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Ph.D. THESIS

**IMPROVING POSITIONING ACCURACY OF
SERVOPNEUMATIC SYSTEM WITH DSP-BASED SLIDING
MODE CONTROL**

Submitted by

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INTRODUCTION, THE AIM OF THE RESEARCH

There are several methods to realize linear motions. Due to the fact that linear actuator owns the advantage of low cost, it has been widely applied. Actuators can be classified as follows: hydraulic-, pneumatic- and electromechanical actuators. All types have advantages both and disadvantages. Nowadays, pneumatic actuators are 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 and inherent compliance. Additional advantages of pneumatic actuators are the lack of fire and explosion hazard, the ease in maintenance and the reliable functioning. Traditionally, pneumatics are used to move a payload between two fixed endpoints. Due to intense research in modelling servopneumatic systems, novel pneumatic solutions have been developed over the course of the past twenty years. This was also greatly contributed by the paralelly developed novel principles and practices of system controls. These factors resulted in greatly improved positioning and tracking performances and the pneumatic actuators became suitable for precision applications in robotics. The main disadvantage of pneumatic servo systems is the inherent nonlinearities. The nonlinearity comes from the nonlinear air flow-pressure relationship through the variable area orifice of a servo valve, the compressibility of the air and from the nonlinear friction in the sliding contacts of the piston along the cylinder walls. Because of difficulties due to the high nonlinear nature of pneumatic systems, the widely used PID can no longer provide satisfactory results by today's standards. Despite the mentioned improvements, the inherent nonlinearities in pneumatic systems still pose a great challenge for design and control engineers.

Review of the relevant literature reveals a great variety of solutions in pneumatic positioning ranging from the traditional to the newest controlling techniques. Common in all that only a few made it to practical applications due to their overly complicated structure, relatively weak functional properties and great computational and design demands.

Based on preliminary works in our Department, I concluded to choose to improve the usability and precisicy of pneumatic motion systems using the the sliding mode control which technique is reported to achieve as high accuracy as 0.01 mm.

Further aims of this study were to widen industrial applications of pneumatic positioning using a simple and robust drive mechanism in combination with state of the art controlling techniques that may be suitable for application in industrial robotics or in other practical use.

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1. BACKGROUND OF THE RESEARCH

A good background of the pneumatic servo systems research can be found in [20]. The foundation works on modelling and control of servopneumatics were conducted as early as in the 1950's, Shearer (1956) Blackburn et al. (1960). Studies on modeling of the pneumatic system can be found in: Sanville (1971) presented a mathematically simple and practical valve air flow model. Further investigation in this topic can be seen in Anderson (1985), and McCloy and Martin (1980). Backe and Ohligschlaeger (1989) investigated and gave a more exact description of the relationship between pressure, volume, mass and temperature in the pneumatic cylinder chamber. The dynamic model of a pneumatic actuator is the foundation of motion control design. There are quite a lot of works dedicated of this issue: Araki et al. (1993), Hahn and Piepenbrink (1994). The research on pneumatic servo systems is almost a reflection of modern control engineering. In some of the earliest works linear PID controller was employed: Shearer (1956), Burrows and Web (1966), and Vaughan (1965). Linear model based approach can be seen in works of Lai et al. (1990), Harada et al. (1988), Liu and Burrows (1988), and Yin and Araki (1998). This methods, only gave acceptable results in small range of operation area. It was later improved by dividing the operational range into several individually linear sections selecting a number of operating points which cover the range of the system operation. A number of works are dedicated to the PID auto-tuning approach. Position error in the work of Fok and Ong (1999) is experimentally found to be less than $\pm 0,3$ mm. In Fujiwara et al. (1995) and Matsukuma et al. (1997) a neural network based self-tuning PID controller was presented. In Jeon et al. (1998), genetic algorithms were applied to optimize the control parameters (position, velocity and acceleration feedback gains). For a rodless cylinder, the positioning error is seen to be less than $\pm 0,1$ mm. Wang et al. (1999) proposed a kind of PID controller learning modification method. Position error is reported to be less than ± 1 mm.

Quite many examples of adaptive control application in servo pneumatics can be found in Wikander (1988), Miyata (1989), Bobrow and Jabbari (1991), Oyama et al. (1990), McDonell and Bobrow (1993), Tanaka et al. (1994), Li et al. (1997) and Soong et al. (1997). With nonlinearities compensation Wikander's work (1988) show 0,01 mm cylinder positioning accuracy.

In trying to achieve satisfactory control performance, Tanaka et al. (1998) presented a technique of model reference adaptive control with neural network for an electro-pneumatic servo system. The control result was $\pm 0,08$ mm. In Kosaki and Sano (1998) a disturbance observer was used to compensate the effects of external load disturbances. The controller gains were adjusted by fuzzy inference.

Another typology of control recently investigated for pneumatic positioning servo systems is sliding mode control. Numerous research works trying to apply the sliding mode approach in servopneumatics: Noritsugu and Wada (1989) [1], Tang and Walker (1995) [2], Pandian et al. (1997) [3][4], Hamerlain (1995) [5], Bouri et al. (1996), Surgenor and Vaughan (1997) [6], Paul et al. (1994) [7], Song and Ishida (1997) [8][9] and Drakunov et al. (1997). Drakunov et al. [10] demonstrated that

the sliding mode control can be successfully used to compensate the cylinder viscous and Coulomb friction. Static positioning error is shown to be less than $\pm 0,2$ mm. Fuzzy control application examples in servo pneumatics can be seen in Matsui et al. (1990), Lu (1993), Araki et al. (1997), Shih and Hwang (1996) and Wang et al. (1996).

