

Investigation of Dynamic Behaviour of Nonlinear Electropneumatic Acting System

V. Geleževičius, A. Grigaitis

Dept. of Control Technology, Kaunas University of Technology,

Studentų str. 48, LT-51367 Kaunas, Lithuania; phone: +370 37 451610, e-mail: vilius.gelezevicius@ktu.lt

Introduction

Electro-pneumatic actuators are widely used in industrial automation systems, such as robotics, handling devices, packing machines and so on. The main advantages of these actuators are low costs, high rapidity and power/weight ratio. Up to now, mostly adequate applications are concerned with control tasks “from end to end” do not requiring high position and force accuracy assurance. The control of such pneumatic actuators is realized using binary electropneumatic directional control valves without any position control elements. The main desirability of such technique is that the dynamical features of pneumatic-mechanical power conversion do not manifest they self a lot on final quality of control process.

The field of applications of electropneumatic acting system can be considerably enlarged applying proportional control principle by using proportional directional control valves and feedback technology. However, in this case the nonlinear behavior of electropneumatic control channel has to be taken into account. Most contributions in this context focus on position control. Feedback linearization as control design method is broadly used [1] and since in this case the relative degree is lower than the system order, the cylinder chamber pressure [2] has to be measured or the zero dynamics have to be compensated by feed forward compensation [3]. As an alternative for control of pneumatic actuator, fuzzy methods [4], neural networks, and genetic algorithms [5] have been suggested. Linearization and linearization along reference trajectories [6] complete the control methods. Robust control approaches extend the applied control techniques [7]. Fewer contributions focus on the force control [8, 9]. However, no matter whether position or force control is considered, the dominant nonlinearities are in the pneumatic part, and not in the mechanical part of the system. There is a fundamental need for a detailed modeling of the dynamic behavior of the pneumatic actuator system.

The non-linear model of electro-pneumatic acting system, consisting of proportional directional valve and

symmetric rodless pneumatic cylinder is developed and investigated in this paper. Proposed model enables to evaluate influence of essential nonlinearities concerned with working characteristics of airflow regulating valves and pneumatic cylinders on dynamics of the system. The modeling results of electro-pneumatic acting system under several initial working conditions are presented and discussed in this article.

Model of an electro-pneumatic actuator discussion

A typical electro-pneumatic system includes a force element (the pneumatic cylinder), a command device (electro-pneumatic valve), connecting tubes, and position, pressure or force sensors. The external load consists of the mass of external mechanical elements connected to the piston and perhaps a force produced by environmental interaction. A schematic representation of electro-pneumatic acting system is shown in Fig. 1.

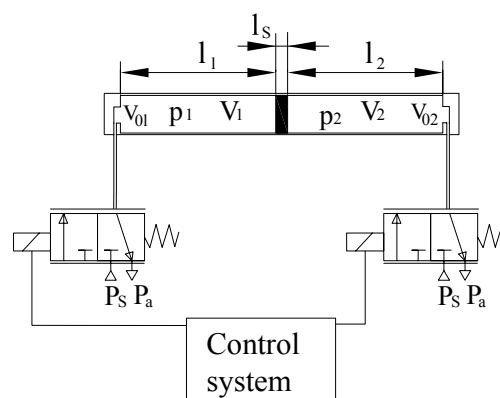


Fig. 1. Schematic representation of the pneumatic cylinder and electro-pneumatic valve system

A mathematical model was built up according to ideal gas law. Assuming that: the gas is perfect, the pressures and temperature within the chamber are homogeneous and

kinetic and potential energy terms are negligible, the ideal gas law can be written as:

$$P = \frac{m \cdot R \cdot T}{\eta \cdot V}; \quad (1)$$

where R – the ideal gas constant, $R=8,31 \frac{Nm}{mol \cdot laipsn}$; m – mass, kg ; T – temperature, $^{\circ}K$; η – mass of the gas of mole, kg ; V – volume of the gas, m^3 . T , η , and R – is constant, so pressure in the pneumatic cylinder chamber depends on mass flow rate and the volume of the chamber.

