NOVEL CYLINDER POSITIONING SYSTEM REALISED BY USING SOLENOID VALVES

L. Földi, Z. Béres, E. Sárközi

Institute for Mechanical Engineering Technology, Faculty of Mechanical Engineering
Szent István University, Gödöllő, Hungary, www.geti.gek.szie.hu

Abstract This paper presents a novel control design, developed to realise fast and accurate position control of a pneumatic actuator using inexpensive on/off solenoid valves. In contrast to conventional control methods, the proposed control method operates chatter free, based on air compression. The control principle was developed by investigating the dynamics of a pneumatic actuator with an identified mathematical model. This new approach is applied to a pneumatic double acting cylinder, controlled by a pair of 5/3 way directional single solenoid valves. The described closed-loop circuit copes with the discontinuities associated with the valve's switching dynamics, and relatively long response time. The experimental apparatus uses an analogue displacement encoder for metering the piston's position and velocity, and doesn't incorporate pressure sensors thus ensuring a low cost system design. The results of experiments with various step responses show that the proposed control method performs well. The measured steady-state position errors are equal to the used potentiometer's travel resolution, which is 0,01 mm. Therefore this novel control and the related pneumatic system design could be a cost effective alternative to the servo-pneumatic positioning systems.

Keywords: Position Control, Solenoid Valve, Pneumatic Cylinder, Control Method, Actuator

1. INTRODUCTION

In order to achieve linear motion, pneumatic, electromagnetic and hydraulic actuators are typically used. Due to their advantageous characteristics in position control applications the latter two are more widespread. Though using hydraulic actuators high velocity and great force is achievable, and their position control can also be relatively easily solved, a handicap is that the leaking of hydraulic fluids might contaminate the workpiece. Electromagnetic actuators on the other hand are clean and reliable in their operation but often require a mechanical transmission, both to convert high speed and low torque to a more useful combination and to convert rotary motion to linear motion. While linear motors overcome the need for transmission, they can be expensive [9].

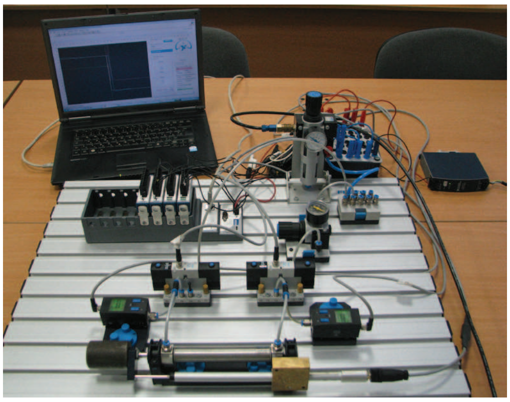

Figure 1. The experiment apparatus

Pneumatic actuators have several advantages: they are fast, cheap, have an outstanding power-to-weight ratio, are easily maintainable and they don't contaminate the work piece. The challenge to the use of pneumatic drives is that due to piston friction and the characteristics of compressed gas flow their behaviour is non-linear. As a result their industrial use is only widespread in applications which require linear motions between end positions.

In the last decade such industrial controllers became available which have adequate computing capacity for real-time usage. Thus there is now opportunity to develop pneumatic systems which don't require costly proportional valves for positioning and hence the usage of the more cost-efficient solenoid valves became possible (Figure 1.). These simple on/off valves are cheap and easily maintainable. Their drawback however is that as yet the required control method is not appropriately elaborate: the solutions offered in the scarce publications on the topic consist of applying conventional control methods (PID, sliding mode, fuzzy logic) or their hybrid versions, with varying success.

A review of the papers on the topic is given in Table 1, where the applied control methods, the number of required valves and the highest positioning accuracy is shown. Even though it is not indicated in the table, in some cases certain speed or force decreasing solutions (eg. throttle valve, reduced pressure) were used, which naturally may have an effect on the operating range of the positioning system and the steady-state error. The employed valve's switching time is also omitted from the table.

Number in References	*Authors*	*Control method*	*Number of required valves*	*Positioning accuracy*
[9]	Thomas, M.B.; Maul, G.P.; Jayawiyanto, E.	Modified PD + PWM control	3	±0,1 mm
[1]	Ahn, K.; Yokota, S.	MPWM + neural network (LVQNN)	8	±0,2 mm
[7]	Parnichkun, M.; Ngaecharoenkul, C.	hybrid of fuzzy and PID + PWM control	1	±3,5 mm
[6]	Nguyen, T.; Leavitt, J.; Jabbari, F.; Bobrow, J.E.	Sliding mode control	4	±0,1 mm
[8]	Shih, M.-C.; Ma, M.-A.	Fuzzy + modified differential PWM control	2 + Proportional pressure valve	±0,075 mm
[5]	Messina, A.; Giannoccaro, N.I.; Gentile, A.	Individual control + PWM	2	±0,1 mm
[3]	Barth, E.J.; Zhang, J.; Goldfarb M.	Individual linear continuous + PWM	2	N/A
[2]	Akdağ, F.N.; Kuzucu, A.	Sliding mode control	2	±0,05 mm
[10]	van Varseveld, R.B.; Bone, G.M.	PID with friction compensation + PWM	2	±0,21 mm