Katsumata et al. (1996), Gross and Rattan (1997) used a multi-layer neural network. Norgaard et al. (1996) and Sorensen et al. (1999) presented a predictive control. In Kimura et al. (1996) a H_{∞} control approach was applied. Feedback linearization is one of the basic nonlinear control techniques. Examples of applying feedback linearization can be found in Bobrow and McDonell (1998), Kawamura et al. (1989), Bouhal et al. (1993) and Kimura et al. (1995). Shu Ning and Gray M. Bone (2005) perform a comparative study between two control methods (the first is position plus velocity plus acceleration feedback combined with feedforward and deadzone compensation, the second is sliding mode control) [19].

There are additional control methods. Kobayashi et al. (1995) dynamic impedance matched robust control. Hamdan and Gao (2000) demonstrated a PID + feedforward + bang-bang + antiwindup approach. Matrukuma et al. (1997) published a nonlinear PID control scheme. Wang et al. (1998) introduced a deterministic full nonlinear states feedback control design method. Nakano et al. (1993) accomplished a positioning accuracy of $2\mu\text{m}$ by employing an active piezoelectric locking mechanism. Force controlled pneumatic servo systems can be found in many applications, for examples, in robotics (Lin és Burrows (1988)). Noritsugu and Takaiwa (1995) indicated that pressure control is essential in improvement of pneumatic positioning control performance. Using pressure inner control loop is investigated in many works, like in Ben-Dov and Salcudean (1995), Wikander, Xiang (1996) and Hamiti et al. (1996).

The highest precision which can be found in special literature is 0,01 mm, but it isn't assured in all position of piston and at several pressure value.

2. METHODS

The following requirements for the investigation were applied:

- the investigation was designed with computer simulation and with physical model;
- the system should be built on equipments and softwares, exist in our department;
- based on previous experimental results I have improved the computer simulation and physical model.

The objective of this work is to increase the accuracy of pneumatic positioning systems beyond the previously published best value of 0,01 mm. It was also important to reduce overshoot to less than 0,5 mm, and to lessen or possibly eliminate spring action overall. These parameters were studied using various piston position and cylinder orientation as well as using varying mass and spring loading.

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Robustness of the usual control methods was tested using cylinders of various sizes. Particular efforts were made to establish the minimum movement that safely can be achieved in pneumatic cylinders.

The following control methods were used:

- linear PID control;
- nonlinear PID control;
- adaptive control;
- intelligent systems (fuzzy systems, neural networks, genetic algorithms)
- sliding mode control.

I designed and constructed the experimental device. The aim of the research is to construct a simple and robust pneumatic servo-system what is suitable for research grade experiments and related measurements. The following requirements for the system have been met:

- the system should be built basing on industry equipment;
- the system should fulfil all requirements imposed by the research program;
- the system should be universal and easy to use;
- except testing of the pneumatic positioning system, the system should be used as a laboratory subject for students.

Among the different control methods I found the sliding mode control the most appropriate in all aspects. Therefore the further experiments are focused on improving the sliding mode control and developing new methods. In the beginning of the research work I had to make an important decision:

The usage of the experimental device seemed to be so simple and flexible, that hereafter I privileged the measurements carried out on the physical model rather than the computer simulation.

I proved experimentally, that the precision and stability, that our aim was, cannot be reached with the classical sliding-mode control. The development of new hardware and software methods was needed. During of development, the most important point of view was the high operating velocity.

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 theses are connected to these three main steps of the design of the sliding mode control.

2.1. Design new hardware and software equipments

In our investigation we have used the „eZdsp™ for TMS320LF2407” DSP target board from Spectrum Digital. The DSP Starter Kit (DSK) can connect to host PC via parallel port for program development program download and debugging.

Differing from the rest of the devices, the applied DSP operated at 3,3 V.

The design included an extra interface board for flexibility to fit I/O ports and to support both of the two main types of position encoder and providing two analogue outputs (AD420) for the servo-valves and serial communication link to a host computer (Fig. 1., Fig. 2.). In this application, the second board can be plugged. DSP has analog input and digital output and input interface on it. During the experiment to control the proportional valve an analog output was needed and an AD420 type D/A converter was selected for this purpose. The system had an extra analog output to allow the system widening. The digital input signals were converted to analog voltages and fed to the DSP input through an optical isolator. For fast programmed valve actuations it was imperative to implement as fast digital outputs as possible. This was achieved using a ULN 2064 type Darlington driver.

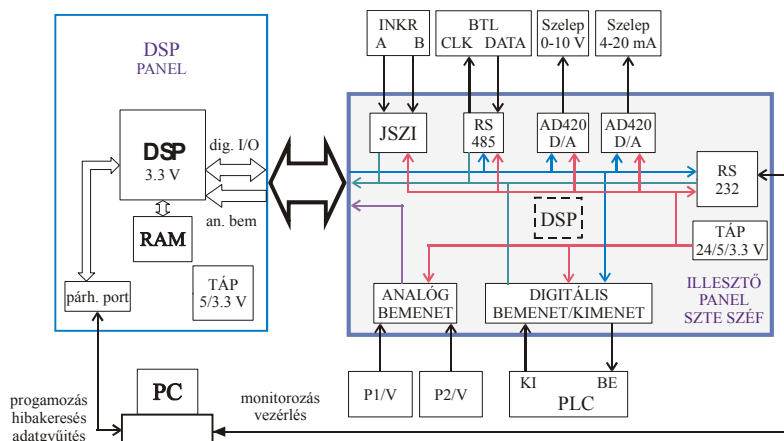


Fig. 1. Block diagram of the DSP target board and interface board

During the construction of interface board we must design the communication to the host PC and to the industrial equipments. Communications to host PC via RS 232 port were designed with ICL3232IBN integral circuit.

