BUDAPEST UNIVERSITY OF TECHNOLOGY AND ECONOMICS
FACULTY OF MECHANICAL ENGINEERING

IMPROVEMENT OF DYNAMIC BEHAVIOR AND POSITIONING ACCURACY OF PNEUMATIC SYSTEMS

PhD booklet

András József Czmerk

Supervisor: Péter Korondi PhD, DSc.

Department of Mechatronics, Optics and Informatics
in Mechanical Engineering

BUDAPEST, 2015.

Nomenclature

Latin letters

A_A	[m²]	piston cross section
A_d	[m²]	piston rod cross section
i	[-]	filling ratio
l	[m]	cylinder stroke length
p_A, p_B	[Pa]	chamber pressures
p_F	[Pa]	pressure difference relative to the whole piston cross section
p_{krit}	[-]	critical pressure ratio (p_{krit} =0,528)
p_R	[Pa]	reference pressure
p_S	[Pa]	supply pressure
q_m	[kg/s]	mass flow
r	[-]	cross section ratio
T	[s]	cycle time
t	[s]	time
t_1	[s]	actuated time
u	[V]	valve control signal
V_A, V_B	[m³]	chamber volumes
x	[m]	position
$\dot{x}$	[m/s]	velocity
$\ddot{x}$	[m/s²]	acceleration

Greek letters

α^+, α^-	[-]	flow factor for charging (+), and exhausting (-)
Γ	[-]	dimensionless pressure
κ	[-]	adiabatic exponent
Ψ	[-]	flow function
Θ	[-]	charge rate
Θ_{PWM}	[-]	duty cycle of valve control signal

Introduction

The increasing of positioning accuracy of pneumatic actuating units not only at the both final positions, but also along the whole actuating track became an important research field during the last two decades of the 20th century. The investigations to improve the control algorithms of these so called “servo pneumatic” systems usually stand in the centrum of attention since the robust control of these systems is a rather complicated task because of the physical complexity of pneumatic systems. According to this, during the last couple of years many research groups are dealing with the dynamical analysis of servo pneumatic systems and are suggesting diverse algorithms to improve the positioning accuracy and the stiffness of these systems. It seems that parallel to this the analysis of the structure and the working of pneumatic actuators and the drawing of conclusions have been overshadowed, although up to date measurement equipment could support the efficiency of the investigations. Based on new measurement methods useful statements could allow to increase the positioning accuracy and the dynamic properties. Considering this not only the improvement of control quality can be the aim of the recent investigations, but also the search for the reserves inside of the pneumatic plant is a real life research task.

According to this statement the dynamical modelling of complex pneumatic systems and the clarification of the relationship between the parameters are in the centre of my research. The results of my investigations – continuously supported by measurement – show, that the state change of the inner pressure of the chambers of pneumatic cylinders mainly determines the ability of precise positioning of the servo pneumatic control. According to this the focus of my research presented in the PhD thesis is the state change of the inner pressure of pneumatic cylinders.

Pneumatic actuator systems can be divided into two groups. The two chambers of the actuator can be served by a common valve or by two separated and independently controlled valves. In the second case rather different values of pressure can be occur in both chambers. But in the first case – often used in the industrial practice as control by single servo valve – this independence can not be realised. My investigations have been focused on the single servo valve, so that I could determine the exact stiffness of the pneumatic system which provides the suitable positioning accuracy and simultaneously the resistance against the disturbing forces, that occur from the various technological processes. It becomes also clear that the suggested new method could be also effective in the compensation of the leakage of the cylinder.

Further investigations have been made in the field of cylinder control with single valve. As an important result of my investigations I could determine an explicit mathematical relation between the duty cycle of the control signal and the stationary pressure in the chambers of the actuator – which also determines the force of the piston. The mathematical relation was also interpreted for pistons with rod. The developed model allows the determination of the control signal of the valve for given desired piston force. By the modulation of the duty cycle of the valve control signal an effective interaction can be achieved without feedback ("input shaping control"). The results of my investigations offer promising help in the case of model reference and nonlinear control methods too. The control of the inner pressure of the chambers determines the positioning accuracy and the actuating force as well.

Characteristics and structure of pneumatic systems

Pneumatic systems are used for producing linear or rotational movement utilizing pressurized air, they serve as building blocks for industrial assembling machines and manipulators thanks to their many advantages. If flexible automated systems are considered, these are usually provided by servo systems from the pneumatic family. The name "servo" refers to the operation mode of the actuator which is usually the cylinder itself. Servopneumatic systems differ from pneumatics commonly used by the industry so far, that the positioning isn't restricted to the two endpoints of the stroke, but they can reach any arbitrary position.