Table 1. A review of the papers on the topic

2. MATHEMATICAL AND SIMULATION MODEL

We carried out the design work of our proposed control strategy based on an identified simulation model [4] of the actual apparatus. In order to develop this system we applied equations number (1) and (2); using these we can express the mass flow filling the chambers of the cylinder as follows:

$$\dot{m}_1 = A_1 \cdot p_{\sup} \cdot \sqrt{\frac{2}{R \cdot T}} \cdot \psi_{(p_1/p_{\sup})} \tag{1}$$

$$\dot{m}_2 = A_2 \cdot p_2 \cdot \sqrt{\frac{2}{R \cdot T}} \cdot \psi_{(p_{atm}/p_2)} \tag{2}$$

The outflow functions (ψ) in the equations are one of the major causes of the nonlinear behaviour of the pistons. Knowing the mass flows, the pressures in the cylinder's chambers can be expressed through the gas law's derivative with respect to time:

$$\dot{p}_1 = \frac{1}{V_1} \cdot \left(\dot{m}_1 \cdot R \cdot T - p_1 \cdot \dot{V}_1 \right) \tag{3}$$

$$\dot{p}_2 = \frac{1}{V_2} \cdot \left(p_2 \cdot \dot{V}_2 - \dot{m}_2 \cdot R \cdot T \right) \tag{4}$$

Using the pressures and knowing the physical parameters of the piston the mechanical equation complemented with the mass loads is as follows. Again, a term causing non-linearity appears: friction (Ff).

$$m \cdot \ddot{x} = p_1 \cdot A_1 - p_2 \cdot A_2 - \left| F_f \right| \cdot sign(\dot{x}) - m \cdot g \cdot \sin\alpha - j \cdot (\dot{x})^2 \cdot sign(\dot{x}) \tag{5}$$

The last term in the equation is the pipe friction loss which, according to our observations is a significant factor in practice but is ignored in the specialised literature we made use of. This friction loss was taken into consideration with a friction factor (j) multiplied by the square of velocity.

Using the presented equations we were able to construct the proposed control system's simulation model in Matlab/Simulink software. After completing the identification process of the model's parameters, we were able to effectively analyse some of the conventional control methods. As a result we decided to elaborate a novel control strategy, which copes with the discontinuities associated with the valve's switching dynamics, and relatively long response time.

3. CONTROL METHOD

Designing the control method we determined to apply two solenoid valves (one for each chamber), the biggest benefit of which is that the number of control signals is thus raised to the second power.

Number of variation	*Signal of solenoid valve 1*	*Signal of solenoid valve 2*	*Movement of the cylinder piston*
1	fill	empty	positive direction, fast
2	fill	close	positive direction, slow
3	fill	fill	immobile
4	close	empty	positive direction, uncontrolled
5	close	close	first immobile, then uncontrolled
6	close	fill	negative direction, slow
7	empty	empty	accidental direction, uncontrolled
8	empty	close	negative direction, uncontrolled
9	empty	fill	negative direction, fast

Table 2. Available variations using two solenoid valves

Out of the variations presented in Table 2, we ignored the uncontrolled ones (where pressure in both chambers changes in an uncontrolled way) which thus leaves us with five useful solenoid valve variations. However, in order to stop the movement of the cylinder piston in variation number 3, a complementary element had to be introduced (due to the asymmetric construction of the cylinder we used). Using a pressure regulator, the positive chamber was filled with supply pressure decreased in proportion to the surfaces of the piston. As a result a balance of force was reached based on equation number (5).

As a first step we started out from a graph representation, where time and position were placed on the axes. This way the most important control parameter, the error (difference between a measured process variable and a desired set-point) can be visualized easily if we represent both the ideal and the actual position values (reference signal and measured output respectively) plotted against the elapsed time.

Based on the scale of the error we might form predictions as to which valve variation has to be realized by the control method in order to decrease the absolute value of the error.

Number of control signal	*Relation between reference signal and measured output*	*Desired piston movement*	*Number of variation in Table 2*
1	$r >> y_m$	positive direction, fast	1
2	$r > y_m$	positive direction, slow	2
3	$r \approx y_m$	immobile	3
4	$r < y_m$	negative direction, slow	6
5	$r << y_m$	negative direction, fast	9

Table 3. Realized valve variations

The variations in Table 3 can be easily indicated in the graph representation we have already created (Figure 2.), since control signal number 3 has to take effect within the close vicinity of the ideal position, namely in a predefined tolerance band where further changing of position is unnecessary. Nearby, above and below this band we can assign the zones where the piston is relatively close to the ideal position, thus slow movement is required (control signals number 2 and 4). Outside these bands those parts of the stroke length are situated which are relatively far from the ideal position, hence a fast piston movement is needed in order for a fast decrease in the absolute value of the error to take place (control signals number 1 and 5).

