SZENT ISTVÁN EGYETEM

ENERGY EFFICIENT POSITION CONTROL OF PNEUMATIC LINEAR DRIVES

Thesis of PhD work

Eszter Mayerné Sárközi

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denomination:	Mechanical Engineering Doctoral School
science:	Agricultural Engineering
leader:	Prof. Dr. István Farkas, DSc Faculty of Mechanical Engineering Szent István University Gödöllő, Hungary
Supervisor:	Prof. Dr. László Jánosi, CSc Institute of Industrial Technologies Faculty of Mechanical Engineering Szent István University Gödöllő, Hungary

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Affirmation of head of school	Affirmation of supervisor

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NOMENCLATURE

Symbol	**Description**	**Unit**
A_2, A_4	surfaces of the piston	[m^2]
A_{x_y}	area of an orifice in x_y direction	[m^2]
c_v	coefficient (Stribeck friction model)	[-]
D	derivative term of PID controller	[-]
e	error value	[-]
f	viscous friction coefficient	[-]
F_C	Coulomb friction	[N]
F_s	friction force	[N]
F_{St}	static friction force	[N]
F_{Str}	Stribeck friction force	[N]
F_t	external load force	[N]
I	integral term of PID controller	[-]
K_v	velocity coefficient	[-]
K_x	position coefficient	[-]
Lf	air consumption coefficient	[-]
Lf_{kr}	critical air consumption coefficient	[-]
m	mass	[kg]
$\dot{m}_{x\_y}$	mass flow in x_y direction	[kg/s]
P	proportional term of PID controller	[-]
p	pressure	[Pa]
P_{St}	proportional coefficient of static controller	[-]
PV	process value	[V]
q_{x_y}	flow rate in x_y direction	[-]
R	universal gas constant	[J/kgK]
s_k	boundary layer	[-]

Symbol	**Description**	**Unit**
SP	set point	[V]
T	temperature	[K]
$u(t)$	control variable	[-]
u_{max}	maximal control variable	[V]
U_v	control signal	[V]
v	velocity	[m/s]
V_n	air consumption	[N(l)]
x	displacement of the piston	[m]
κ	adiabatic index	[-]
λ	the slope of the sliding surface	[-]
Acronym		
ASH	absolute steady state error	
ITAE	integral time absolute error criterion	
PID	proportional-integral-derivative controller	

1. INTRODUCTION AND OBJECTIVES

Below I present the actuality of the topic and then I outline the objectives of my work.

1.1. Actuality and relevance of the topic

There is a great demand in the industry for linear drives that can be stopped in any position along their stroke with high accuracy. This task is primarily solved by electric linear actuators, because the electrical network is easily accessible, the drive can be well controlled, maintenance is not required and its operation is relatively inexpensive.

In recent years, in addition to electric linear actuators, the use of pneumatic cylinders has been extended in positioning tasks thanks to the benefits of servopneumatic drive technology. The investment cost of the servopneumatic drive system is much lower than that of electric drives, the pneumatic actuators are not sensitive to the overload, they are explosion-proof and do not contaminate the work piece in case of leakage.

Pneumatic cylinders are actuators that are traditionally stopped in the two end positions by mechanical elements, mostly controlled by way valves which have two or three discrete switching positions. Other than end positioning require special solutions. Since pneumatic systems exhibit non-linear behaviour due to air compressibility and friction, their precise positioning cannot be solved by open-loop control, in these case closed loop control must be applied with the feedback of the piston position signal. Therefore, the emergence of industrial control devices with adequate computing capacity and real-time operation was a prerequisite for the widespread deployment of positioning pneumatic drive systems. These control devices run the control algorithm, they intervene into the process through the way-valve primarily on the basis of the position signal. Depending on the configuration of the pneumatic system several different control algorithms are available, since in the past decades many researches have been carried out to design and develop them. From the point of view of system behaviour, the applied control algorithm is crucial; in their design the typical primary goal was to increase the available accuracy and to eliminate the negative effects of load change.

However, there is another important aspect to consider during the design and evaluation of control algorithms, which is the energy consumption. Although the investment cost of the servopneumatic drive system is relatively low, the operating cost is very high. The reason for this is that compressed air is one of the most expensive energy sources due to the very poor overall efficiency of pneumatics and the high proportion of gap losses. For positioned pneumatic linear drives, the applied control algorithm also has an impact on energy consumption, as each algorithm interferes with the process in different ways. Therefore, it is important to examine the control algorithms from the energy efficiency aspect in order to reduce the energy demand of pneumatic linear drives, thereby improving its environmental impact and its competitiveness against electric drives.

1.2. Objectives

The main objective of the research is to perform a unified, complex evaluation of certain control algorithms of positioned servopneumatic linear drives, with particular regard to air consumption. This complex evaluation can be used as a decision support to choose the most appropriate control algorithm for each practical applications.

Furthermore an aim of the research is to examine the solutions that can reduce air consumption without structural changes in the servopneumatic system.

In the course of the research I aimed to examine the widespread elements and solutions that are commonly used in industry.

The research can be divided into the following tasks:

- assembling of the examined servopneumatic system, the control system and the measuring system, implementing the measuring softwares,
- creating a mathematical model of the servopneumatic system and the block-oriented solution of the model, accomplishing the identification and validation of the model,
- selecting the examined control algorithms and implementing them into the system’s model,
- creating a unified optimization method of control algorithms, selecting the criteria function and implementing the optimization,
- evaluating the control algorithms through positioning tasks based on the following criteria: absolute steady state error, settling time, overshoot, air consumption, response to change in load force, response to target position change,
- comparing the control algorithms based on technological approach,
- studying the possibilities of reducing air consumption without structural modifications of the servopneumatic system.

2. MATERIAL AND METHOD

In the following, I present the experimental apparatus which has been used to achieve my research goals, the modelling procedures of the investigated system and the evaluation methods.

2.1. Experimental apparatus

The circuit diagram of the examined servopneumatic system is shown in Figure 1. The used linear actuator is a double acting rodless Festo DGPL-25-450-PPV-A-KF-B pneumatic cylinder (C) with slide, which is controlled by an MPYE-5-1/8-LF-010-B 5/3 proportional way valve (Y). Movement of the piston is detected by an MLO-POT-0450-TLF analogue encoder (X) with a resolution of 0.01 mm. In addition to the encoder, three SDE-1-D10-G2 analogue pressure sensors operate in the system, sensing the supply pressure and the pressure entering and exiting the cylinder.

