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ADVANTAGES OF DIGITAL SLIDING MODE POSITION CONTROL OF THE RODLESS PNEUMATIC CYLINDER

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Vladislav Blagojević[1], Miodrag Stojiljković[1], Boban Veselić[2]

[1]University of Niš, Faculty of Mechanical Engineering, Department of Production-information Technologies and Management, Aleksandra Medvedeva 14, 18000 Niš, Serbia
[2]University of Niš, Faculty of Electronic Engineering, Department of Automation Control, Aleksandra Medvedeva 14, 18000 Niš, Serbia
E-mail: vlada@masfak.ni.ac.rs, misa@masfak.ni.ac.rs, boban.veselic@elfak.ni.ac.rs

Abstract. *The paper deals with the task of achieving the desired position of the rodless pneumatic cylinder's piston and improving energy efficiency. The control system is designed by applying the variable structure system theory. The applied control algorithm is based on the digital sliding mode and has been compared to the conventional PI control law. The conducted experiments have demonstrated that this control algorithm provides a satisfactory positioning and great robustness, as well as energy efficiency.*

Key words: *servo pneumatics, PI, variable structure system, sliding mode, energy efficiency*

1. INTRODUCTION

Pneumatic actuators, such as rodless pneumatic cylinders, have the advantages of (a) low cost; (b) high power-to-weight ratio; (c) ease of maintenance; (d) cleanliness; (e) having a readily available and (f) cheap power source. Owing to numerous advantages, pneumatic actuators and systems are widely applied in industrial automation for the so-called sequential control. Nowadays, increasing demands with respect to accurate positioning, especially in robotics, have generated other control techniques. However, pneumatic actuators exhibit highly nonlinear characteristics due to the compressibility of air, the complexity of the friction presence and the nonlinearity of valves.

There are many papers dealing with the position control of the pneumatic actuators, including the work by Shearer [1, 2, 3], Mannetje [4], Ben-Dov and Salcudean [5], Wang et al. [6], Maeda et al. [7], Ning and Bone [8], Bobrow and McDonell [9], and Richer and Hurmuzlu [10, 11], among others.

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The paper discusses application of digital sliding mode control (SMC) algorithm [12] in positioning of a pneumatic cylinder [13,14,15]. To demonstrate the effectiveness of the chosen nonlinear control methodology, SMC has been compared to the performance of the conventional PI controller. Digital SM controller design has been carried out using a simplified linear model of the nonlinear plant, which was obtained by the experimental identification.

2. Pneumatic System Description

A common positional pneumatic servo system is presented in Fig. 1. It is composed of rodless pneumatic cylinder, designated as 1.0 and proportional valve 5/3 designated as 1.1, where M_L is the external mass, M_p is the piston mass, y is the position, F_L is the external load, P_1 and P_2 are the absolute pressures in the cylinder's chambers, A is the piston surfaces, V_1 and V_2 are the volumes of cylinder's chambers and i_c and u_c are the input DC current and voltage of proportional valve [16].

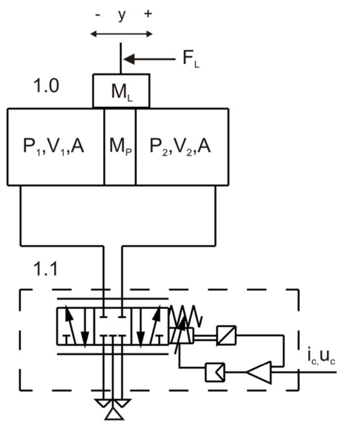

Fig. 1. Pneumatic scheme of the servo system for positioning

The third order linearized mathematical model of the presented system can be represented in the state space as:

$$\dot{\mathbf{x}} = \mathbf{A}\mathbf{x} + \mathbf{b}u_c$$

$$\mathbf{A} = \begin{bmatrix} 0 & 1 & 0 \\ 0 & 0 & 1 \\ 0 & -\dfrac{2\kappa A^2 P_i}{(M_L + M_P)V_i} & -\dfrac{c_f}{(M_L + M_P)} \end{bmatrix} \tag{1a}$$

$$\mathbf{b} = \begin{bmatrix} 0 \\ 0 \\ \dfrac{2\kappa R T_S C_1 A}{(M_L + M_P)V_i} \end{bmatrix} \tag{1b}$$

where c_f is the viscous dampening, R is the universal gas constant, Ts is the supply gas temperature and κ is the specific heat ratio. The state space variables are defined by the vector $\mathbf{x} = [\delta y \quad \delta\dot{y} \quad \delta\ddot{y}]^{\mathrm{T}}$. It holds $P_i=P_1=P_2$ and $V_i=V_1=V_2$ for the midpoint of the rodless cylinder stroke as initial position.