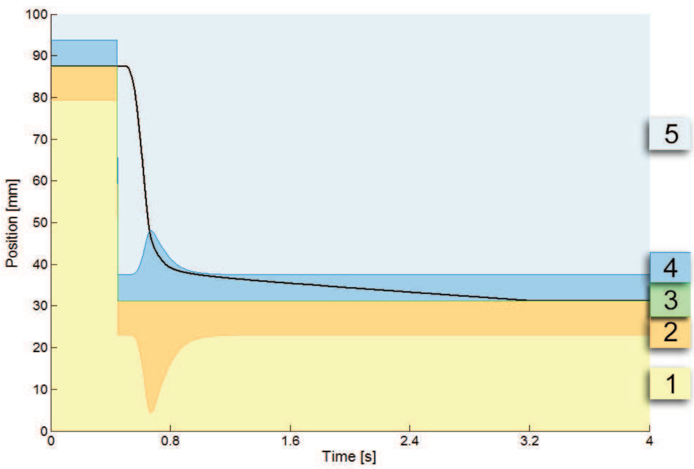

Figure 2. Control Strategy

Since the width of the tolerance band is a parameter given during utilization, the control method only has to determine the width of bands number 2 and 4 in a given moment. By rendering the respective valve variations to the given bands, the solenoid valves can be controlled harmoniously and in an essentially discrete way, with few valve switches, i.e. chatter free.

In order to determine the width of the bands we can set up the following equation based on the work-energy theorem:

$$0-\frac{1}{2}\cdot m\cdot v^{2}=-(\frac{p_{b}\cdot A\cdot(q+z)}{R\cdot T}\cdot R\cdot T\cdot\ln\frac{p_{a}}{p_{b}}) \tag{6}$$

Of the terms of the equation pa and pb are the initial and final pressures of the closed chamber, q is the distance needed to stop the piston while z is the remaining distance. Setting up the equation we regarded the process to be isothermal, thus we overestimated the necessary distance to stop the piston, this way compensating for the losses which appear in the positioning system.

After reducing the values regarded as constants and expressing q from the equation, we come to the following relation:

$$q=c_{2}\cdot v^{2}-z \tag{7}$$

The equation shows that the ideal deceleration distance of the piston (q) changes according to a second-degree function of piston speed. It follows that the widths of the deceleration bands of the piston have to be adjusted in a given moment proportional to the square of the measured output's first derivative; at this point a proportional coefficient (c2) has to be introduced. Furthermore, it is also necessary to set a constant c1 bandwidth which guarantees the slow movement near the ideal value.

As the next step, we have to assign the adequate control signs to the bands. This is done in a similar way to the operation of a relay, by sign functions, which decide whether the piston's measured output is below or above the band limit set. Summing up the values of these sign functions we arrive at 4 discrete numeric values which unequivocally determine which band the measured output falls into (e.g. we get -3 if it is below all band limits, i.e. in band number 1). By assigning the adequate control signs to these numeric values and to the tolerance band around the ideal position we achieve a control method which satisfies the criteria set.

In accordance with notations of control engineering, the mathematical description of the control method is as follows: r stands for the reference signal, while ym is the measured output, t means tolerance and F;C;E are respectively the filling, closing and emptying switch states of the first and second solenoid valves:

$$sign\left\{y_{m}-\left[r-(c_{1}+c_{2}\cdot(\dot{y}_{m})^{2})\right]\right\}+sign\,(y_{m}-r)+sign\left\{y_{m}-\left[r+(c_{3}+c_{4}\cdot(\dot{y}_{m})^{2})\right]\right\}=k$$

$$\begin{array}{ll} if\ |y_{m}-r|<t,\ then & \Rightarrow \quad cont=[F;F] \\ if\ |y_{m}-r|>t,\ then & \begin{cases} if\ k=-3 \Rightarrow cont=[F;E] \\ if\ k=-1 \Rightarrow cont=[F;C] \\ if\ k=1 \Rightarrow cont=[C;F] \\ if\ k=3 \Rightarrow cont=[E;F] \end{cases} \end{array} \tag{8}$$

The constants c3 and c4 figuring in the equation have the same function as the already known c1 and c2 constants; their introduction into the equation is necessary because of the asymmetric setting options which reflect the asymmetry of the cylinder.

The behaviour of the system is calculable and compared to the examples in the specialized literature the command signal operates at a lower frequency; thus, due to the smaller amount of gas let into the environment the efficiency of the overall system is increased. It is worth noting that the system is capable of adjusting itself to the reference signal even if per chance we have chosen too high parameters at the control settings. In this case the balance of force required to stop the cylinder piston sets in earlier than necessary, but since we only fill one of the chambers (see bands number 2 and 4) in the other chamber the pressure will slowly decrease, which is a consequence of the ever-present cylinder loss. The force arising from this pressure difference will always slowly move the piston to its ideal position. This is advantageous because this way, as far as the positioning systems most significant quality factor, the steady-state error is concerned, we can say that our system operates without predefined working-positions. Another great advantage is that in spite of the low frequency and delays of the solenoid valves we are able to make the piston stop with a high accuracy thanks to the slow movement around the reference signal.