Modelling of pneumatic systems

The two most important parts of pneumatic systems are the cylinder and valve or valves charging the chambers with pressurized air. Figure 1 shows a combination of a rodless cylinder with single proportional valve for charging and exhausting of the chambers. The position sensor (x) and the pressure sensors (p_A, p_B) provide feedback for the controller. There is a control arrangement shown on Figure 1 which is controlled in a decentralized way i.e. there is a pressure control loop inside the position control loop.

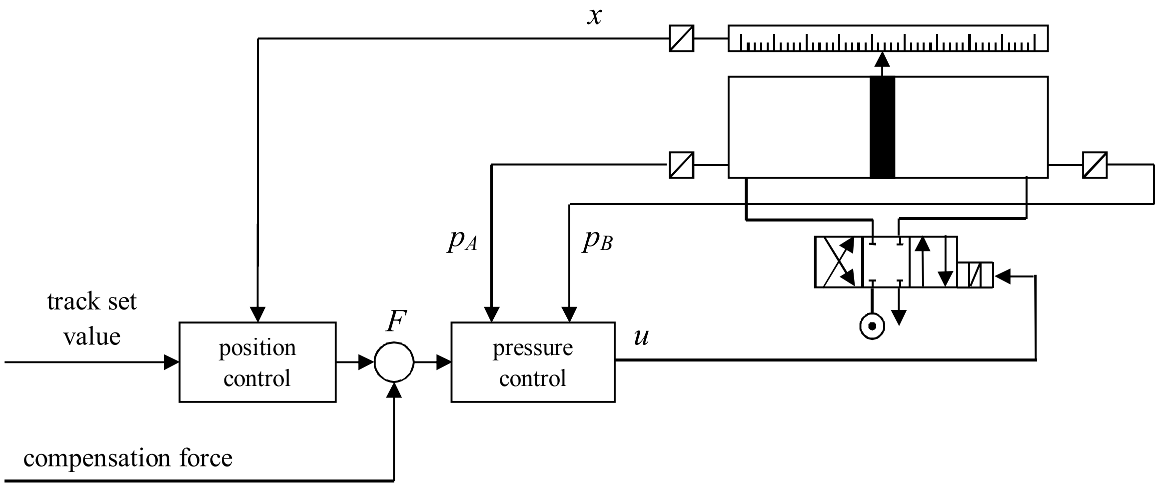

Fig. 1 Operation of a servopneumatic system with a single proportional valve

Determining the stationary pressure

The stiffness of the system plays an important role in the dynamic properties, because among other things this feature determines the resistance against forces and other disturbances when keeping a position. In contrast to mechanical propulsion systems (gearbox, toothed belt gear, worm gear, etc.) or hydraulic systems, pneumatic systems have a special feature because of the low value of the stiffness. The stationary chamber pressures are controllable by the duty cycle of the valve control signal, in the case of a relatively short duration of cycles. The chambers of the rodless pneumatic cylinders – depending on the construction of cylinder – have a leakage, the pressure drops even in closed valve position, which is disadvantage when keeping a position and/or the applied force. In a number of cases – first of all by proportional valves – the loss from valve leakage must be taken into account as well. This is of course harmful for keeping a position, since it causes pressure drop in the chambers, which is asymmetric and leads to displacement of the piston sooner or later. To avoid this, the correction of the chamber pressures by a valve operation is needed, possibly even before the undesired displacement of the piston. The pressure drop of the chambers can be effective compensate by the cyclical operation of the valve.

By single valve control of pneumatic systems – whether it is a simple electromagnetic version or proportionally actuated – during the unloaded movement of the piston, the chamber pressures fluctuate around a specific value regardless of the piston position (Fig. 2). The analysis of the operation is possible with the help of the mass flow terms.