Dynamical process in the pneumatic system depends on these parameters: supply pressure p_s , pressure in the first chamber p_1 , pressure in the second chamber p_2 , exhaust pressure p_a , starting position of the piston, force of the resistance F_p , leakage between cylinder chambers and orifice area of the valve. After the control signal was applied to the control solenoid of the valve, one chamber of the cylinder becomes connected to the pressure line and the compressed air flows inward. The other chamber simultaneously becomes connected to the atmosphere through the exhaust path, and the air flows outward. The balance of the static ($p_1 = p_2$) becomes discomfit. Difference of the pressures $\Delta p = p_1 - p_2$ generates force F_c on the piston surface, depending on the effective cross section area S . When force $F_c > F_p$, piston starts moving and varying volumes of the cylinder chambers. This will guide to additional pressure changes in the cylinder chambers.

When supplying air to chamber the mass flow rate, m , through an orifice can be written as [10, 11]:

$$\frac{dm}{dt} = C_q \cdot C_m \cdot A_d \cdot \frac{p_{in}}{\sqrt{T}}; \quad (2)$$

where C_q – orifice discharge coefficient $C_q = 0,7$ (poppet valve); A_d – orifice area, m^2 ; T – supply temperature, $^{\circ}K$; p_{in} – is supply pressure, bar ; C_m – mass flow coefficient which can be written as:

$$C_m = 0.1562 \sqrt{\left(\frac{p_{is}}{p_{in}}\right)^{\frac{2}{\gamma}} - \left(\frac{p_{is}}{p_{in}}\right)^{\frac{\gamma+1}{\gamma}}}, \quad (3)$$

when $\frac{p_{is}}{p_{in}} < 0,528$ – air flow is bigger then sonic flow

$$C_m = 0,04042;$$

when $\frac{p_{is}}{p_{in}} > 0,528$ – air flow is smaller then sonic flow;

$$p_{in}$$

where: γ – is the specific heat ratio, $\gamma = 1,4$.

The dynamics of the pneumatic cylinder can be defined by the following equations:

$$\begin{cases} p_{1,2}(t) = \frac{RT}{\eta} \int \left(\frac{dm_{1,2}}{dt} \right) dt; \\ F_c(t) = [\pm p_1(t) \mp p_2(t)]S; \\ v_c(t) = \frac{1}{m} \int (F_c(t) \mp F_a(t)) dt; \\ V_{1,2}(t) = [l_{0(1,2)} \pm \Delta l(t)]S, \end{cases} \quad (5)$$

where $p_{1,2}$ – pressures in the cylinder chambers, Pa ; F_c – force of the cylinder, N ; v_c – speed of the piston, m/s ; l – displacement, m ; $V_{1,2}$ – volumes of the cylinder chambers, m^3 .

These equations clearly demonstrate essential nonlinearities of power conversion process in the pneumatic cylinder. Supposing electro-pneumatic directional valve being ideal, on the base above given equations the structural model of electro-pneumatic acting system is developed (Fig. 2).

There are three interdependent loops: supplying air in to the first chamber, air flow outward from the second chamber and air-mechanical conversion of the energy in this model.