4. APPARATUS

The circuit diagram of the pneumatic positioning system is presented on figure number 5. As an actuator we applied a Festo DSNU-20-100-PPV-A P606 cylinder of 100mm stroke length, to which we attached a Festo MLO-POT-225-LWG analogue displacement encoder, which has a 0,01 mm travel resolution. The applied encoder is a potentiometer which provides a voltage signal in proportion to the displacement. In order to move the cylinder we applied two Festo VSVA-B-P53C-H-A2-1R2L 5/3 way solenoid valves, but we only used one output connection each and the remaining output ports were plugged. We measured the mentioned solenoid valve's ON and OFF switching time at 6 bar supply pressure; in case of switching on it was 14 ms (Figure 3.) while at switching off 36 ms (Figure 4.).

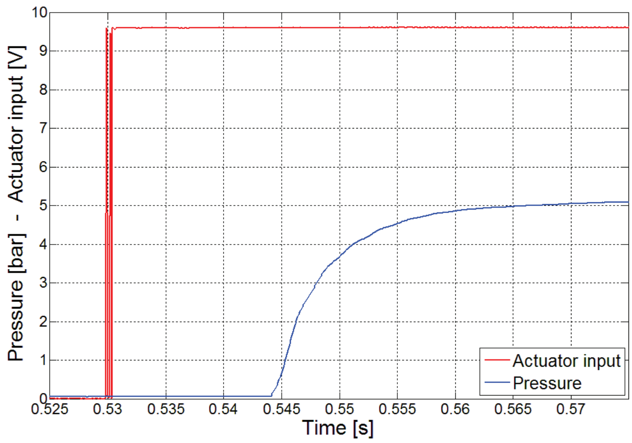

Figure 3. Valve switching – On

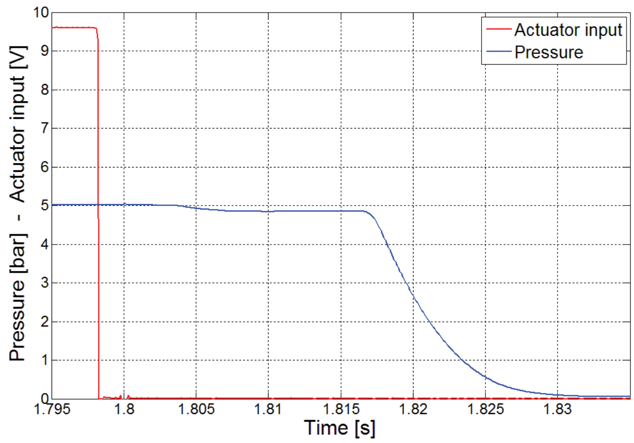

Figure 4. Valve switching - Off

A further constituent is a Festo D:LR-1/8-0-MINI pressure regulator, and we also connected a Festo SDE1-D10-G2-H18-C-PU-M8 pressure sensor to both chambers to serve as feedback, which however we did not use in the control process in order to minimize the number of sensors necessary for the operation of the system. This system was constructed to test the planned control method but by changing the different elements of the scheme, it can be freely scaled to achieve a faster operation or the movement of heavier loads.