C	Cylinder
X	Encoder
P1, P2, P4	Pressure sensors
Y	5/3 proportional valve
ZSz	3/2 way valve
LTE	Air service unit

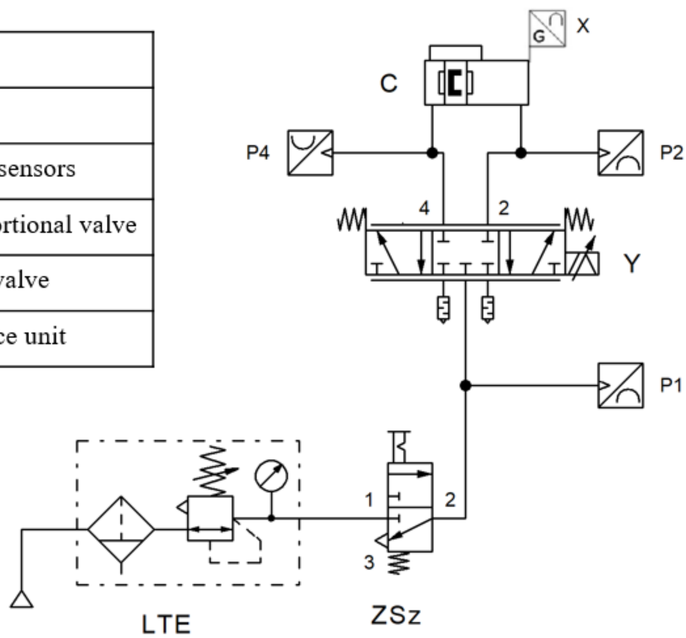

Figure 1. Pneumatic circuit diagram of the examined system

2.2. Mathematical model of the servopneumatic system

As I chose such a method for unified tuning of the five controllers, that requires large amount of computing power and settings, I performed it on the model of the system. I applied Kelvin-Thomson's approach to create block-oriented mathematical modelling of the system. After identification and validation, this model provided an appropriate basis for tuning and testing the examined control algorithms.

2.2.1 Mathematical model of proportional valve

The mathematical model of the proportional valve defines mass flows and air consumption of the valve as a function of control signal and entering and exiting pressures. The model contains four throttle cross sections according to the physical configuration of the valve, and defines the mass flows in these cross sections based on the equation 1.:

$$\dot{m}_{x\_y} = A_{x\_y}\, p_x \sqrt{\frac{2\,\kappa}{(\kappa-1)\,R\,T}\left(\left(\frac{p_y}{p_x}\right)^{\frac{2}{\kappa}} - \left(\frac{p_x}{p_y}\right)^{\frac{\kappa+1}{\kappa}}\right)}. \qquad (1)$$

In proportional way valves the throttle cross sections vary depending on the control signal. To describe the operation of the 5/3 proportional valve, I have defined a new flow characteristic that takes into account the effect of leakages in all four flow cross sections.

According to the manufacturer's practice, I have interpreted this characteristic as a flow rate [%] - control signal [V] function. The flow rate (q_{x_y}) indicates the ratio of the current flow cross section (A_{x_y}) to the completely open state (A_{max}) as a function of the control signal in a given flow direction (x_y):

$$q_{x\_y}\big(U_V(t)\big) = \frac{A_{x\_y}(t)}{A_{max}} \quad . \qquad (2)$$

Based on these, I have determined the new flow characteristic of the 5/3 proportional valve in the four flow directions as the following:

$$q_{1\_4} = \begin{cases} q_{1\_4\_min}, & U_V \leq 5V \\ q_{1\_4} = f(U_V), & U_V > 5V \end{cases},$$

$$q_{1\_2} = \begin{cases} q_{1\_2} = f(U_V), & U_V < 5V \\ q_{1\_2\_min}, & U_V \geq 5V \end{cases},$$

$$q_{2\_3} = \begin{cases} q_{2\_3\_min}, & U_V \leq 5V \\ q_{2\_3} = f(U_V), & U_V > 5V \end{cases},$$

$$q_{4\_5} = \begin{cases} q_{4\_5} = f(U_V), & U_V < 5V \\ q_{4\_5\_min}, & U_V \geq 5V \end{cases}. \qquad (3)$$

In the identification of the proportional valve model, I assigned values to the model parameters. These parameters are the time constant of the valve (T_{sz}), the maximum flow cross-section (A_{max}), which were determined on the basis of the parameters specified in the valve catalogue, and the flow characteristics in the four flow cross sections. These were determined by container filling measurements and parameter identification.

2.2.2. Mathematical model of pneumatic cylinder

The cylinder model includes the solution of chamber models and the application of the force equation of the piston.

The force equation of the piston:

$$m\,\ddot{x} = A_4\,p_4 - A_2\,p_2 - F_s - F_t. \tag{4}$$

The pressure in a cylinder chamber (*n* is the indexing of the chambers; 2 or 4):

$$\frac{dp_n}{dt} = \frac{1}{V_n}\left(\frac{dm_n}{dt}\,R\,T - p_n\,\frac{dV_n}{dt}\right). \tag{5}$$

I defined four input parameters in the cylinder model. Two of these, the load force (F_t [N]) and the moved weight (m_m [kg]) are system parameters that are specified by the given task. The other two input parameters are the mass flows (dm/dt_2 and dm/dt_4 [kg/s]) arriving to the pneumatic ports of the cylinder. The instantaneous values of these variables are calculated by the valve model. The cylinder block has one output parameter, which is the current position of the piston (x [m]).

In order to calculate the friction Stribeck friction model was applied:

$$\mathrm{F_{Str}} = \left(\mathrm{F_C} + (\mathrm{F_{st}} - \mathrm{F_C})\,\exp(-\mathrm{c_v}\,|\mathrm{v}|)\right)\mathrm{sign(v)} + \mathrm{f\,v}\,, \tag{6}$$

whose parameters were identified by measurements.