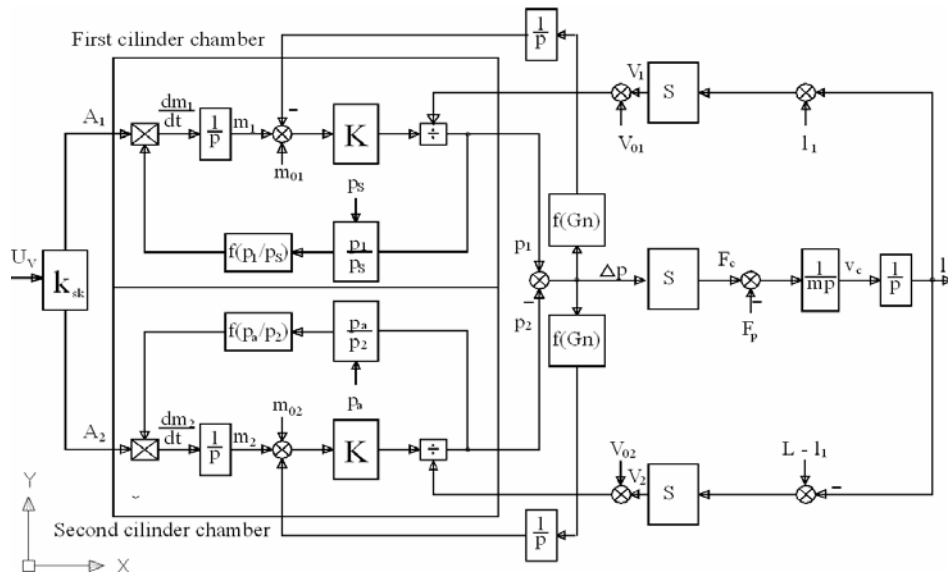


Fig. 2. Structural model of the pneumatic cylinder

The piston movement quality depends on the leakage between cylinder chambers.

Air leakage can be appreciated with extra negative loop according to the difference of the pressure Δp .

$$G_n = \frac{1}{24} \cdot \frac{d^3}{l} \cdot \frac{p_1^2 - p_2^2}{\mu RT} \cdot P, \quad (7)$$

where: d – interstice height; l – interstice length; $P=2\pi R_C$ – perimeter.

Modeling of the dynamics of pneumatic actuator

The following restraints are taken into account during of the development of the electropneumatic actuator model:

– piston has limited in motion. ($l_0 \leq l \leq L$) so, we can write these conditions:

$$v_s = 0 \text{ and } \frac{dv}{dt} = 0;$$

when $l = l_0$ and $p_1 A_1 - p_2 A_2 - F_p < 0$ or when $l = L$ and $p_1 A_1 - p_2 A_2 - F_p > 0$.

Using this model the following situations were modeled and investigated:

- Traditional control “from end to end” control mode. In this case the primary pressure in the first cylinder chamber is 1 bar and in the second one – 6 bar . The piston starting position is the left marginal one and the end position – the right marginal one;
- An influence of cylinder piston initial position on force generation process dynamics (cylinder piston in the start moment can be in separate initial positions). The start pressure of the both cylinder chambers is the same and equal to 6 bar ;
- An influence of initial pressure in the cylinder chambers on force generation process dynamics (the piston start position is equal to the middle piston stroke);
- The cylinder piston stopping in the intermediate position process by applying 6 bar pressure to the exhaust chamber of the cylinder.

The results of control mode “from end to end” modeling are given investigation are given in Fig. 3a and 3b. In the Fig. 3a the pressure changes of the first chamber – p_{in} and of the second one – p_{is} are presented. In the Fig. 3b the diagrams of cylinder generated force F_C and piston velocity v_c change are given. These diagrams affirm functionality of the model; obtained results are coincident with results obtained of the other authors [10, 11, 12] and demonstrate the satisfactory quality of the process.

An influence of initial cylinder piston position on process quality is demonstrated in the diagrams given in Fig. 4 and 5. The starting position of piston of the 0.3 m length cylinder was taken equal to 0,2 m (modeling results are given in Fig. 4) and 0,1 m (modeling results are given in fig.5). The initial pressure in the both cylinder chambers was 6 bar . Comparison of pressure diagrams given in 4a

and 5a diagrams and force and velocity diagrams given in 4b and 5b diagrams clearly demonstrate that pressure, force and velocity oscillations amplitude grows visibly when starting position of cylinder piston enlarges.

An influence of initial pressure in working chambers of the pneumatic cylinder was investigated in the case when initial piston position was fixed in the middle of the stroke of cylinder and initial pressure of the working chambers was taken equal to 1 bar (modeling results are given in Fig. 6) and to 6 bar (modeling results are given in Fig. 7). This investigation has demonstrated remarkable influence of initial pressure in working chambers of the cylinder on force generation process quality. The pressure in chambers of the cylinder plays damping role and diminishes amplitude of arisen oscillations.