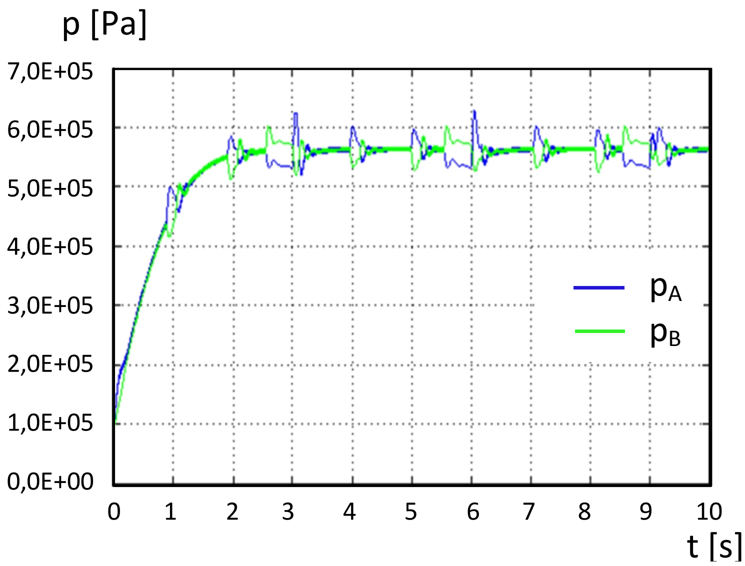

Fig. 2 Absolut pressure of the pneumatic cylinder chambers depending on the time

Determining the mass flows

Using a single valve cylinder the pressure of the chambers can not be increased or reduced at the same time. During periodic operation of a single valve the stationary pressures can be calculated. Condition of that is the amount of the material for the charge and the exhaust, i.e. the mass of the particles during upstream and downstream into the chambers (in this sense in a container), must be equal. The process for the charging is marked by "+", for the exhausting by "-". The time rates in a cycle period for the chamber are as follows, „A" *t1* for charging, *T-t1* for exhausting, and at the same time for chamber „B" *T-t1* for charging, *t1* for exhausting.

$$\mathrm{q}_{\mathrm{mA}}^{+} \cdot \mathrm{t}_1 - \mathrm{q}_{\mathrm{mA}}^{-} \cdot (\mathrm{T} - \mathrm{t}_1) = 0$$

$$q_{mB}^{+} \cdot (T - t_1) - q_{mB}^{-} \cdot t_1 = 0$$

The mass flows of the valve can be described by the ideal nozzle model. The filling ratio i as feature of the construction, and the charge rate Θ as operation dependent property are defined for the effective description of the relationship. The compound function of the flow functions in the mass flow equations makes the inverse calculation extremely difficult, thus an approximate relationship has been defined with sufficient accuracy. A dimensionless pressure has been defined for the general form Γ_A too.

Determination of duty cycle of the valve control signal as a function of the load by rodless pneumatic cylinders

In addition to the description of the system behaviour the determination of the duty cycle depending on the exerted force is important. By the backward calculation of the chamber pressures it is possible to determine the duty cycle by the charge rate

Θ for a given desired actuation force. This time by the investigation of the charge rate Θ, not the construction feature I of the valve is analysed, but the effects of the switching operation. The goal is to find a preferably explicit relation for the expression of charge rate Θ depending on the dimensionless pressure difference Γ_F. The appropriate relationship for the stationary dimensionless chamber pressures is described by tangent hyperbolic functions written in exponential form.

Determination of duty cycle of the valve control signal as a function of the load by pneumatic cylinders with cylinder rod

Cylinders equipped with a piston are more prevalent in industrial applications. The calculation of the duty cycle Θ_{PWM} for the exerted force using cylinders equipped with a piston rod can be made analogous to that of cylinders without piston rod. For the calculations the cross section ratio should be introduced, i.e. the cross-section of the piston reduced by the cross section of the piston rod relative to the cross section of the piston, by which the piston rod side chamber pressure can be converted into pressure value regarding the piston side. Using this conversion the duty cycle Θ_{PWM} can be determined by an explicit mathematical relation depending of the actuated force F.

Measurement results

The parameters of stationary dimensionless pressure equations can be clarified by the approximate equation of the ideal nozzle model for pneumatic valves actuated by periodic control signal by the measurement results. The measurements were performed on varying frequencies and on varying valve duty cycles using the proportional valve MPYE-5-1/8 LF-010B produced by the company Festo. Because of the leakage between the opening edges of the valve, air goes from the chamber with the higher pressure into the other chamber during operation. The deviation of the measured values do not lead to unusability of the derived relationship, only the parameters must be clarified (Fig. 3). The parameters of the fitted curve calculated by the least squares method are: a=0,87, b=0,60.