Position controlling requires very fast encoding and data relaying. For position sensing with 0,01 mm resolution we used a LINIMIK MSA 320 type incremental encoder. When needed, a Balluff BTL-S101 absolute value transducer was applied to reach resolution as high as 0,001 mm. The 5 V output level signals of the incremental encoder were converted to 3,3 V level for DSP.

The Balluff transducer relayed data via the RS 485/422 interface. A MAX488 type IC was applied to interfacing the output and input of the DSP and the RS-485 serial ports. One digital output and one input of the DSP are used by the software program for synchronic serial communication. The output

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is fed to the BTL-S101 encoder and the input receives signals from it. The panel of the DSP interface board was designed for later being able to operate as a stand-alone positioning unit (Figs. 1 and 2).

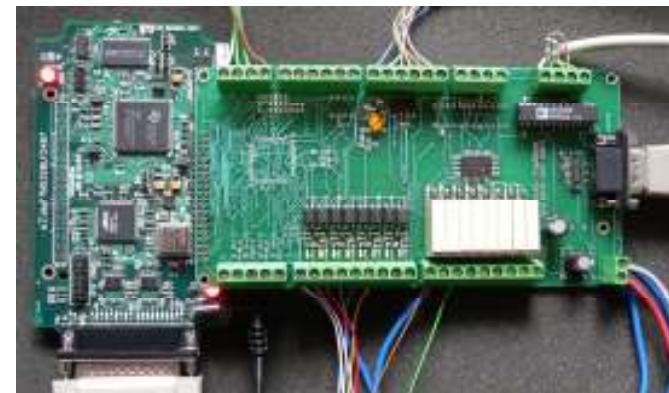


Fig. 2. Top view of DSP target board and interface board



Fig. 3. The controller operating window

Serial port of the interface circuit was used to implement communication between the DSP and the personal computer and to monitor the actual position. The actual position was displayed on the screen (Fig. 3).

The control algorithm was written in “C” language, using the Spectrum Digital Code Composer. Accuracy of the positioning depends mostly on the speed of the controlling and executing systems. In our case, the sampling time was set to 2 ms. During this period of time, the encoding and reading of position data, running of controlling algorithm and saving all parameters must be completed. The control software program was stored in the program memory of the DSP allowing it to operate independently. If needed in case of complicated tasks position controlling and system regulation could

be separated. In this situation, positioning is performed by the DSP whereas system controlling can be done using a PLC. So the required position data is generated by the PLC and transmitted to the DSP and the latter one executes the motion, and, finally reports the actual position back to the PLC.

2.2. The fundamentals

To design the model of the cylinder we must investigate the air mass flow rate through an ideal nozzle. Also, filling model of a chamber should be made and studied. To accomplish these goals, the following simplifications were made:

- the flow is considered one-dimensional;
- air can be considered as an ideal gas and the kinetic energy of the air is neglected

Four variables are encountered in treating flow:

velocity (w);

pressure (p);

temperature (T);

density (ρ), respectively specific volume ($v = \frac{1}{\rho}$).

Four independent relations needed to solve for these variables:

- Ideal gas equation of state $p \cdot v = R \cdot T$; (3.1)

- Balance of the input, output and inner energies $c_p \cdot T + \frac{w^2}{2} = \text{const}$; (3.2)

- Euler- equation $w \cdot dw + \frac{dp}{\rho} = 0$; (3.3)

- Balance of the input, output and inner masses $A \cdot w \cdot \rho = \text{const}$. (3.4)

From these fundamental equations, Bernoulli-equation for compressible, ideal frictionless gases can be derived and they similarly applicable to an air mass flow rate through an ideal nozzle and to flow rate from a chamber through a pipe. The 0 subscript refers to parameters in the chamber.

The equation of motion for the piston is as follows:

$$M \cdot \ddot{x} = p_a(u) \cdot A_a - p_b(u) \cdot A_b - d \cdot \dot{x} - k \cdot x - F_f \quad (3.5)$$

For pressure build-up equations, geometrical parameters (A_a and A_b), the load factors (M , k , d and E_f) and pressure values of p_a and p_b are required.

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At the end of the completion of my PhD paper I got to know Ming-Chang Shih, professor of the National Cheng kung University of Taiwan, who affirmed my assumption that the precision of the positioning is limited primarily by the resolution of the transducers. Shih's work shows 20 nm positioning accuracy [15, 16]. My aim is to get hold of a 4µm resolution of incremental encoder from Heidenhein and a subdivision and counter electronics subdivide the sinusoidal input signals up to 1024-fod. Thus, it becomes possible to perform measurements in the nanometric range.