The last investigations of dynamical behavior of electropneumatic acting system was carried out by stopping piston of he cylinder in he intermediate position by applying pressured air of working pressure (6 bar) to the exhaust chamber of the cylinder. The modeling results indicate on the generation slowly fading out vibrations of pressure in the cylinder chambers (modeling results are given in Fig. 8a) and force and velocity (modeling results are given in Fig. 8b). This is one of essential factors complicating development of the proportional position control systems on the base of electropneumatic acting devices.

The modeling results have been obtained using the modeling software MATLAB. Mathematical model was built up according to real pneumatic cylinder and valve. Equipment specifications are given in the table 1.

Table 1. Equipment Specifications

Pneumatic cylinder	Piston diameter: 25mm Stroke:300mm
Electro-pneumatic proportional directional control valve MPYE-5-1/8	Pressure range:0-10bar Flow rate:100-2000l/min

Conclusions

- The model of electro-pneumatic acting system, consisting of proportional directional control valve and symmetric rodless pneumatic cylinder considering essential nonlinearities of pressured air supply tract and pneumatic-mechanical power conversion process is developed and investigated in this paper.
- An influence of starting conditions such as initial piston of the cylinder position and initial pressure in the working chambers of cylinder were investigated. Modeling results demonstrate remarkable dependence force generation process in pneumatic cylinder on initial piston position and pressure in working chambers.
- On the base of obtained results the adaptive method of pneumatic cylinder force control can be recommended.

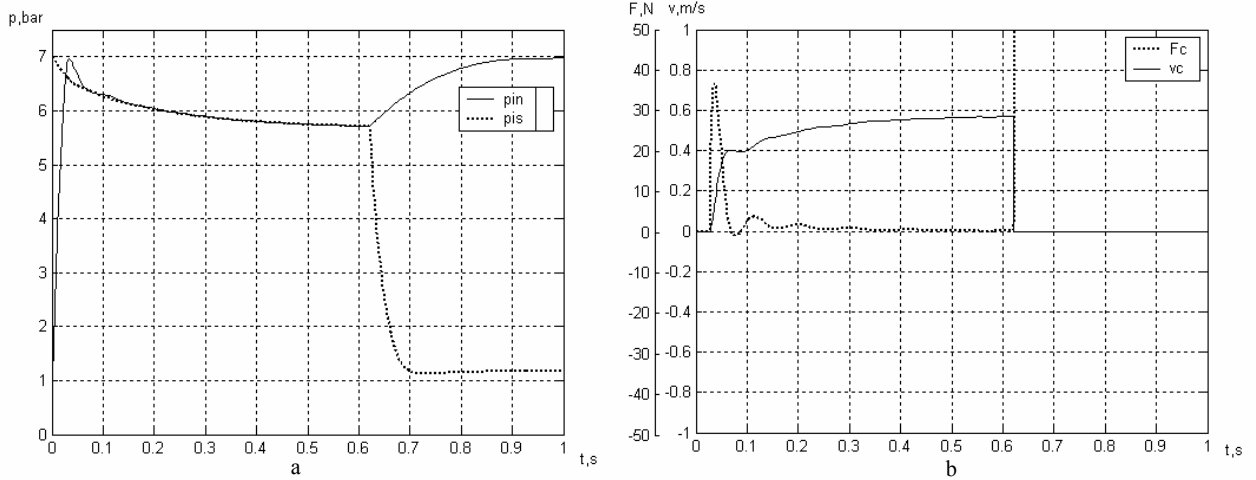


Fig. 3. The pressure change in cylinder chambers (a) and force and velocity (b) diagrams in the control “from end to end” mode