2.2.3. Validation of system model

I compared the behaviours of the actual pneumatic system and the parameterised system model based on simulations and measurements. In each case, the unloaded cylinder was moved from its initial position to the end position with different proportional valve openings, resulting different piston speeds. During the simulations and measurements, the current piston position was recorded as a function of time. I have shown separately the cases where stick-slip occurs at low speed of the piston (Figure 2) and those where the piston speed is higher due to the larger propotional valve opening, so there is no stick-slip (Figure 3).

The diagrams show that the identified system model simulates properly the actual operation of the system at low and high speeds too. At low speeds, the stick-slip also occurs in the simulated system, and the movement is alike the real system's behaviour in both its nature and speed.

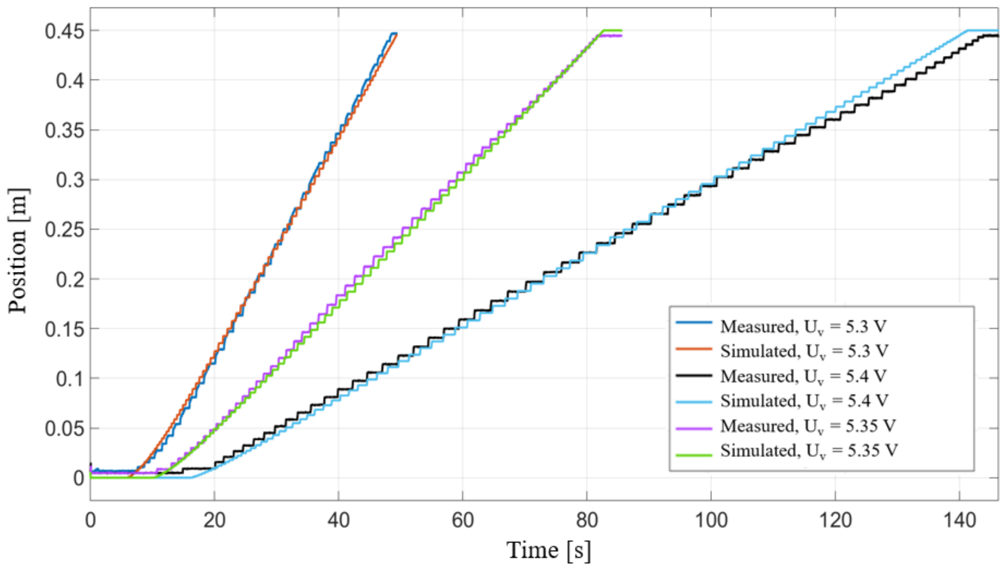

Figure 2. Measured and simulated displacements of the piston at low speeds due to different proportional valve openings

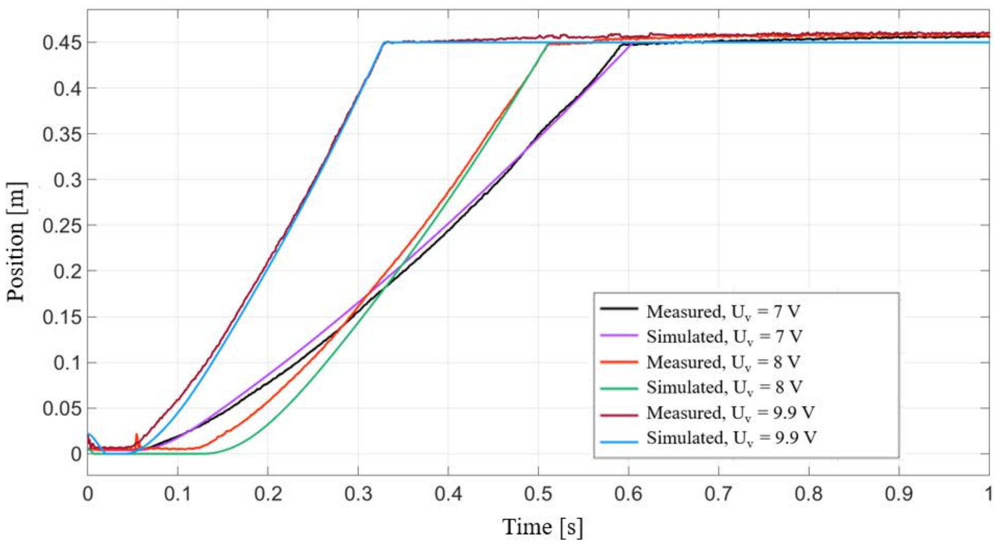

Figure 3. Measured and simulated displacements of the piston at high speeds due to different proportional valve openings

2.3. Examined control algorithms

Five control algorithms were selected for the examinations: PID controller, modified status controller, chattering-free sliding mode controller and two cascade-structured controllers.

The PID controller interferes according to the error value (e), where error is the difference between the desired setpoint (SP(t)) and the process value (PV(t)):

$$e(t) = SP(t) - PV(t). \tag{7}$$

The control variable (u(t)) provided by the PID controller is the weighted sum of the current error, the integral of the error and the derivative of the error, according to the following equation:

$$u(t) = P\, e(t) + I \int_0^t e(t)dt + D\, \frac{d}{dt} e(t), \tag{8}$$

where P is the proportional term, I is the integral term and D is the derivative term of the PID controller.

In addition to the error value (e), the modified status controller also takes into account the piston speed ($\dot{x}$) as follows:

$$u(t) = P_{st}\big(K_x\, e(t) - K_v\, \dot{x}(t)\big). \tag{9}$$

In order to use the chattering-free sliding mode controller, the maximum control signal (u_{max}) as well as the sliding surface (s) must be specified in the $e - \dot{e}$ state space:

$$s = e + \lambda \dot{e} = 0. \tag{10}$$

In addition, an s_k boundary layer should be defined in both directions parallel to this sliding surface. The control signal:

$$u = u_{max}\, sat(s) = \begin{cases} u_{max}\, sign(s), & |s| > s_k \\ \dfrac{u_{max}}{s_k}\, s, & |s| \le s_k \end{cases} \tag{11}$$

The block-oriented model block of these controllers has two inputs: the setpoint (SP), which is in this case the signal corresponding to the target position and the current position (PV) feedback signal from the cylinder output. The controller output is the control signal (U_v).