3. Digital SMC Algorithm

For the position control of a pneumatic cylinder, a discrete-time chattering-free SMC algorithm [12] has been employed, for a number of advantages [13,14,15]:

- theoretical invariance to external loads and system parameter variations and uncertainties, under so-called matching conditions [17];
- system dynamics is predefined by appropriate selection of the sliding manifold;
- dynamics in SM is only defined by the parameters of the sliding manifold;
- exact knowledge of plant parameters is not required, only the range of their change;
- system stability is easier to achieve through the decomposition into two simpler sub-systems;
- the order of the differential equation describing SM dynamics is reduced by the sliding manifold dimension.

Apart from these exceptional features of variable structure control systems (VSCS) operating in SM, there are also several shortcomings, such as:

- measurability of all state space coordinates is necessary, and
- occurrence of high-frequency oscillations, i.e. chattering, due to the excitation of unmodelled dynamics by finite switching frequency of the control signal.

The first shortcoming has been successfully solved by application of state observers, which at the same time significantly alleviate the other shortcoming. However, there are several SMC techniques that eliminate chattering occurrence. One of them is digital SMC algorithm [12], which has been employed in this paper. It contains two control phases: nonlinear and linear.

The non-linear control mode drives the system trajectory from an arbitrary initial point to the predefined vicinity of the sliding plane. Afterwards, the linear control ensures one-step reaching and subsequent sliding along the sliding manifold towards the origin.

Since the control will be digitally implemented, it is necessary to perform time-discretization of the model (1). If T_d denotes the sampling period and

$$\mathbf{A}_{\boldsymbol{\delta}}(T_d) = \frac{e^{\mathbf{A}T_d} - \mathbf{I_n}}{T_d},\ \mathbf{b}_{\boldsymbol{\delta}}(T_d) = \frac{1}{T_d}\int_0^{T_d} e^{\mathbf{A}\tau}\mathbf{b}\,d\tau \tag{2}$$

the discrete-time state-space model of the nominal system, by using δ-transform, can be expressed in the form:

$$\boldsymbol{\delta}\mathbf{x}(kT_d) = \mathbf{A}_{\boldsymbol{\delta}}(T_d)\mathbf{x}(kT_d) + \mathbf{b}_{\boldsymbol{\delta}}(T_d)\mathbf{u}(kT_d) \tag{3}$$

The commonly used linear scalar switching function s(x) in SMC systems that defines a sliding hyperplane in the state space (s=0), along which the SM is organized, is chosen as:

$$s = \mathbf{c}_{\boldsymbol{\delta}}(T_d)\mathbf{x} \tag{4}$$

where $\mathbf{c}_{\boldsymbol{\delta}}(T_d)$ is the switching function vector of appropriate dimension.

The detailed control design procedure of the selected discrete-time SMC (DSMC) algorithm is given in [12]. Here, only the important relations will be briefly recounted. The positioning control law is obtained as:

$$u_c = -\mathbf{c}_{\boldsymbol{\delta}}(T_d)\mathbf{A}_{\boldsymbol{\delta}}(T_d)\mathbf{x}(k) - \mathbf{\Phi}(s(k), \mathbf{X}(k)) \tag{5}$$

The switching function vector $\mathbf{c}_\delta(T_d)$, which exclusively defines system dynamics in the SM, is selected according to the relation:

$$\mathbf{c}_{\boldsymbol{\delta}}(T_d) = [\mathbf{c_1}(T_d) \mid 1]\mathbf{P}_1^{-1}(T_d) \tag{6}$$

The values required for calculating matrix $\mathbf{c}_\delta(T_d)$, are obtained through the following equations [12]:

$$c_i(T_d) = \frac{1}{(i-1)!} \frac{d^{i-1}\prod_{j=1}^{n-1}(\delta - \delta_j(T_d))}{d\delta^{i-1}}\Big|_{\delta=0} \tag{7}$$

$$\delta_i(T_d) = \frac{e^{-\alpha_i T_d} - 1}{T_d}, \ \alpha_i > o, \ \begin{matrix} i \neq j \Rightarrow \alpha_i \neq \alpha_j, \\ i, j = 1, \ldots, n-1 \end{matrix} \tag{8}$$

$$\mathbf{P}_1^{-1}(T_d) = [b_\delta(T_d) \quad \ldots \quad A_\delta^{n-1}(T_d)b_\delta(T_d)] \begin{bmatrix} a_1(T_d) & \cdots & a_{n-1}(T_d) & 1 \\ a_2(T_d) & \cdots & 1 & 0 \\ \vdots & \ddots & \vdots & \vdots \\ 1 & \cdots & 0 & 0 \end{bmatrix} \tag{9}$$

Function $\Phi(s,X)$ is defined as:

$$\mathbf{\Phi}(s, X) = \Phi(s) = \min\left(\frac{|s|}{T_d}, \sigma + \rho|s|\right)\mathrm{sgn}(s); \ \begin{matrix} 0 \leq \rho T_d < 1, \\ \sigma > 0 \end{matrix} \tag{10}$$

Parameters σ and ρ define SM reaching dynamics and should be chosen to provide as short as possible reaching phase.