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$$\dot{p}_a = \frac{1}{V_a} [R \cdot T \cdot \dot{m}_{abe} - p_a \cdot \dot{V}_a] = \frac{1}{V_a} [R \cdot T \cdot \dot{m}_{abe} - p_a \cdot A_a \cdot \dot{x}] \quad (3.6)$$

$$\dot{p}_b = \frac{1}{V_b} [-R \cdot T \cdot \dot{m}_{bki} - p_b \cdot \dot{V}_b] = \frac{1}{V_b} [-R \cdot T \cdot \dot{m}_{bki} + p_b \cdot A_b \cdot \dot{x}] \quad (3.7)$$

Substituting (3.6) and (3.7):

$$M \cdot \ddot{x} = \frac{1}{V_a} [R \cdot T \cdot \dot{m}_{abe} - p_a \cdot A_a \cdot \dot{x}] \cdot A_a - \frac{1}{V_b} [-R \cdot T \cdot \dot{m}_{bki} + p_b \cdot A_b \cdot \dot{x}] \cdot A_b - d \cdot \ddot{x} - k \cdot \dot{x} \quad (3.8)$$

$$\ddot{x} = \frac{A_a}{M \cdot V_a} R \cdot T \cdot \dot{m}_{abe} - \frac{A_a}{M \cdot V_a} p_a \cdot A_a \cdot \dot{x} + \frac{A_b}{M \cdot V_b} R \cdot T \cdot \dot{m}_{bki} - \frac{A_b}{M \cdot V_b} p_b \cdot A_b \cdot \dot{x} - \frac{d}{M} \ddot{x} - \frac{k}{M} \cdot \dot{x} \quad (3.9)$$

Substituting $A_a \cdot x + V_{a0}$ and $A_b \cdot (L - x) + V_{b0}$

$$\ddot{x} = \frac{A_a}{M \cdot (A_a \cdot x + V_{a0})} R \cdot T \cdot \dot{m}_{abe} - \frac{A_a}{M \cdot (A_a \cdot x + V_{a0})} p_a \cdot A_a \cdot \dot{x} + \frac{A_b}{M \cdot (A_b \cdot (L - x) + V_{b0})} R \cdot T \cdot \dot{m}_{bki} - \frac{A_b}{M \cdot (A_b \cdot (L - x) + V_{b0})} p_b \cdot A_b \cdot \dot{x} - \frac{d}{M} \ddot{x} - \frac{k}{M} \cdot \dot{x} \quad (3.10)$$

As the pressure build-up in chamber depends on the charging and discharging mass flow rate, so we must define the \dot{m}_{be} and \dot{m}_{ki} .

$$\dot{m}_{be} = \mu_{fa} \cdot p_{be} \cdot A_{fa}(u_b) \cdot \sqrt{\frac{2}{R \cdot T_{be}}} \cdot \Psi \quad (3.11)$$

$$\dot{m}_{ki} = \mu_{fb} \cdot p_{ki} \cdot A_{fb}(u_b) \cdot \sqrt{\frac{\chi}{R \cdot T_{ki}}} \cdot \Psi' \quad (3.12)$$

Detailed discussion can be seen in dissertation.

2.3. Sliding mode control

Let us assume that a system is described by the n^{th} order nonlinear differential equation.

$$x^{(n)}(t) = f(x(t)) + G(x(t), u_b(t)) + z(t) \quad (3.13)$$

$$y(t) = x(t) \quad (3.14)$$

Where

$$\frac{d}{dt}x^{(i)} = x^{(i+1)} \quad (i = 0, \dots, n-1)$$

and

$$\mathbf{x}(t) = (x(t), \dot{x}(t), \dots, x^{(n-1)}(t)) \in \mathbb{R}^n$$

is the state vector of the system,

$$y(t) \in \mathbb{R}$$

x is the output signal (position). The Eqn. (3.10) and (3.13) implies that, our pneumatic system can be described by a 3rd order differential equation. Substituting (3.11) and (3.12) into (3.10) yields the nonlinear functions $f(\mathbf{x}(t))$ and $G(\mathbf{x}(t), u_b(t))$:

$$f(\mathbf{x}(t)) = -\frac{A_a}{M \cdot (A_a \cdot x + V_{a0})} p_a \cdot A_a \cdot \dot{x} - \frac{A_b}{M \cdot (A_b \cdot (L-x) + V_{b0})} p_b \cdot A_b \cdot \dot{x} - \frac{d}{M} \ddot{x} - \frac{k}{M} \cdot \dot{x}$$

$$G(\mathbf{x}(t), u_b(t)) = \frac{A_a}{M \cdot (A_a \cdot x + V_{a0})} R \cdot T \cdot \mu_{fa} \cdot p_{be} \cdot A_{fa}(u_b) \cdot \sqrt{\frac{2}{R \cdot T_{be}}} \cdot \Psi +$$

$$+ \frac{A_b}{M \cdot (A_b \cdot (L-x) + V_{b0})} R \cdot T \cdot \mu_{fb} \cdot p_{ki} \cdot A_{fb}(u_b) \cdot \sqrt{\frac{\chi}{R \cdot T_{ki}}} \cdot \Psi'$$

The control problem is to force $y(t)$ to track a desired $y_d(t)$ reference signal. The tracking error for the system is defined as

$$e(t) = y_d(t) - y(t) \quad \text{- vel,} \quad (3.15)$$

idő szerinti i -edik deriváltját pedig

$$e^{(i)}(t) = y_d^{(i)}(t) - y^{(i)}(t) \quad \text{- vel.} \quad (3.16)$$

Where $z(t)$ indicates the time-function of the disturbance. We assumed about the disturbance that it is limited. The limits are due to the physical limits of the system. If the value of $y_d(t)$ is constant, we can speak about positioning. In available literature usually the effect of the step response is examined. We assume about the $y_d(t)$ set value that it can be at least n -times differentiated. As a consequence, the $e(t)$ error-signal can be also at least n -times differentiated.