The last two types of examined control algorithms differ from the previous ones, both have cascade structures with internal pressure differential feedback and proportional gain, one of them is with external PID controller and the other one is with external modified status controller. (Simplified name is referred to as PID cascade and modified status cascade controller.) For these controllers, the structure of the external PID or external modified status controller is the same as the previous one, their setpoint is the desired position, and their feedback signal is the current piston position. In the cascade structure, the output of the external controller is used as the setpoint of the internal proportional gain controller, while the feedback signal of the internal controller in our case is the pressure difference of the cylinder chambers. Due to the same piston surfaces, its value is proportional to the force caused by the pressures, which affects to the piston. Since the inner circle must be faster, I used proportional gain there.

2.4. Optimization process of control algorithms

After selecting the control algorithms, the next step was their tuning, that is finding the optimal values of the control parameters. In order to make a unified comparison of the five control algorithms, I have been looking for a global optimization method that can be applied universally to all five control algorithms with adequate performance. Due to the complexity and non-linear behaviour of the system, I have chosen genetic algorithm from random search methods that I have supplemented with local gradient optimization. The selected criterion function was the integral time-weighted absolute error criterion (ITAE):

$$ITAE = \int_0^t |e| t \, dt \, . \tag{12}$$

As the genetic algorithm is not an exhaustive search method and the target function and the system are complex, the optimization results were only near-optimal settings. Therefore, the tests were not carried out on an optimally tuned controller, but were carried out at 10-10 settings close to the optimum. To determine the 10-10 near-optimal settings, I defined an optimization cycle.

For the optimization of the controllers and the tests I have determined a positioning task based on a typical workflow of industrial application of position controlled pneumatic drives. During the specified positioning task, the piston starts from its initial position and then stops at a position corresponding to 80% of the stroke length (360 mm). So the work space of the cylinder is used, but there is room for the controller to intervene properly. During this positioning task external force load was not applied, I took into account the weight of the moving slide unit as a mass load. The optimization was done in a 5 s time interval.

2.5. Evaluations

In order to compare the control algorithms, the positioning results of the servopneumatic system were evaluated on the basis of the absolute steady state error [mm], settling time [s], overshoot [mm] and air consumption [(N)l]. The steady state error and overshoot were determined at 0.01 mm resolution according to the resolution of the encoder.

For running the models, I used 5 s run time and fixed 0.001 s increments, which corresponds to the cycle time of the controller.

During the first positioning task, the piston was moved from its initial position (0 mm) to the target position (360 mm) of 80% of the total stroke, with the weight of the slide unit without external force load.

In the next positioning task I have examined the response to the external load force change. Here, too, the piston was moved from the initial position to the 360 mm target position, but besides the weight of the slide unit further external load forces of different values were applied during each positioning. The external force values were: 50 N, 100 N, 150 N, 200 N.

Finally, I have examined the effect of the target position change on the effectiveness of tuned control algorithms. In doing so, the piston was moved to a different target positions from initial position without external force load, with moving the weight of the slide unit. The used target position values were the followings (expressed as a percentage of stroke length/as the control signal/as piston position): 20%/2V/90 mm, 40%/4 V/180 mm, 50%/5 V/225 mm, 60%/6V/270 mm, 70%/7V/315 mm, 90%/9V/405 mm.

One of the key questions in evaluating the features listed above is whether the results of each control algorithms are different. To decide this, I have performed statistical calculations and tests: each sample was compared in pairs. The used test method were two-sample t-test, as well as Welch- or d-test.

3. RESULTS

In the first part of this chapter, I evaluate the five control algorithms on the basis of the results of the positioning tasks, then I present the new complex evaluation system and finally I present the function criteria of reducing the air consumption.

3.1. Comparison and evaluation of control algorithms

3.1.1. Absolute steady state error

Table 1 summarizes the results of the control algorithms during the optimized positioning task, where the piston is directed from the initial position to the 360 mm position without external force load, with the moving mass is the weight of the slide unit. In this table, the mean and standard deviation values of the absolute steady state error, overshoot, and settling time of the 10-10 tuned control algorithms are shown.

Table 1 The positioning results of the five control algorithms in the optimized positioning task

Control algorithm		**Absolute steady state error**		**Overshoot**		**Settling time**	
		Mean [mm]	**Devia-tion**	**Mean [mm]**	**Devia-tion**	**Mean [s]**	**Devia-tion**
1.	PID controller	0,009	0,003	0,228	0,261	0,201	0,003
2.	Modified status controller	0,010	0,000	1,879	3,154	0,200	0,000
3.	Sliding mode c.	0,011	0,003	0,501	0,507	0,207	0,005
4.	PID cascade c.	0,010	0,000	1,046	2,020	0,200	0,000
5.	Modified status cascade c.	0,010	0,000	1,702	0,829	0,200	0,000

These values were compared with test statistics to determine whether there was a significant difference between the achieved results of the control algorithms. In the following, I present the comparison of absolute stead state errors of the control algorithms.

Two-sample t-test and d-test were performed in sample pairs, the used significance level was p = 95%, the results are shown in Table 2.

Table 2. Comparison of control algorithms based on absolute steady state error

Compared pairs	**applied test method**	**statement**
PID / Sliding mode controller	t test	no difference
PID / Modified status c., PID cascade c., Modified status cascade c.	d test	no difference
Sliding mode c. / Modified status c., PID cascade c., Modified status cascade c.	d test	no difference

On this basis, it can be stated that at 95% significance level there is no difference between the five examined control algorithms in the achieved steady state error while performing the optimized positioning task.

If, in addition to the statistics, we look at the positioning errors of the control algorithms from a technological point of view, it can be also stated that each algorithm fulfils the 0.02 mm steady state error limit requirement. Considering the 0.01 mm accuracy of the encoder, this is an excellent result.

3.1.2. Air consumption

Table 3 summarizes the air consumption results of the five control algorithms during the optimized positioning task, where the piston is directed from initial position to the 360 mm position without an external force load with the moving mass is the weight of the slide unit. The table shows means and standard deviations of air consumption values of the 10-10 tuned controller settings.