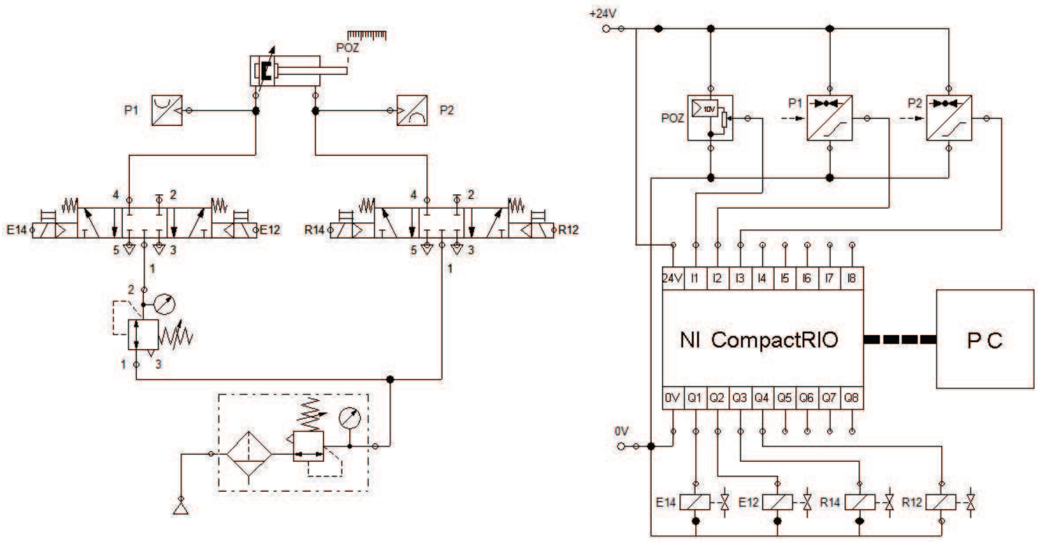

Figure 5. The circuit diagram of the pneumatic positioning system

The major elements of the electronic system are a 0-24 V direct current power supply (NI PS-15), an electronic instrument board (Festo), a NI CompactRIO™ (cRIO 9073) programmable automation controller and the already mentioned electro-pneumatic elements (displacement encoder, pressure sensors and solenoid valves). The applied NI CompactRIO™ programmable automation controller is a modular system; out of its modules we used the analogue-to-digital converter (NI 9201), for a dual purpose. On the one hand we applied it in the controlling process to measure the voltage signal (which is in proportion to the displacement) provided by the displacement encoder. On the other hand we used it in collecting data about the voltage signals corresponding to pressure values (expressed in bars) provided by the analogue pressure sensors. We controlled the two solenoid valves with the help of the digital output module (NI 9472). The communication between the CompactRIO™ and the computer was ensured by an Ethernet connection. We realised the real-time control based on equation number (8) by applying the FPGA module of CompactRIO™ programming it in the LabVIEW 2009 software.

5. MEASUREMENT RESULTS

The testing of the compiled system was done by determining the quality factors of the control method. During this process we have determined the settling time for step responses from end positions, overshoots and steady-state error graphically based on measurement results. Under settling time we conventionally understand the time required for the measured output to finally reach the $\pm$5% vicinity of the reference signal.

The moved load was m=0.542 kg, the value of supply pressure in the case of the negative chamber was p2=6·105 Pa, and accordingly the decreased supply pressure of the positive chamber was $p_1 = \frac{A_2}{A_1} \cdot p_2 = 5{,}04 \cdot 10^5\ Pa$. The measurements were carried out at room temperature.

The control setting parameters were c1=18 [mm], c2=950 [-], c3=18 [mm], c4=250 [-]; based on previous experiences with the system the width of the tolerance band was set to be $\pm$0.025 mm, we regarded the position as adequate within this range.

First we tested the dislocations of the cylinder piston in the positive direction, i.e. when the piston was moving outwards of the cylinder.

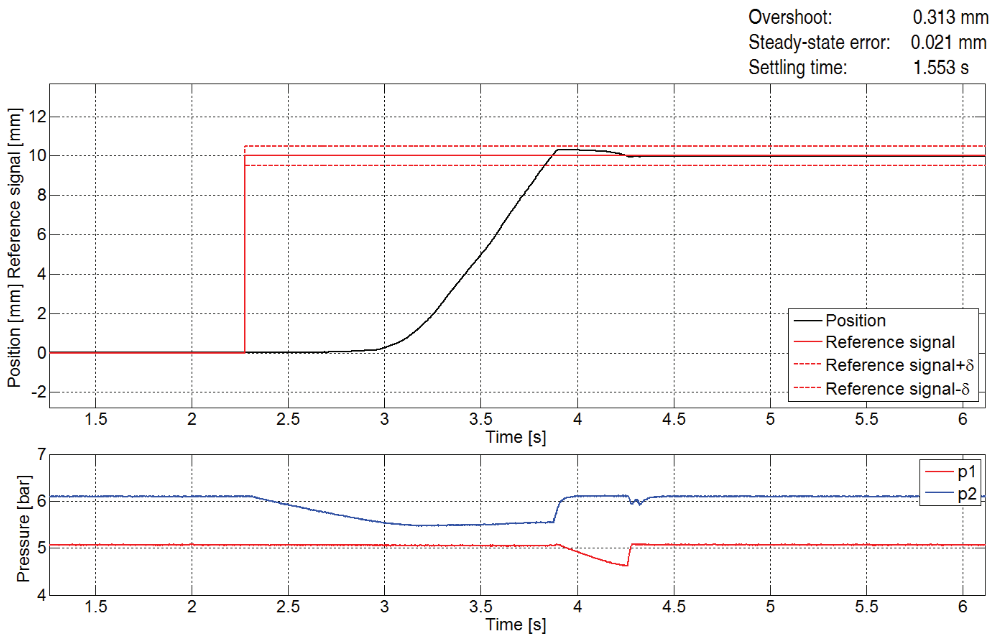

Figure 6. Step response - 0-10 mm

In Figure 6. we examined a settling onto the 10 mm reference signal from initial position, which is 10% of the stroke length of the cylinder. At such a small change of position the control system works in the deceleration zone (2) all the way through, i.e. the speed of the displacement of the piston is only dependent upon the cylinder loss. It is visible that the settling time is under 2 seconds, the overshoot can be considered minor. The steady-state error is greater than the resolution of the displacement encoder, but it nevertheless remained within the tolerance band and is still better than most of the best values to be found in the specialised literature.

