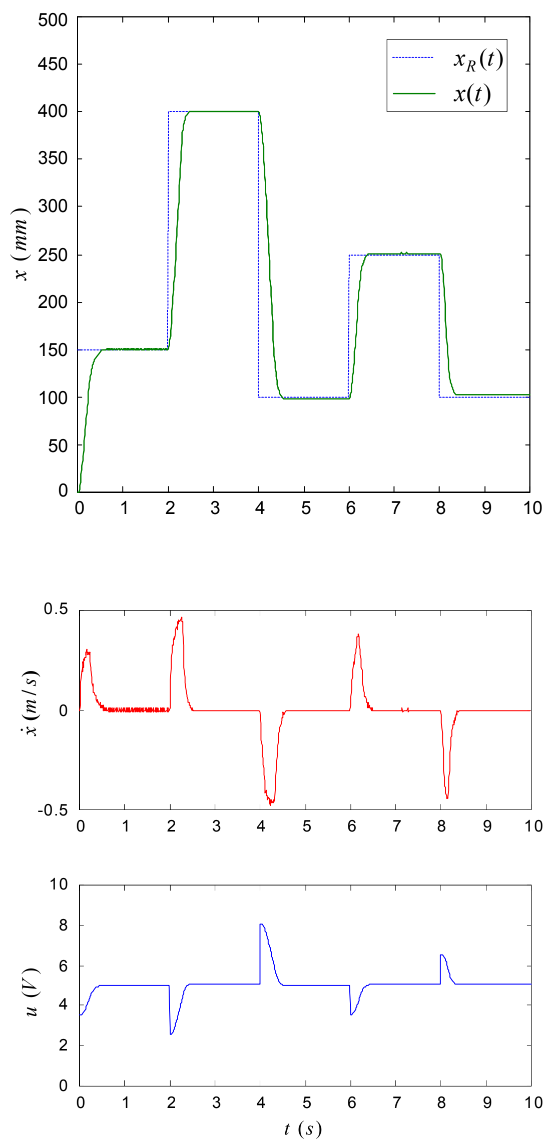

Fig. 7. Experimental results for cylinder position control

The effectiveness of the proposed SLMC is also verified by experiment on the real pneumatic servo actuator. The experimental results for position control of rodless cylinder on square-wave reference signal are illustrated in Fig. 7., and agreed with the simulation results. The velocity signals are obtained by differentiating the position measurements and instead of acceleration signals the pressure difference signals according to equation (17) are used. The controller sampling time of 10 ms was employed.

The parameters that are found to provide compromise between occurrence of chattering and acceptable accuracy are as follows: $c_1=1087$, $c_2=66$, $\varphi=11500$, $V=5$. The control signal has an acceptable behavior and signal amplitudes are far from the saturated values. The actuator was regulated smoothly to the desired positions. The choice of the boundary layer determined by the size of parameter φ and the time of discretization define the appearing of chattering and the steady-state error. It could be noted that there exist a design conflict between requirements for smoothing of control signals and for control accuracy. A large boundary layer is preferred for avoiding the chattering and a small boundary layer is preferred for better positioning accuracy.

V. Conclusion

A sliding-mode control approach has been applied to position control of a rodless cylindrical pneumatic actuator controlled by a proportional valve. The main difficulties in pneumatic servo actuator arise from significant nonlinearities such as a negative effect of friction and air compressibility and also variations of process parameters with time. The proposed SLMC is able to maintain the control system requirements with a reasonable accuracy. A linearized third-order dynamic model of the process, without the dynamics of the proportional valve, was used to perform a simulation study. Undesirable chattering in control signal, which implies extremely high control activity and may excite high-frequency dynamics neglected in modeling procedure is avoided by replacing the sign function by a smooth control interpolation using a saturation function. Finally, the developed controller is experimentally tested on a laboratory model of a servo pneumatic drive. The experimental results have shown that SLMC, which is based on a simple process model, can successfully control the motion of the drive despite the highly nonlinear nature of the controlled system. But, it should be emphasized that the positioning accuracy obtained in this work is still lower than the control structure, which includes friction compensation (steady-state error less than 1 mm) and structure which includes an integral action in a narrow region of low-speed near the operating point [7]. However, SLMC is simple to design, easy to implement, practically does not need an explicit model of the system, and also has a low sensitivity to disturbances. Due to its advantages it could be successfully implemented in less demanding industrial applications.

Appendix

The simulation and experimental study are performed using parameters listed below.

TABLE I
NUMERICAL VALUES OF SYSTEM PARAMETERS

Cross-sectional area	$A = 1.767 \cdot 10^{-4}\ m^2$
Volume of the cylinder	$V_c = 8.835 \cdot 10^{-5}\ m^3$
Supply pressure	$p = 5 \cdot 10^5\ Pa$
Initial load	$M = 0.91\ kg$
Friction coefficient	$k_f = 65\ Ns/m$
D/A converter gain	$K_{\mathrm{DA}} = 2.44 \cdot 10^{-3}\ V$
A/D converter gain	$K_{\mathrm{AD}} = 204.8\ 1/V$
Measuring system gain	$K_{\mathrm{m}} = 20\ V/m$
Velocity gain	$C_0 = 0.7\ m/Vs$
Natural frequency	$\omega_0 = 32.97\ rad/s$
Damping ratio	$\zeta = 1.1$
Sampling time	$T = 0.01\ s$

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