On the basis of the above mentioned, the error-signal and its derivated up to the n^{st} derivated are certainly continuous, thus in all cases the trajectory of the error-signal can be written down with a curve in the n -dimensional state space. Hence the control system must conform to this, too. Let the set value

We carried out positioning experiments with the so-developed regulator, with a horizontal and a vertical orientation cylinder, without loading, and with weight and spring loading. The experimental results show that the controller is robust (Fig. 17.).

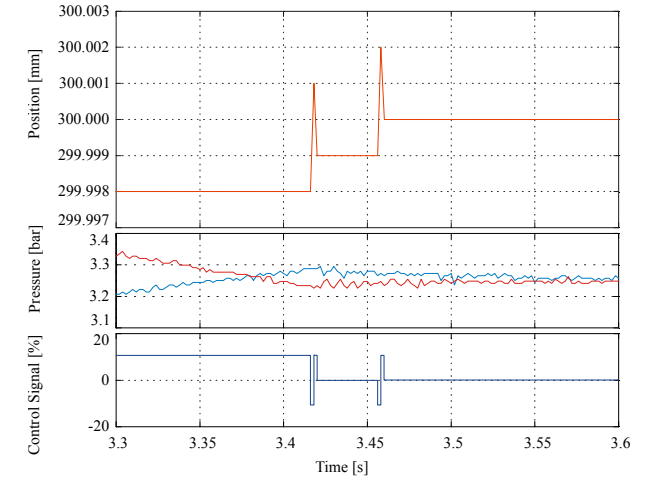


Fig. 18. Positioning with double switching line and double modified boundary layer, with acceleration correction (enlarged)

If we look at figure 18. we can see that thanks to the acceleration-correction we can approach the desired position with even a 0.001 mm pace. If we examine the control-signal, we can see that the regulation reminds us to the „input shaping” method advised by Singhose et. [16][17] to reduce the vibration of flexible systems.

4. SUMMARY AND PROSPECTS

I presented the results that were achieved during the research in numerous national and international conferences and I also published them in journals. The research can also be well-connected with the higher education, I used the gained knowledge during education, the students were also involved in the research. Apart from the numerous national TDK successes my students performed well in international competition, too (<http://www.handson.org.tw> and <http://www.handson.org.tw/video2/3/3.mpg>).

The interest coming from the pneumatic-manufacturers, the industrial appliers and the educational institutions motivates me to appoint primarily as the direction of the further steps the development of a separate and intelligent positioning unit that can communicate with industrial computers and PLCs (Programmable Logic Controllers). The unit gets the requested position from the controlling device, executes the positioning and then gives signal to the controlling device.

if $0 < s < s_a$ and $\ddot{e} > 0$ then $u_{ba} = 0$

if $s = 0$ then $u_{ba} = 0$

if $-s_a < s < 0$ and $\ddot{e} \leq 0$ then $u_{ba} = 0$

if $-s_a < s < 0$ and $\ddot{e} > 0$ then $u_{ba} = u_b$

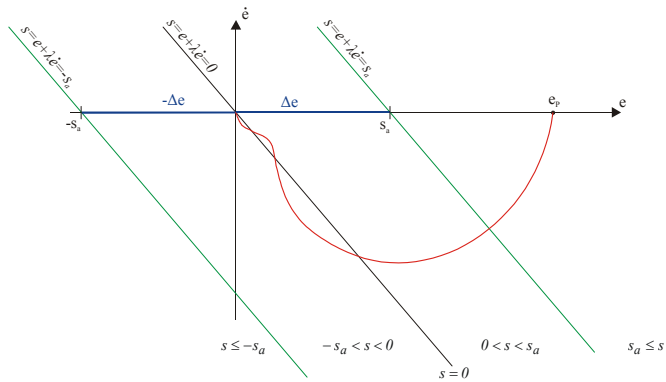


Fig. 16. Sliding mode control with acceleration correction

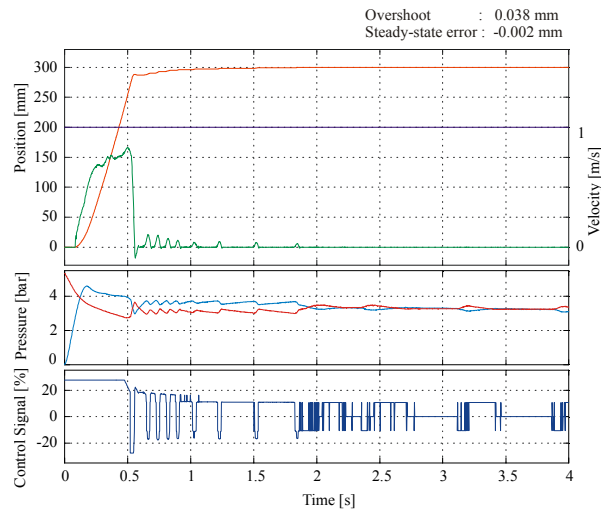


Fig. 17. Positioning with double switching line and double modified boundary layer with acceleration correction,

be step-input, which can be derived in the $t > 0$ range arbitrary times. For the dissolving of the error appearing due to the step-input let's design in the n-dimensional phase plane of the error-signal a continuous trajectory, which ends in the origo and which can be theoretically precisely followed. Naturally this presumes, that during the design of the trajectory we take into consideration the physical limits of the system and the control-signal. In our case I have chosen a two-dimensional phase-plane instead of the needed three-dimensional phase-plane because of the practical difficulties of the realisation. This means non-modelled dynamics and causes chattering. The used servo-valve is closed in the middle position. As its result, chattering cannot only be harmful but also useful, it prevents that the pressure in the cylinders decrease up to the pressure of the atmosphere. The first thesis shows the design of a two-dimensional sliding surface (sliding line) that can be used also in practice.