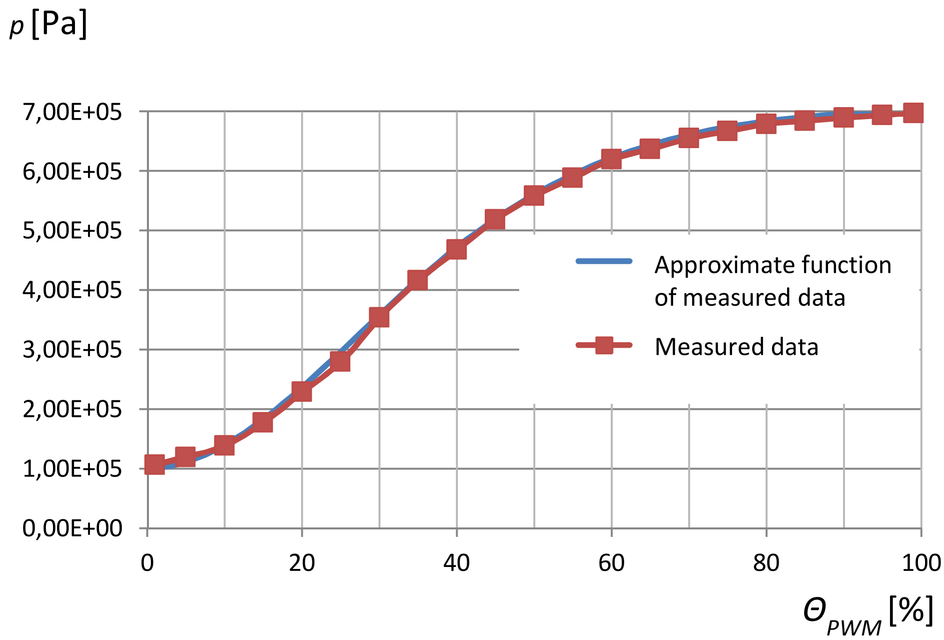

Fig 3 Measured stationary pressures, and approximate function depending on the duty cycle Θ_{PWM} (control signal frequency: f=10Hz)

Thesis Statements

Thesis 1

Using the equation of the ideal nozzle model the dimensionless stationary pressures (Γ_A, Γ_B) of the cylinder chambers can be approximated by a two-parameter explicit relationship depending on the filling ratio i (a feature of construction) and the charge rate Θ (an operation property):

$$\Gamma_A = \frac{1}{2} \cdot th[a_0 \cdot ln(\Theta \cdot i) + b_0] + \frac{1}{2}$$
$$\Gamma_B = \frac{1}{2} \cdot th\left[a_0 \cdot ln\left(\frac{1}{\Theta} \cdot i\right) + b_0\right] + \frac{1}{2}$$

where a_0 and b_0 are constants depend on the supply pressure p_S and reference pressure p_R:

$$a_0 = 1 + c \cdot (p_S - p_R)$$
$$b_0 = d \cdot (p_S - p_R)$$

and where c and d for leak-proof systems, and for valves with i=1 filling ratio are defined by the least-squares method:
c= $3{,}78 \cdot 10^{-7}$ 1[/Pa], d= $1{,}23 \cdot 10^{-6}$ [1/Pa].

The stationary cylinder chamber pressures can be expressed by the dimensionless pressures:

$$p_A = \Gamma_A \cdot (p_S - p_R) + p_R$$
$$p_B = \Gamma_B \cdot (p_S - p_R) + p_R$$

Publications related to the thesis statement are: [1-8] [12-14].

Thesis 2

The dimensionless stationary pressures (Γ_A, Γ_B) of the cylinder chambers derived in Thesis 1 can be made independent of the filling ratio *i* and the relation can be traced back to the dependence of the charge rate Θ:

$$\Gamma_A = \frac{1}{2} \cdot th[a \cdot ln(\Theta) + b] + \frac{1}{2}$$

$$\Gamma_B = \frac{1}{2} \cdot th\left[a \cdot ln\left(\frac{1}{\Theta}\right) + b\right] + \frac{1}{2}$$

where *a* and *b* are constants of the actual valve:

$$a = a_0$$

$$b = a_0 \cdot ln(i) + b_0$$

Publications related to the thesis statement are: [1-8] [12-14].

Thesis 3

For pneumatic cylinders without piston rod, the charge rate Θ which ensures the stationary pressure difference, can be expressed depending on the dimensionless pressure difference Γ_F with an explicit relationship by the next formula:

$$\Theta = \sqrt[2a]{\frac{(e^{4 \cdot b} + 1) \cdot \Gamma_F + \sqrt{(e^{4 \cdot b} - 1)^2 \cdot {\Gamma_F}^2 + 4 \cdot e^{4 \cdot b}}}{2 \cdot e^{2 \cdot b} \cdot (1 - \Gamma_F)}}$$

where Γ_F is the dimensionless pressure difference, $\Gamma_F = \Gamma_A - \Gamma_B$, and *a* and *b* are the constants of Thesis 2.