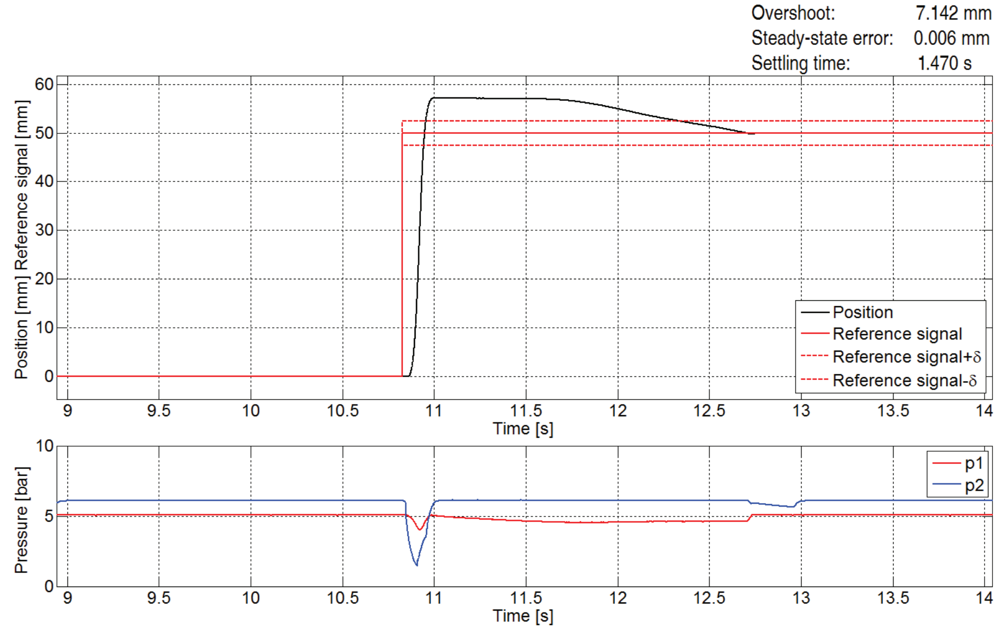

Figure 7. Step response - 0-50 mm

The evaluation of the second test (Figure 7.) sheds light on the limits of our positioning system, given that in the case of the settling to the 50 mm reference signal a significant overshoot was observed. This is the consequence of the fact that the chamber filling at the adjusted supply pressure is fast and because of this the acceleration of the low-friction cylinder piston is too great to be followed by the control system. This is the consequence of the operating limits of valves with long switching time. The deceleration of the piston movement, i.e. the closing of the ventilated chamber begins later than would be ideal, which is why the significant overshoot and the same settling time appear. It is important to be noted though, that all these do not influence the positioning accuracy, i.e. the steady-state error is equivalent to the travel resolution of the displacement encoder.

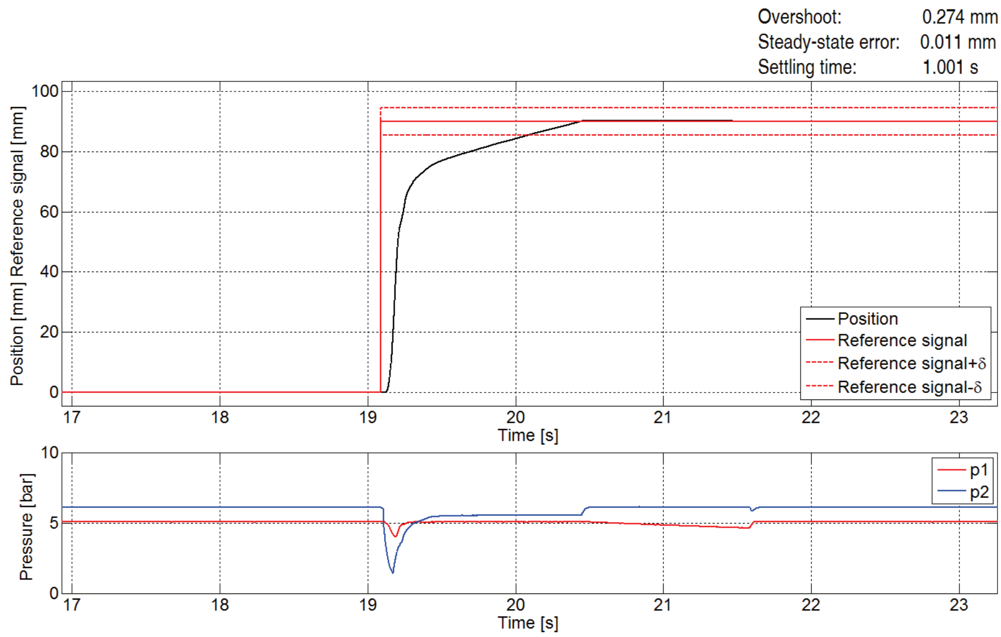

Figure 8. Step response - 0-90 mm

At the third measurement (Figure 8.) we have examined a displacement which is long compared to the stroke length of the cylinder by setting the reference signal at 90 mm. It is visible that in the case of large step size the control method is able to follow the dynamics of the cylinder, the overshoot is minimal while the settling time is 1 second. The steady-state error is again equivalent to the travel resolution of the displacement encoder.

After this we examined the movement of the piston in the negative direction, namely when it moves backwards into the cylinder.