3. NEW SCIENTIFIC RESULTS

THESIS 1: (Attach to design of the sliding surface)

I designed a new positioning, along two sliding lines. I verified with experimental results that the designed controlling method is fast, stabil, robust and the gained positioning precisiy is the same as the resolution of the applied incremental encoder, which is 0.01 mm. I proved with experiments that the examined system is only dynamically handleable, this is helped by the applied two-dimensional phase-space. I worked out a new method, the Pulse Number Modified Sliding Mode Control, the PNMSMC. The new method guarantees that the pressure in any area of the work-cylinder drops under a critical value.

In the literature we can find about the choosing of the sliding line, that the optimum needs to be found between two assumptions effecting against each other. The less the gradation of the line is, the faster the trajectory reaches the sliding line, but also the slower the laying is along the line. Because of the fulfillment of the requirements of speed and robustness at the same time, the adaptive changing of the gradation of the sliding line is advised [11][12][13](figure 4).

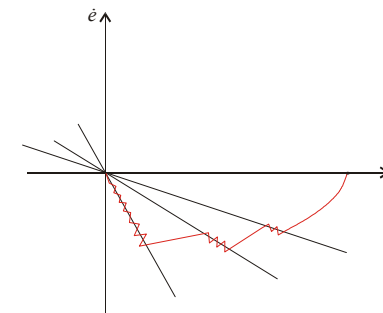


Fig. 4. Adaptive changing of the gradation of the sliding line

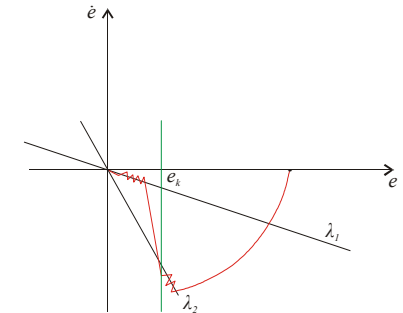


Fig. 5. Two sliding lines in the state space

I did not get a good result when applying the suggested solution in pneumatic positioning with sliding-mode control. We can put it this way, that I united the disadvantages of the high and low gradation sliding line.

I suggested a new solution (figure 5) to eliminate the disadvantages and unite the advantages. In the solution I applied two sliding lines in a way that in the beginning I executed the positioning (far from the end position) along the higher gradation line, and in the last phase of the positioning along the lower gradation line.

The higher gradation line guarantees the fast approaching and the lower gradation line guarantees the precise positioning. Where e_k means that threshold where we change from the higher gradation line to the lower gradation line.

$$\begin{aligned} \text{if } e > e_k & \quad s = e + \lambda_2 \cdot \dot{e} \\ \text{if } e < e_k & \quad s = e + \lambda_1 \cdot \dot{e} \end{aligned}$$

The experimental results (Fig. 6. and Fig. 7.) indicate that the proposed control method is fast, accurate and the overshoot is advantageous too.

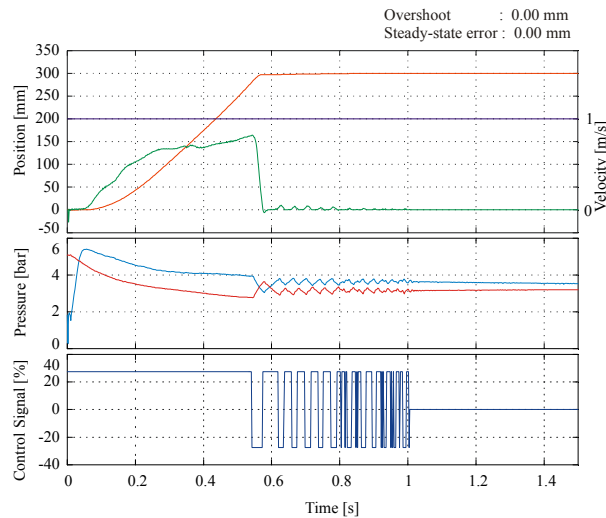


Fig. 6. Positioning with double switching line (time response)

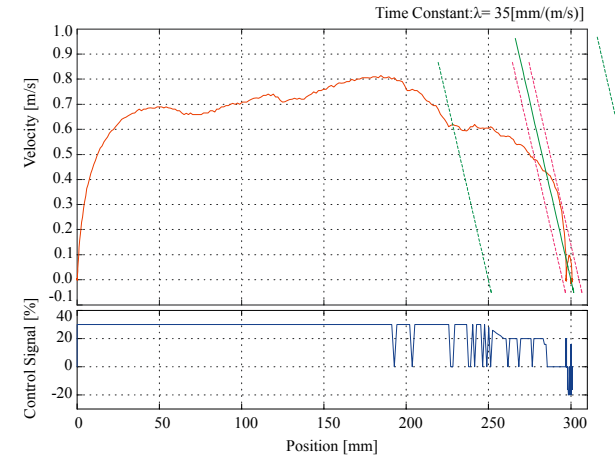


Fig. 15. Positioning in vertical orientation (phase plane trajectories)

THESIS 3: (Attach to law of sliding mode control with chattering free implementation)

I showed by experimental results that in case of the higher precision positioning (0,001 mm) the stability is much worse than in case of the usage of 0.01 dissociation incremental encoder. To improve stability and to increase the precision of the positioning I worked out a new controlling law, the essence of which is that I modify the degree of the intervention according to the indication sign of the acceleration, hence I partly bring back the third dimension (the second derivated of the error-signal) which was left out because of simplification.