Table 3 Air consumption results of the five controllers during the optimized positioning task

Control algorithm		**Mean [(N)l]**	**Deviance**
1.	PID controller	1,341	0,111
2.	Modified status controller	1,388	0,037
3.	Sliding mode controller	1,287	0,061
4.	PID cascade controller	3,344	0,792
5.	Modified status cascade controller	3,460	0,382

I have also evaluated the air consumptions of the tuned control algorithms during positioning at varying external load forces. Figure 4 shows the air consumption results of each control algorithm at varying external load forces during the 0 → 360mm positioning task. For each depicted column, 50 position data (10 settings per algorithm * 5 load forces values) were processed.

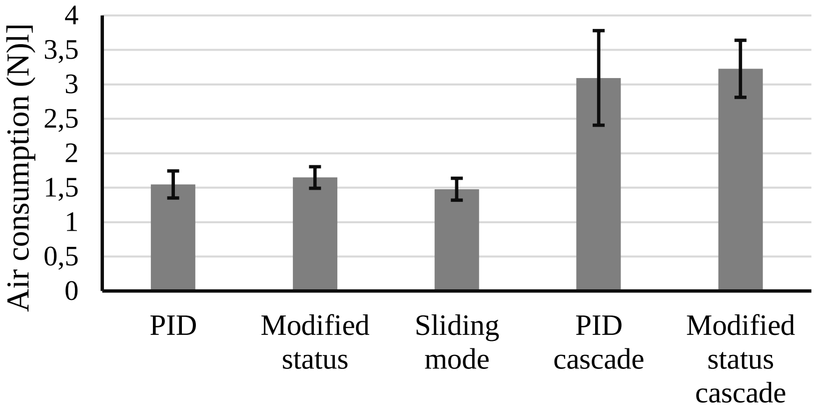

Figure 4. Air consumption results due to varying external load forces

Similarly, the air consumption of each control algorithms were evaluated during positioning to different targets (Figure 5). There were 6 target positions, so with the 10-10 controller settings the air consumption values were determined by 60 positioning tasks per control algorithms.

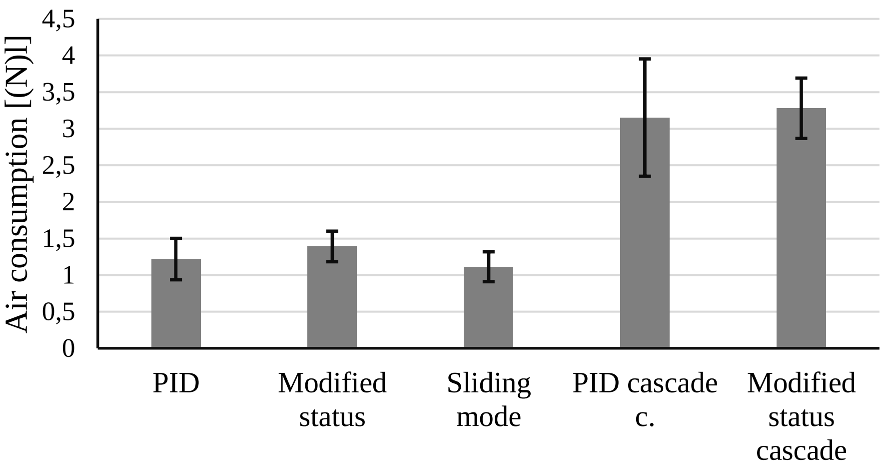

Figure 5. Air consumption results due to varying target positions

The results of air consumption were analysed with test statistics, the conclusions of which are summarized in Table 4.

Table 4. Comparison of the results of air consumptions

Compared pairs	**Statement**		
	in the optimized positioning	**due to varying external force load**	**due to varying target position**
PID / Modified status	no difference	s. difference*	s. difference*
PID / Sliding mode	no difference	no difference	s. difference*
PID / PID cascade	s. difference*	s. difference*	s. difference*
PID / Modified status cascade	s. difference*	s. difference*	s. difference*
Modified status / Sliding mode	s. difference*	s. difference*	s. difference*
Modified status / PID cascade	s. difference*	s. difference*	s. difference*
Modified status / Mod. status cascade	s. difference*	s. difference*	s. difference*
Sliding mode / PID cascade	s. difference*	s. difference*	s. difference*
Sliding mode / Mod. status cascade	s. difference*	s. difference*	s. difference*
PID cascade / Mod. status cascade	no difference	no difference	no difference

*significant difference

Summarizing the results of the comparisons and the nominal air consumption data, examining both the cases of basic optimized positioning, of varying external force load and of varying target position, the following statement can be made (applied significance level p = 95%) on the energy consumption of the control algorithms: the two cascade-structured controllers do not differ significantly from each other, but they differ from other non-cascade controllers. In each positioning task the cascade-structured controllers accomplished them with more than twice as high average air consumption as the other controllers.

3.1.3. Response to the varying external load force

The evaluation criterion of the response to the varying external load force is the achieved absolute steady state errors, the averages of which are shown in Figure 6. The figure shows that for all control algorithms the absolute steady state errors values increase in proportion to the external load force change, but the rate of increase is not the same. The best results were achieved by PID controller, as it responded with the smallest absolute steady state error increase to the external force load increase, PID was followed by the modified status controller and the sliding mode controller. With the largest steady state error, the PID cascade and the modified status cascade controller responded to the force load change.