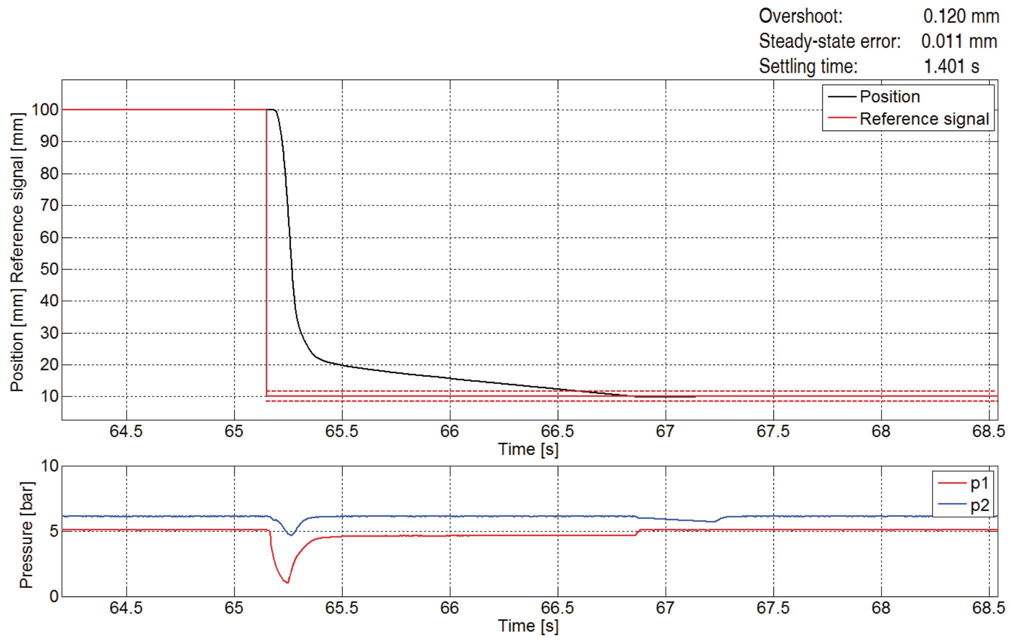

Figure 9. Step response - 100-10 mm

In the course of the fourth test at Figure 9. we can see a displacement similarly great to the previous experiment, but in the negative direction. The overshoot is minimal, and the settling time is still under 2 seconds. The steady-state error is once again equivalent to the resolution of the displacement encoder.

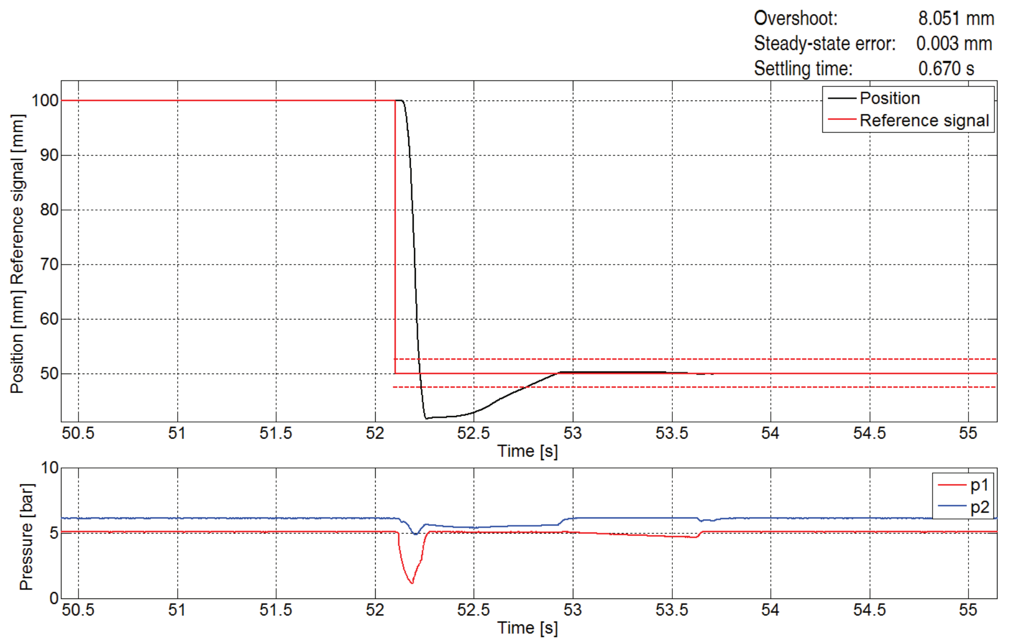

Figure 10. Step response - 100-50 mm

The fifth measurement (Figure 10.) examined a movement in the negative direction the scale of which is 50% of the stroke length of the piston. It is visible that an overshoot similar to that of the second testing appears. The steady-state error is minor, and in this case the settling time is exceptionally good, which is partly due to the fact that the piston moves faster in the negative direction because of the asymmetric construction of the cylinder.