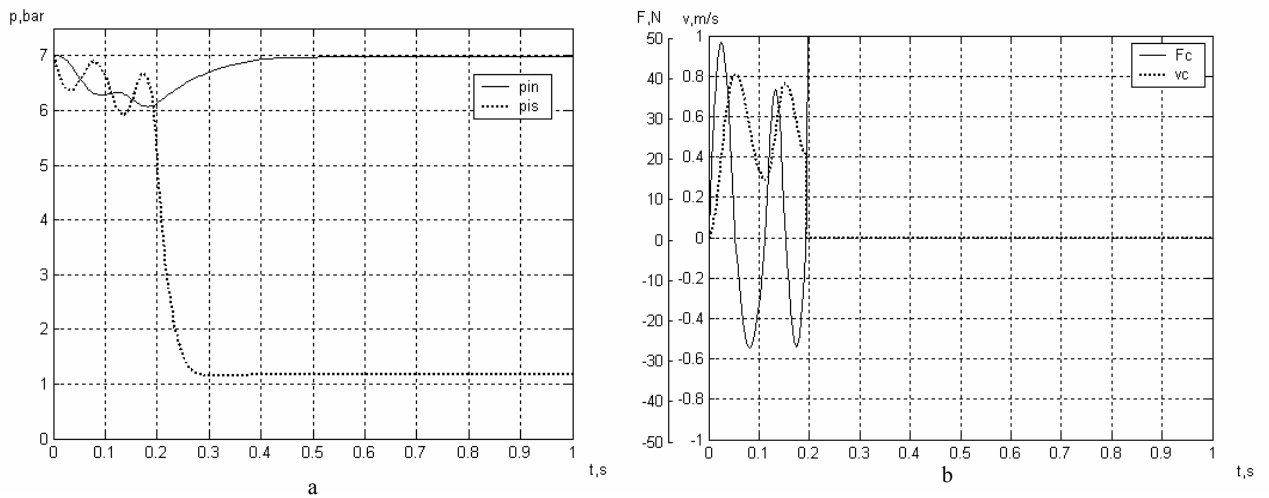


Fig.4. The pressure change in cylinder chambers (a) and force and velocity (b) diagrams in the cylinder piston start position being equal 0,2 m; The initial pressure in both chambers is equal 6 bar

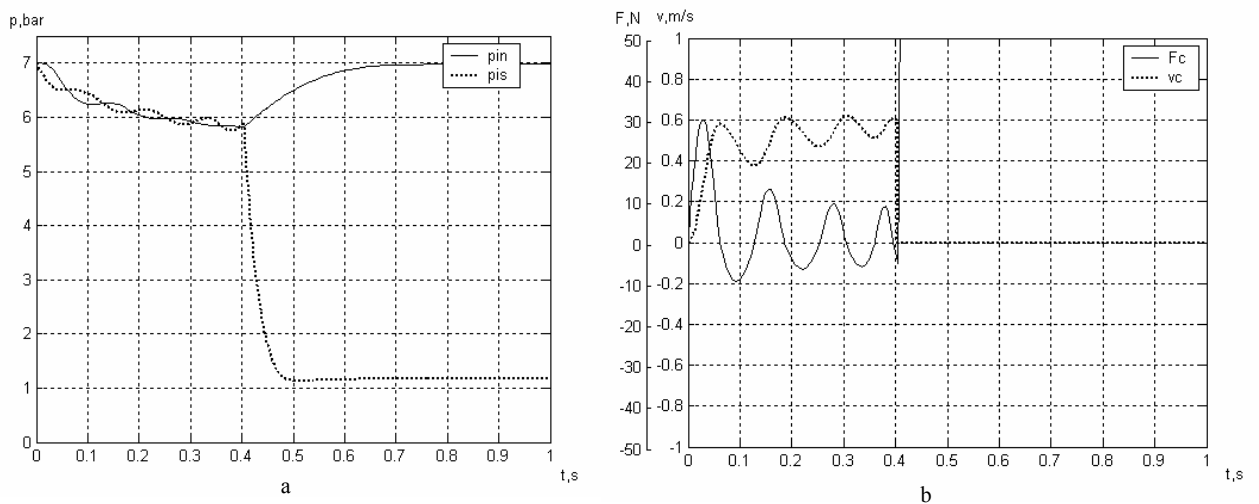


Fig.5. The pressure change in cylinder chambers (a) and force and velocity (b) diagrams in the cylinder piston start position being equal 0,1 m; The initial pressure in both chambers is equal 6 bar