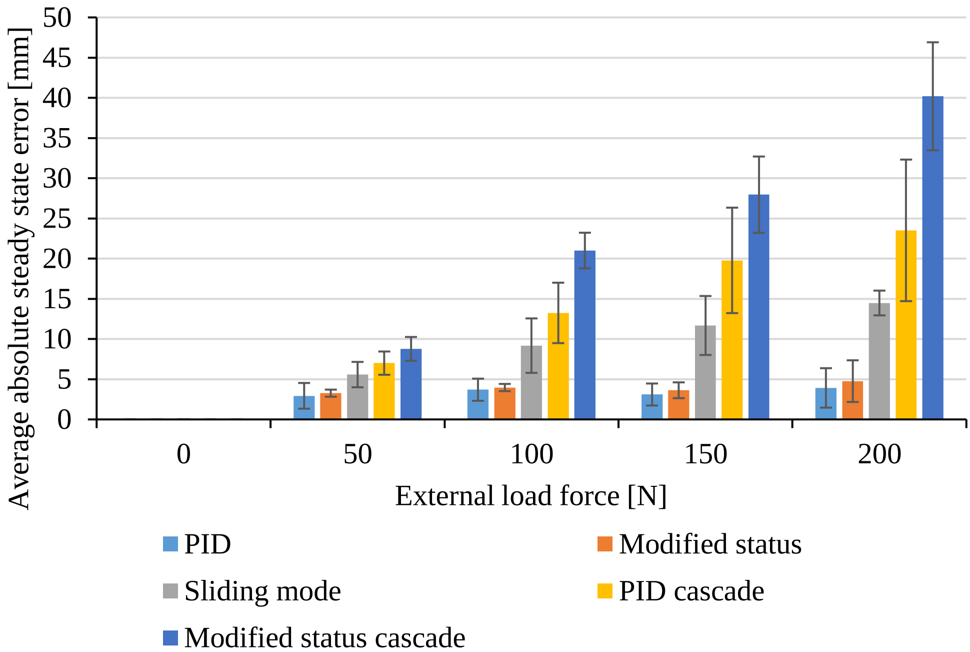

Figure 6. The average absolute steady state errors of the control algorithms due to the varying external load force

Evaluating the response of the tuned control algorithms to the change in the external load force from technological aspect, it can be said to be that in terms of absolute steady state error, none of the control algorithms performed as it was expected in the positioning task. In contrast to the 0.01 mm positioning error under optimum conditions, the best PID controller responded with an absolute steady state error of nearly 3 mm to 50 N extra force load. In addition, in case of PID, modified status and sliding mode controller there were fluctuations around the target position, which is a particularly unfavourable phenomenon in position control applications. In cascade structured controllers, this fluctuations did not occur, but there were especially large absolute steady state errors (over 5 mm due to 50 N force, average values over 10 mm due to 100 N force).

3.1.4. Response to the varying target position

In the evaluation of the response to the varying target position, I considered the each of the 10 tuned controllers with 6 target positions as one sample, so the samples of 60 elements per algorithms were compared based on the absolute steady state error (Figure 7).

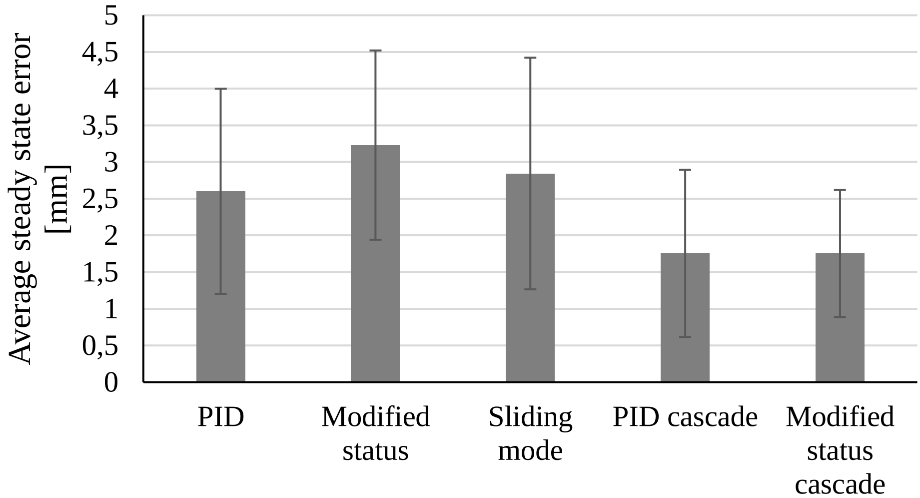

Figure 7. The averages of absolute steady state errors due to varying target position

I performed statistical analysis to evaluate and compare these results, and also examined the nominal values and relative frequencies of the achieved absolute steady state errors. Based on these analyses, I evaluated the responses to the varying target position for each control algorithm as follows: the most favourable results were achieved by cascade controllers (ASH average <2 mm), followed by the PID and sliding mode controller (ASH average <3 mm), the worst results were achieved by the modified status controller (ASH average> 3 mm).

Evaluating these data from technological aspect, it can be declared that none of the control algorithms performed as it was expected in position control tasks (ASH <0.02 mm) due to the varying target position. There were only three cases (1% of the total 5 * 60 positioning data) in which the absolute steady state error did not exceed of 0.02 mm, and in 83% of the positioning cases the absolute static error takes over 1 mm.

3.2. Evaluating system for comparing control algorithms from technological approach

The results of the 3.1. chapter are summarized in a complex, technology-oriented evaluation and decision support system, that supports the selection of the control algorithm for a given technological problem (Table 5). With this table, the most appropriate control algorithm that fits a specified task can be determined by taking into account the circumstances of the problem (e.g. varying load force) and user requirements (e.g. favourable air consumption).

Table 5. Complex technology-oriented comparative system of control algorithms

Control algorithm	**Steady state error**	**Overshoot**	**Settling time**	**Air consumption**	**External load force dependency**	**Target position dependency**
PID	*****	*****	*****	*****	***	**
Modified status	*****	***	*****	*****	***	*
Sliding mode	*****	****	****	*****	**	**
PID cascade	*****	***	*****	**	*	***
Modified status cascade	*****	****	*****	**	*	***

Evaluation criteria were: steady state error, overshoot, settling time, air consumption, external load force dependence and target position dependence.

For the evaluation, a five-step scale was applied, expressing the results of previous chapters. The best-performing algorithms were given five stars, algorithms that behave less favourably were given a reduced number of stars proportionally to the performance difference. However, in case of external load force and target position dependence the maximum number of stars were determined in 3, as in these parameters even the best-ranked algorithms did not meet the expected steady state error.

3.3. New optimization criterion for reducing air consumption

A possible way to reduce the air consumption of the position-controlled servopneumatic system without structural modifications is to tune the control algorithms so as the air consumption is taken into account. In order to achieve this objective, I have developed a new optimization criterion function and examined its impact on the quality of control process characteristics and air consumption.

In addition to the ITAE criterion, the new optimization criterion function that takes into account the air consumption also includes a member, which is proportional to air consumption according to equation 13:

$$Kr_{Lf}(t) = V_n * Lf + \int_0^{\infty} t|e(t)|dt, \tag{13}$$

where $Kr_{\mathrm{Lf}}(t)$ is the new criterion, Vn is the air consumption [(N)l], Lf is the air consumption coefficient [-], e is the error [mm], t is the time [s].