The essence of the solution is that along the sliding line an acceleration correction threshold is constructed and then I correct the control-signal beyond the threshold according to the indication sign of the acceleration. (Fig. 16.).

Outside of boundary layer ($s < -s_a$ or $s_a \leq s$) we can get the control signal (u_b) with the leave signum of acceleration out of consideration:

$$\text{if } s \leq -s_a \quad \text{or} \quad s_a \leq s \quad \text{then} \quad u_{ba} = u_b$$

Inside of boundary layer ($\pm s_a$) we can get the control signal with the take signum of acceleration into consideration.

$$\text{if } 0 < s < s_a \quad \text{and} \quad \ddot{e} \leq 0 \quad \text{then} \quad u_{ba} = u_b$$

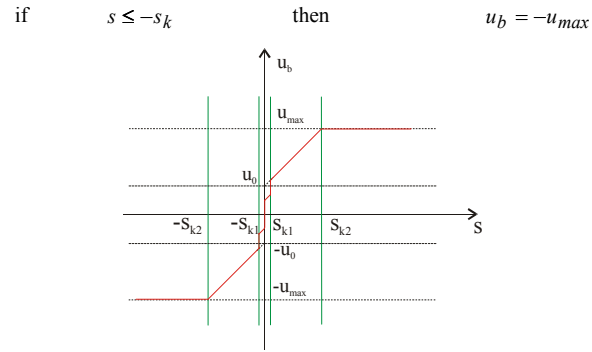


Fig. 13. Using two modify boundary layers

if $-s_{k1} < s < s_{k1}$ then $u_b = \delta \cdot \left(u_0 \cdot \text{sign}(s) + \frac{u_{max} - u_0}{s_{k2}} \cdot s \right)$
 where $0 < \delta \leq 1$

if $-s_{k2} < s \leq -s_{k1}$ or $s_{k1} \leq s < s_{k2}$ then $u_b = u_0 \cdot \text{sign}(s) + \frac{u_{max} - u_0}{s_{k2}} \cdot s$

if $s \geq s_{k2}$ then $u_b = u_{max}$

if $s \leq -s_{k2}$ then $u_b = -u_{max}$

The proposed control scheme is implemented on an vertical orientation cylinder (gravitational force is the disturbance). Fig. 14. and Fig. 15. shows the result.

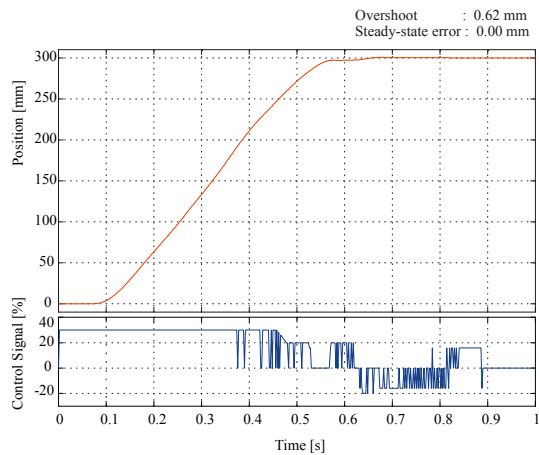


Fig. 14. Positioning in vertical orientation (time response)

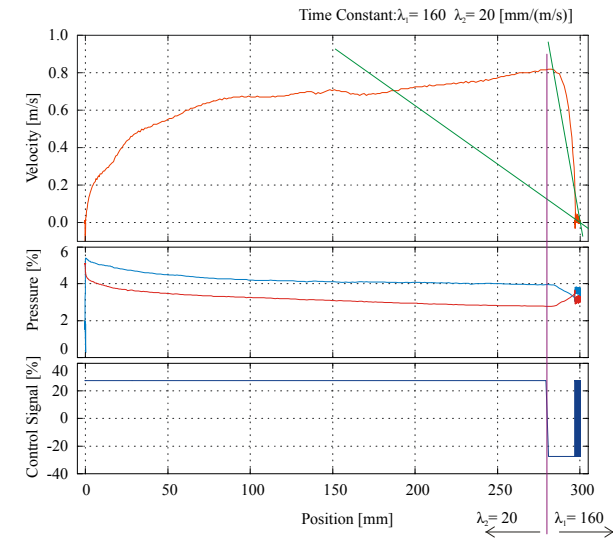


Fig. 7. Positioning with double switching line (phase plane trajectories)

A Pulse Number Modified Sliding Mode Control control programme runs down in every 2 milliseconds ($\Delta T = 2$ ms). For a cycle time in the beginning of every $n(e) \times \Delta T$ period the opposite side of the cylinder is under pressure, preventing the decrease of the pressure (Fig. 8.). This modification needs to be applied only outside of the border-layer. The e error defines the $n(e)$ value of the impulse number.