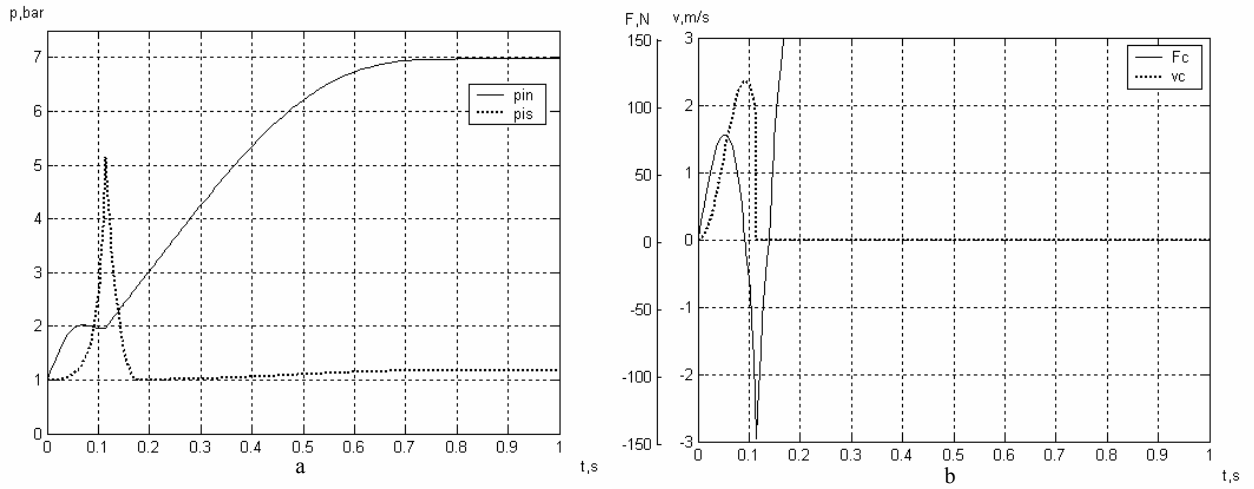


Fig.6. The pressure change in cylinder chambers (a) and force and velocity (b) diagrams in the cylinder piston start position being equal to middle stroke. The initial pressure in both chambers is equal 1 bar

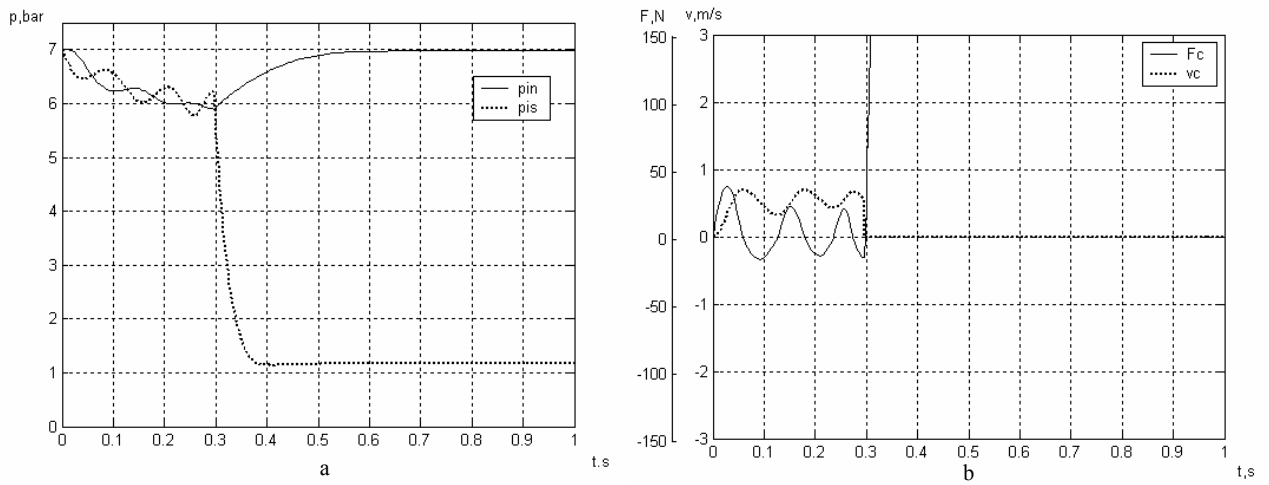


Fig. 7. The pressure change in cylinder chambers (a) and force and velocity (b) diagrams in the cylinder piston start position being equal to middle stroke. The initial pressure in both chambers is equal 6 bar