As a null hypothesis, I assumed that the control algorithm settings, which were optimized using the new Kr_{Lf} criterion function would achieve lower air consumption in positioning tasks, depending on the value of the Lf factor, than those control algorithm settings, which were tuned using only the ITAE criterion, while the steady state error and the setting time of the new control settings would deteriorate.

I verified the hypothesis on the modified status control algorithm. During this, I performed the optimization process of the modified status control algorithm using the new criterion function of equation 13 with various Lf values. Results were evaluated based on air consumption (Figure 8) and absolute steady state error (Figure 9).

Based on the results it can be declared that the hypothesis was correct. Thus, the control settings optimized using the new Kr_{Lf} criterion function to achieved lower air consumption as the value of the Lf coefficient increased, while the steady state error and the setting time deteriorated. It can also be stated that, beyond a limit, the increase of the LF coefficient does not lead to a further decrease in air consumption, while the absolute steady state error increases further beyond this limit. This limit Lf coefficient value is the critical air consumption factor (Lf_{kr}). With regard to the practical utilization of the results, it can be suggested that during the optimization process the critical air consumption factor should be never exceeded.

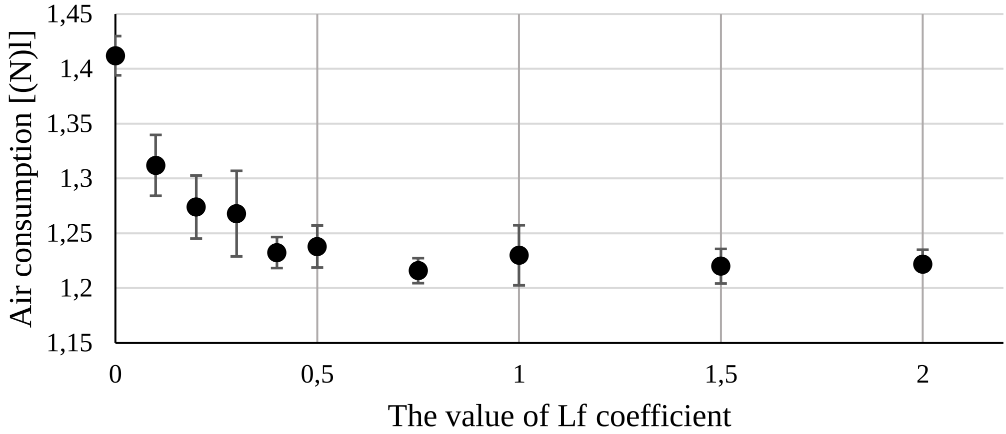

Figure 8. Air consumption results of modified status controller tuned with the new criterion function

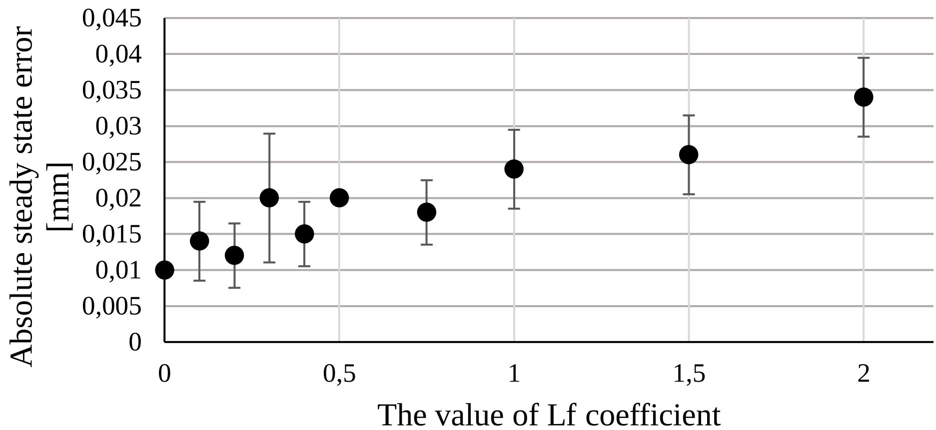

Figure 9. Absolute steady state results of modified status controller tuned with the new criterion function

4. NEW SCIENTIFIC RESULTS

In the following, I summarize the new scientific results of my research.

1. The new flow rate characteristics of the 5/3 pneumatic proportional directional valve

I have created a new flow rate characteristic of the 5/3 pneumatic proportional directional valve (flow rate [%] - control signal [V] characteristic curves) that takes into account the effect of leakages in all four flow cross-sections, so that the actual flow conditions of the 5/3 proportional directional valve can be modelled.

2. Differences in steady state error during pneumatic positioning

With comparative studies and statistical analyses I have proved that, after a unified optimization process at typical working point, there is no difference in the achieved steady state error values between the five examined control algorithms in a position-controlled servopneumatic system with a statistically significance level of 95%. The examined control algorithms: PID controller, modified state controller, chattering-free sliding mode controller, cascade structured PID controller with internal pressure differential feedback and proportional gain, and cascade structured modified status controller with internal pressure differential feedback and proportional gain.

3. Differences in air consumption during pneumatic positioning

With comparative studies and statistical analyses I have proved that, after a unified optimization process at typical working point, there is no difference in air consumption between cascade structured PID controller with internal pressure differential feedback and proportional gain, and cascade structured modified status controller with internal pressure differential feedback and proportional gain, in a position-controlled servopneumatic system with a statistically significance level of 95%. However, the examined cascade structured controllers have performed the positioning with significantly different, higher air consumption, such as PID controller, modified status controller and chattering-free sliding mode controller.

4. Evaluating system for comparing control algorithms from technological approach

I have developed a new complex evaluation and decision support system that allows unified, multi-faceted comparison of the different position control algorithms of the positioned pneumatic linear drives from technological approach.

The used optimization procedure is genetic algorithm combined with a local gradient method with an ITAE criterion as objective function; the complex evaluation criteria: absolute steady state error, overshoot, settling time, air consumption, response to change in external load force and response to change in target position.

5. Optimization criterion for reducing air consumption

I have developed a new optimization criterion to reduce the air consumption of the position controlled pneumatic linear drives during the positioning without structural modification of the pneumatic system. The new criterion: $Kr_{Lf}(t) = V_n * Lf + \int_0^\infty t|e(t)|dt$, where $Kr_{\mathrm{Lf}}(t)$ is the new criterion, *Vn* is the air consumption [(N)l], *Lf* is the air consumption coefficient [-], *e* is the error [mm], *t* is the time [s].