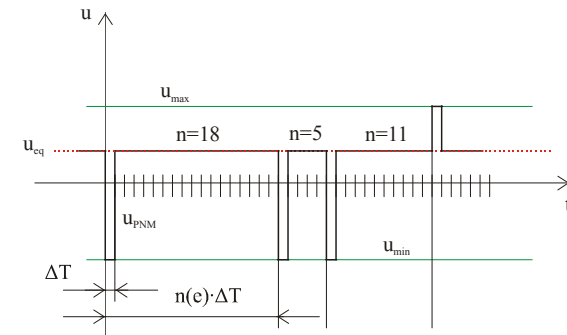


Fig. 8. Pulse Number Modified Sliding Mode Control

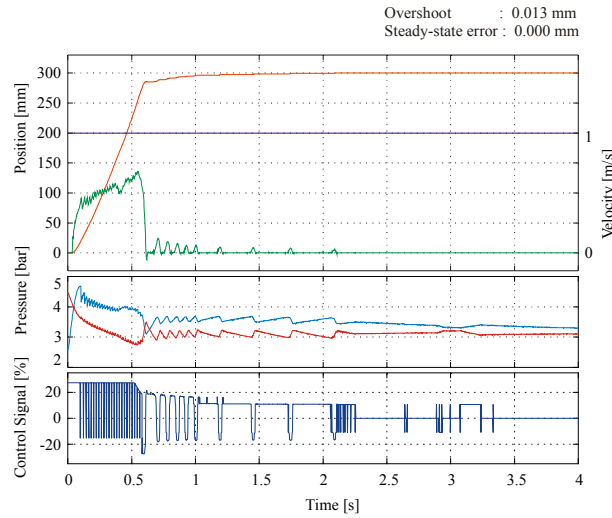


Fig. 9. Positioning with double switching line and modified boundary layer, with acceleration correction and Pulse Number Modified Sliding Mode Control (time response)

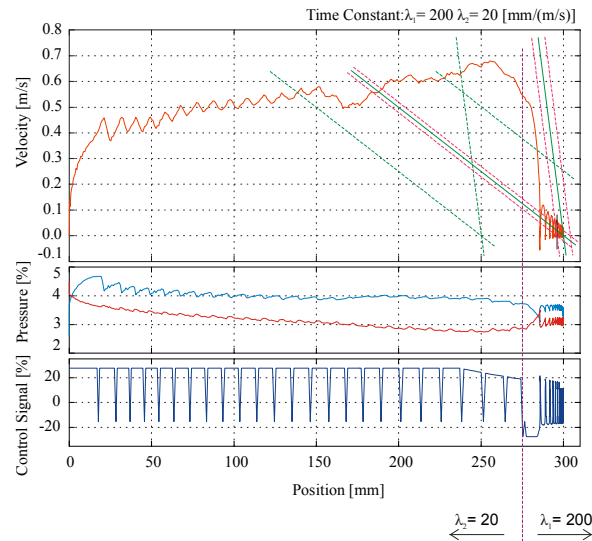


Fig. 10. Positioning with double switching line and modified boundary layer, with acceleration correction and Pulse Number Modified Sliding Mode Control (phase plane trajectories)

THESIS 2: (Attach to law of sliding mode control with chattering free implementation)

To reduce the chattering, I worked out a positioning along a new, multiple boundary layer sliding line which is with an interloping-signal that has a threshold. I proved by experimental result that the method is effective.

Based on paper [14] we can improve the performance of positioning and reduce the chattering, by introducing a boundary layer from both sides of the sliding line where the magnitude of the control signal u is defined by the distance between the sliding line (Fig. 11.).

I proved with experiments that though the chattering was reduced, close to the desired position the control-signal decreased, and because of this the phenomenon of the stick-slip appeared and the precision of the positioning declined intensely according to the previous results. Thus, the boundary layer in case of pneumatic positioning cannot be used without a modified control-signal. To eliminate the disadvantage I suggested to introduce the threshold that can be seen in figure 12. With this solution we can prevent the control-signal to be reduced under the u_0 threshold. We can improve the stand further if we introduce a new threshold along the sliding line and we reduce further proportionately the control-signal in two steps. (figure 13)

$$u_b = \frac{u_{max}}{s_k} \cdot s \quad \text{ha} \quad -s_k < s < s_k$$

$$u_b = u_{max} \quad \text{ha} \quad s \geq s_k$$

$$u_b = -u_{max} \quad \text{ha} \quad s \leq -s_k$$

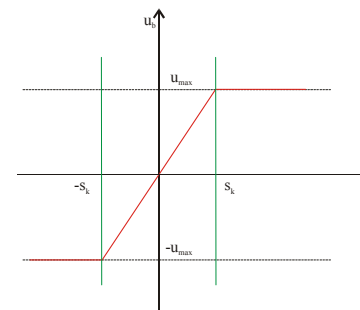


Fig. 11. Using boundary layer

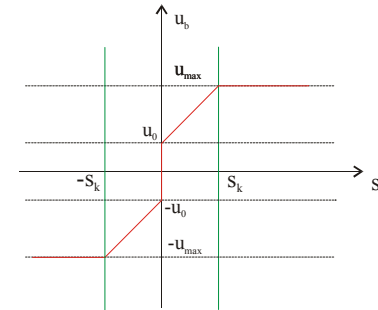


Fig. 12. Using modify boundary layer

$$\text{if } -s_k < s < s_k \quad \text{then} \quad u_b = u_0 \cdot \text{sign}(s) + \frac{u_{max} - u_0}{s_k} \cdot s$$

$$\text{if } s \geq s_k \quad \text{then} \quad u_b = u_{max}$$