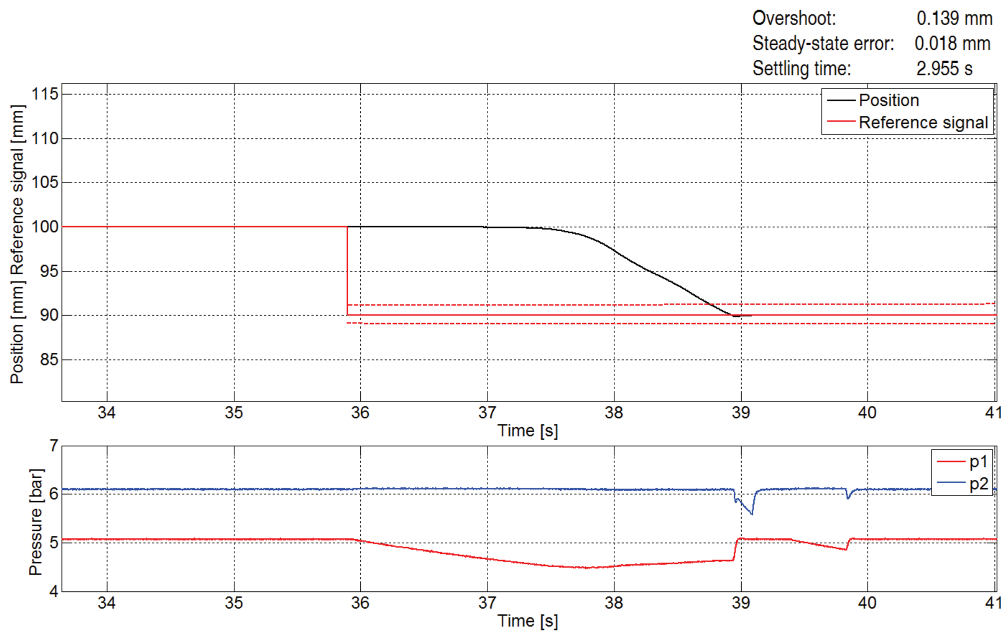

Figure 11. Step response - 100-90 mm

In the last, sixth testing (Figure 11.) we observed the small-scale negative movement with a displacement from 100 mm to 90 mm. The movement of the piston remains in the deceleration band (4) all the way through, thus the speed of displacement is only dependent upon the cylinder loss. As a result, the settling time is significant, but the overshoot is minimal and the steady-state error is almost equivalent to the travel resolution of the displacement encoder.

6. SUMMARY

A novel control strategy and the according experimental apparatus to achieve accurate positioning of a pneumatic cylinder using solenoid valves is presented. The most significant features of this pneumatic positioning system are the following:

- *it substitutes the costly proportional valve with the conventional solenoid valve*
- *thanks to a novel control design, the system operates in a chatter free way*
- *the maximal operating velocity and force of the applied pneumatic actuator is not decreased*
- *it contains the least amount of sensors and the least expensive electro-pneumatic elements possible*

Thanks to all these, the system can achieve an adequately high positioning accuracy and reach a favourable price/value ratio at the same time. The paper also sheds light on the fact that the system's steady-sate error is highly dependent on the displacement encoder's travel resolution. This holds out the promise that the application of more advanced technology in the area (e.g. using digital sensor with higher travel resolution, or a fast-switching solenoid valve to reduce the overshoot) will further improve the system's positioning accuracy.

7. REFERENCES

[1] Ahn, K., Yokota, S., Intelligent switching control of pneumatic actuator using on/off solenoid valves, Mechatronics,15, 683–702, 2005.

[2] Akdağ, F.N., Kuzucu, A., Highly accurate pneumatic position control, Istanbul Technical University Mechanical Engineering Department, http://digital.ni.com/

[3] Barth, E.J., Zhang, J., Goldfarb, M., Control Design for Relative Stability in a PWM-Controlled Pneumatic System, Journal of Dynamic Systems, Measurement, and Control, 125, 504-508, 2003.

[4] Isermann, R., Mechatronic systems : Fundamentals, Springer, London, 2005.

[5] Messina, A., Giannoccaro, N.I., Gentile, A., Experimenting and modelling the dynamics of pneumatic actuators controlled by the pulse width modulation (PWM) technique, Mechatronics, 15, 859–881, 2005.

[6] Nguyen, T., Leavitt, J., Jabbari, F., Bobrow, J.E., Accurate Sliding-Mode Control of Pneumatic Systems Using Low-Cost Solenoid Valves, IEEE/ASME Transactions on mechatronics, 12(2), 216-219, 2007.

[7] Parnichkun, M., Ngaecharoenkul, C., Kinematics control of a pneumatic system by hybrid fuzzy PID, Mechatronics, 11, 1001-1023, 2001.

[8] Shih, M.-C., Ma, M.-A., Position control of a pneumatic cylinder using fuzzy PWM control method, Mechatronics, 8, 241-253, 1998.

[9] Thomas, M.B., Maul, G.P., Jayawiyanto, E., A Novel, Low-Cost Pneumatic Positioning System, Journal of Manufacturing Systems, 24(4), 377-387, 2005.

[10] van Varseveld, R.B., Bone, G.M., Accurate position control of a pneumatic actuator using on/off solenoid valves, IEEE/ASME Transactions on Mechatronics, 2(3), 195-204, 1997.