Publications related to the thesis statement are: [2] [4] [9] [11]

Thesis 4

Taking into account the different active cross sections of pneumatic cylinders equipped with a piston rod, the charge rate Θ which ensures the stationary pressure difference, can be expressed depending on the dimensionless pressure difference Γ_F with an explicit relationship by the next formula:

$$\Theta =$$

$$= \sqrt[2a]{\frac{e^{4b}(\Gamma_F + r - 1) + \Gamma_F + \sqrt{(e^{4b}(\Gamma_F + r - 1) + \Gamma_F)^2 - 4e^{4b} \cdot (\Gamma_F - 1) \cdot (\Gamma_F + r)}}{2e^{2 \cdot b} \cdot (1 - \Gamma_F)}}$$

where r is the cross section ratio which expresses the difference of the whole piston cross section and the rod cross section divided by the whole piston cross section.

$$r = \frac{A_A - A_d}{A_A}$$

where A_A is the cross section of the cylinder piston, and A_d is the cross section of the piston rod, a and b are the constants of Thesis 2.

Publications related to the thesis statement are: [2] [4] [9] [11]

Publications

Journal articles

[P-1] K. Széll and A. Czmerk, "Linear identification of a servo-pneumatic system," RECENT INNOVATIONS IN MECHATRONICS, pp. 1-7, (accepted to be published in 2015.)

[P-2] A. Czmerk, "Increasing of Stiffness of Double-acting Pneumatic Cylinder," ANALECTA, pp. 1-5, 2015 (accepted to be published in 2015.)

[P-3] K. Széll, A. Czmerk and Z. Péntek, "Egy szervopneumatikus rendszer identifikációja és szabályozása távoktatáshoz," DEBRECENI MŰSZAKI KÖZLEMÉNYEK, vol. 13, no. 2, pp. 41-48, 2014.

[P-4] K. Széll, A. Czmerk and P. Korondi, "Friction with Hysteresis Loop Modeled by Tensor Product," AUTOMATIKA, vol. 55, no. 4, pp. 463-473, 2014.

[P-5] László Valenta, János Halas, András Czmerk, László Molnár ”Silicon-elastomer-based sensor for measuring tire load.” TIRE TECHNOLOGY INTERNATIONAL-ANNUAL 1: pp. 14-18. (2010)

[P-6] Molnár László, Czmerk András ”Model linearization of servopneumatic system.” MŰSZAKI SZEMLE 38: pp. 267-270. (2007)

[P-7] Molnár L, Czmerk A ”Pneumatikus munkahenger dinamikája.” MAGYAR ELEKTRONIKA 2006:(12) pp. 39-40. (2006)

[P-8] A. Huba, L. Molnár, A. Czmerk, T. Fischl ”Dynamic Analysis of Silicone Elastomers.” MATERIALS SCIENCE FORUM 473: pp. 85-90. (2005)

Conference publications

[P-9] Z. Péntek, K. Széll and A. Czmerk, "Szervopneumatikus rendszer lineáris modellidentifikációja," in Proceedings of ARES'14, Budapest, BUTE, 2014, pp. 71-75.

[P-10] A. Czmerk, K. Széll and P. Korondi, "Development of a servo-pneumatic system in distant learning," in *Proceedings of CERiS'13 - Workshop on Cognitive and Eto-Robotics in iSpace*, Budapest, BUTE Department of Mechatronics, Optics and Mechanical Engineering Informatics, 2013, pp. 50-54.

[P-11] T. Szabó, A. Czmerk ”Sliding mode control model of a nonlinear pneumatic system.” In: 53rd IWK. Ilmenau, Németország, 2008.09.08-2008.09.12. Ilmenau: pp. 415-416. (ISBN: 978-3-938843-37-6)

[P-12] Dr László Molnár, András Czmerk ”Linearization of a servopneumatic system with modified pid controller.”In: Penninger A, Kullmann L (szerk.) Proceedings of the Fifth Conference on Mechanical Engineering, GÉPÉSZET 2006. Budapest, Magyarország, 2006.05.25-2006.05.26.

[P-13] András Czmerk ”Dynamics of a servopneumatic drive.” In: VII. International PhD Workshop. Gliwice, Lengyelország, 2005.10.22-2005.10.25. pp. 343-346.

[P-14] Molnár L, Czmerk A ”A pneumatikus hajtás tulajdonságai és dinamikai modellje.” In: OGÉT 2004.: XII. Országos Gépészeti Találkozó. Csíksomlyó, Románia, 2004.04.22-2004.04.25. Kolozsvár: Erdélyi Magyar Műszaki Tudományos Társaság, pp. 208-212.