4. EXPERIMENTAL RESULTS

The experimental platform [15], where the properties of the proposed DSMC algorithm have been verified, contains the pneumatic components as in Fig. 1, compressed air flow rate sensor, position sensor, and ED2000 acquisition and control card that is hosted by a PC. The applied rodless pneumatic cylinder is DGPL-18-150-PPV-A-B-KF-SH manufactured by FESTO company. Proportional valve 5/3 bears the designation MPYE-5-1/8-LF-010-B, whereas the by-pass valve is MC-2-1/8 manufactured by FESTO. The flow rate sensor is FESTO SFE1-LF-F200-WQ8-P2I-M12 with measuring range from 0 to 200 l/min. The HBM position sensor has a range ±200 mm. ED2000 operates with the sampling time $T_d = 0.005$ s.

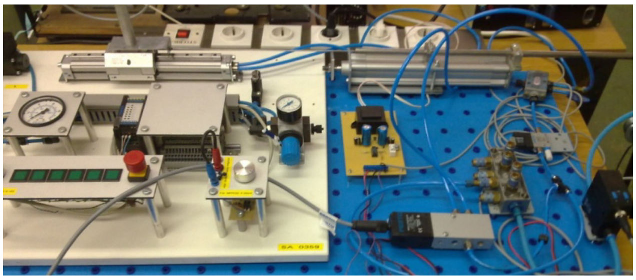

Fig. 2. Experimental set-up of the servo pneumatic system with rodless pneumatic cylinder

It is very difficult to use linear model (1) in control system design, because it has some unknown parameters, whose determination is a problem. Also, in case of the third order system, as it is one described by (1), SMC requires measurement of position, velocity and acceleration. In the experimental investigation only the position has been measured, whereas the velocity has been estimated. Estimation of the acceleration $\ddot{y}$ would also be a problem. Therefore, the second order model has been used in controller design, which has been acquired according to the experimental identification.

Step responses of the cylinder piston position of pneumatic servo system, for supply pressure 600 kPa, and at different input DC voltages u_{cv}, are presented in Fig. 3.

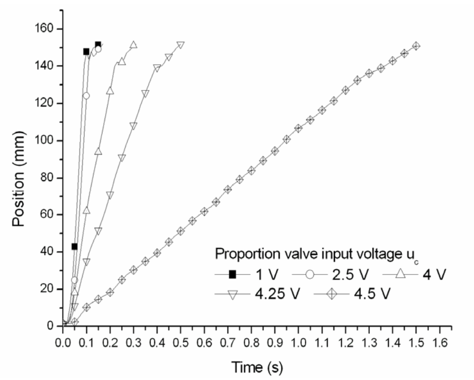

Fig. 3. Response of rodless cylinder piston position

The experimentally identified discrete-time model of the plant [15], is given by:

$$\delta y(kT_d) = \begin{bmatrix} 1 & 0.00464 \\ 0 & 0.86167 \end{bmatrix} y(kT_d) + \begin{bmatrix} -0.222 \\ -86.6645 \end{bmatrix} u_c(kT_d) \tag{11}$$

The controller parameters have been selected as: $\sigma = 2.5$, $\rho = 0$, $c_\delta = [-0.000562 \quad -0.0000562]$.

The first test was performed without load force and when the cylinder piston was moving from y = 50 mm to the desired y =100 mm, after which it was retracted into the initial position y = 50 mm again. After achieving the positions along the stroke length, the cylinder piston should hold the position during 1 s.

In Fig. 4, the results of positioning of the cylinder piston when DSMC and traditional PI controller with gain $K_P = 0.009$ and $K_I = 0.00007$ are used, are shown. The supply pressure was 600 kPa.