5. CONCLUSIONS AND SUGGESTIONS

During my research, I have compared five control algorithms, looking for the answer to the question of which one is best suited from them for positioning pneumatic systems. Based on the studies, I have concluded that each algorithm can be tuned to a given work point so that the accuracy of the positioning is appropriate, but in the case of a change in the working point, this accuracy could not be ensured by any of the control algorithms. There were also significant differences between the individual algorithms in energy efficiency.

In the following, that conclusions and suggestions will be described which I made during my research work and which are scientifically not relevant in their current form, but they are significant from practical viewpoint or require further investigation.

One way to reduce the air consumption of the position-controlled servopneumatic system is to reduce the leakages of the 5/3 proportional valve. Due to the spool design of the 5/3 proportional valve, there are significant gaps between the cross-sections, which can not be avoided. Even so the leakages in the direction of the exhausting ports can be minimized by use of closing valves. In this solution, 2/2 closing valves are connected to the exhaust ports of the 5/3 proportional way valve. Closing valves are expected to provide a gap-free closure that can be achieved with poppet design.

Examining the control signal-time diagrams for positioning tasks and comparing them with each other, it is noticeable that the amount and extent of control signal changes are assumed to be closely correlated with the air consumption. The physical background of this phenomenon can be well explained, as the more frequent (with higher frequency and amplitude) the 5/3 proportional way valve is switched, the more air is exhausted, increasing the air consumption. The exact explanation of this relationship can help to develop an energy-saving control algorithm or optimization criteria function.

I have previously described and evaluated the PID cascade controller's response of the changing external force. In these cases, I found that the PID cascade controller does not respond to the change of external force in the expected quality. However, in the previous chapters I have not emphasized that the 5 s test time at the PID cascade controller is not sufficient for achieving steady state. In the case of extended test time, it can be observed that the PID cascade controller slowly compensates the external force change in a much better quality than the previously mentioned values. Therefore it is worth to examine the PID cascade controller's development possibilities to improve the control accuracy of the algorithm during a shorter settling time in case of external force change.

6. SUMMARY

The primary purpose of the research was to develop an evaluation system and methodology that allows the control algorithms of the positioned pneumatic linear drives to be uniformly compared and evaluated in terms of the quality of the position control and the system's energy consumption.

In the first step I have designed and assembled the experimental servopneumatic equipment, then I have created the block-oriented mathematical model of the system. Then the model was identified and validated based on measurements. I have developed a new flow characteristic of the 5/3 proportional way valve that takes into account the effect of gap losses in all four flow cross-sections of the valve.

After that the examined control algorithms were chosen, which were the followings: proportional-integral-derivative algorithm that is most commonly used in industry; a modified state observer control algorithm and a non-linear robust algorithm: chattering-free sliding mode controller. In addition, I have examined two cascade structured control algorithms. In these two the internal control circuit contained differential pressure feedback and proportional amplification, one of the external control circuit worked with a PID controller, the other contained modified status control algorithm.

The tuning of controllers was executed by an optimization process. In order to achieve unified optimization of the various controllers, genetic algorithm was applied that was supplemented by a local gradient minimum search method for a particular positioning task. Time function weighted absolute error integral criterion (ITAE) was the optimization criteria of the procedures.

With the tuned controllers the positioning tasks were performed, the quality of the positioning and the air consumption were determined. I also examined that how each of the control algorithms respond to the changes in the load force and the target position. The results were analysed by statistical analysis and also evaluated from a technological point of view. On the basis of the results, an evaluation and decision supporting chart was outlined, which helps the user to select a control algorithm that is the most suited to the needs of the technology and its own priorities.

In addition, I have developed and evaluated a new optimization criteria function that allows to reduce air consumption during positioning without altering the structure of the servopneumatic system. Based on my research, I have defined new scientific results, made suggestions for the practical utilization of the results and finally presented the questions that require further research.

7. MOST IMPORTANT PUBLICATIONS RELATED TO THE THESIS

Referred articles in foreign languages

1. **Sárközi E.,** Földi L. (2017): Evaluation of PID-P cascade control algorithm used in positioned pneumatic drives, R&D Mechanical Engineering Letters, Vol. 15., pp. 148-158.
2. **Sárközi E.** (2016): Reduction of air consumption of positioned pneumatic drive by the optimization criteria of GA, R&D Mechanical Engineering Letters, Vol. 14., pp. 78-87.
3. Földi L.**, Sárközi E.**, Jánosi L. (2015) Positioning algorithms of pneumatic cylinders, R&D Mechanical Engineering Letters, Vol. 12. pp. 50-60.
4. Földi L., Béres Z., **Sárközi E.** (2013) Pneumatic cylinder positioning system realised by using on-off solenoid valves R&D Mechanical Engineering Letters Vol. 9. pp. 48-58.
5. Földi L., Béres Z., **Sárközi E.** (2011) Novel cylinder positioning system realised by using solenoid valves Sustainable Construction & Design Vol. 3 (1) pp. 142-151.
6. Földi L., Béres Z., **Sárközi** E., Jánosi L. (2010) Novel cylinder positioning system with solenoid valves, R&D Mechanical Engineering Letters Vol. 4 pp. 151-165.
7. Földi L., **Sárközi** E., Jánosi L. (2008) Mathematical analysis of electro-rheological flow control valve, R&D Mechanical Engineering Letters, Vol. 1. pp. 107-112.

Referred articles in Hungarian language

1. **Sárközi E.**, Jánosi L. (2017) Pozíciószabályozási lehetőségek a pneumatikus rendszerekben, Mezőgazdasági Technika, 2017. év márciusi szám
2. **Sárközi E.**, Jánosi L. (2017) Pneumatikus rendszerek energiafelhasználásának csökkentési lehetőségei, Mezőgazdasági Technika, 2017. év januári szám
3. **Sárközi E.,** Jánosi L. (2008.) Különböző repcefajták olajainak összehasonlítása elemi C- és H-tartalom alapján, Gép, 59. évf. 12. sz., 37-40. old.