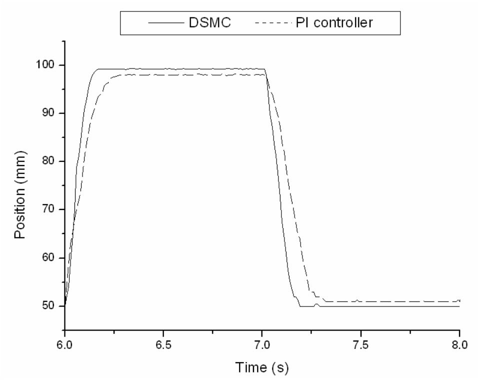

Fig. 4. Diagram of the rodless cylinder piston position without load force

The second test has been performed with load force F = 50 N in the same direction as position cylinder piston increase and for the same conditions and durations as in the first measurements.

In Fig. 5, the results of positioning of the cylinder piston when DSMC and traditional PI controller with the same gain as previous, are shown.

In the first test, without load force, the position error is nearly equal to zero when DSMC is used. When PI controller is used, for the same conditions, position error is up to 1.5 mm. The DSMC is quicker in reaching the desired position than PI controller.

In the second test, with 50 N load force, the position error is the same as without load force and still nearly equal to zero when DSMC is used. When PI controller is used, for the same conditions, position error is up to about 6 mm.

The results show that DSMC of the rodless cylinder exhibits superior performance in comparison to the case when conventional PI controller is used. The DSMC approach successfully achieves the desired positions of the rodless cylinder piston, even with load force. For the same conditions, traditional PI controller shows much bigger positioning errors.

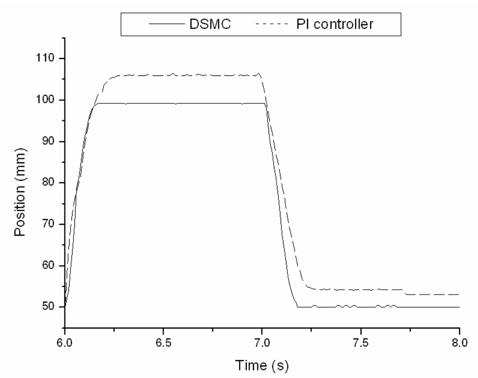

Fig. 5. Diagram of the rodless cylinder piston position with 50 N load force

The compressed air consumption of the rodless pneumatic cylinder, with and without load force, when DSMC and conventional PI controller are used [15], is shown in Fig. 6.

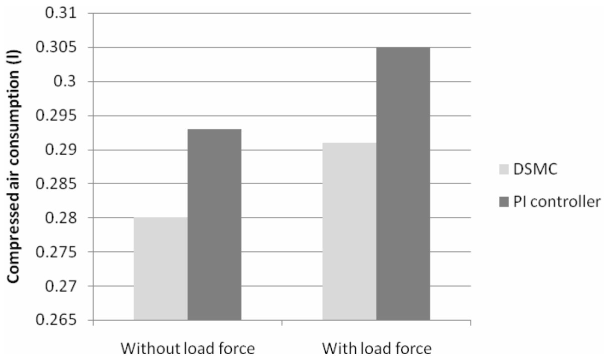

Fig. 6. Diagram of the rodless pneumatic cylinder compressed air consumption

This result shows that the DSMC uses about 5% less compressed air in comparison to the case when conventional PI controller is used.

4. Conclusion

The paper discusses the positioning problem of the rodless pneumatic cylinder and compressed air consumption as well. DSMC algorithm is employed in solving these problems. The designed DSM controller performs its task successfully, since it ensures robustness, quick reaching of the desired position without overshoot and oscillation and it uses less energy. The advantages of this nonlinear type of control are shown in Chapter IV, where its performance is compared to the PI controller. The position error when DSMC is used is less than in the case of the traditional PI controller, for the same conditions, regardless of the load force. Also, the compressed air consumption when DSMC is used is about 5% lower than with PI controller.

It may be concluded that application of DSMC strategy in positioning of pneumatic cylinders is justified due to much better response, greater robustness and energy efficiency.

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PREDNOSTI POZICIONIRANJA PNEUMATSKOG CILINDRA BEZ KLIPNJAČE DIGITALNIM KLIZNIM REŽIMOM

Vladislav Blagojević, Miodrag Stojiljković, Boban Veselić

Rad se bavi problemima postizanja željene pozicije klipa pneumatskog cilindra bez klipnjače i povećanju energetske efikasnosti. Upravljački sistem je projektovan primenom teorije sistema promenljive strukture. Primenjeni algoritam upravljanja baziran je na digitalnom kliznom režimu i upoređen je sa konvencionalnim PI zakonom upravljanja. Sprovedeni eksperimenti demonstrirali su da predloženi algoritam upravljanja daje zadovoljavajuću tačnost pozicioniranja i veliku robustnost jednako dobro kao i energetsku efikasnost.

Ključne reči: *servo pneumatika, PI, sistem promenljive strukture, klizni režim, energetska efikasnost*