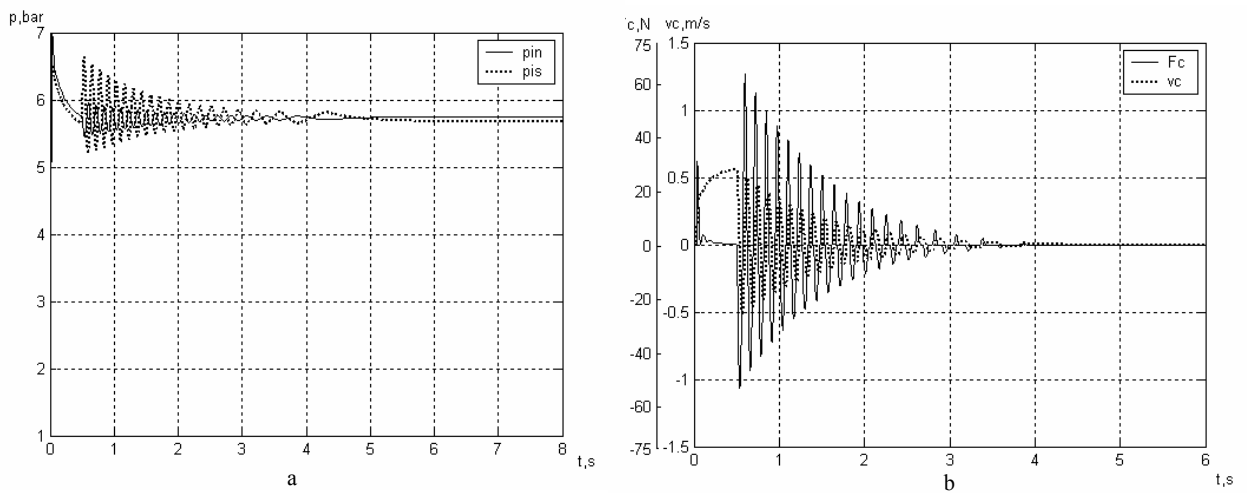


Fig. 8. Cylinder piston stopping in the intermediate position process

References

1. **Xiang and Wikander J.** Block-oriented approximate feedback linearization for control of pneumatic actuator system. *Control engineering practice*, 12(4):387-399,204.
2. **Richardson R., Plummer A.R.** Modeling and simulation of pneumatic cylinders for a physiotherapy robot // School of Mechanical Engineering, University of Leeds, UK. - [was looked 2004-02-03]. Access to the internet: <<http://www.future-motion.co.uk/modelling.pdf>>.
3. **Sawodny O. and Hildebrandt A.** Aspects of the control of differential pneumatic cylinders .In E. Shimemura and M. Fujita, editors, *Proc. of German-Japanese Seminar*, pages 247–256, Noto Hanto, 2002.
4. **Llagostera M.** Control of a pneumatic servo system using fuzzy logic, 2000, 189-201 p.
5. **Tanaka, Y. Yamada, T. Satoh, A. Uchibori, and S. Uchikado.** Model referent adaptive control with multi rate neural network for electro pneumatic servo system. In *Proc. of IEEE Conf. on Control Applications (CCA)*. – P. 1716–1721, Hawaii, 1999.
6. **Jeon Y.-S., Lee C.-O., and Hong Y.-S.** Optimization of the control parameters of a pneumatic servo cylinder drive using genetic algorithms. *Control Engineering Practice*, 6(7):847–854, 1998.
7. **Wang J., Pu J., and Moore P.** A practical control strategy for servo-pneumatic actuator systems. *Control Engineering Practice*, 7(12):1483–1488, 1999.K
8. **Kimura T., Hara S., and Tomisaka T.** H-infinity control with minor feedback for a pneumatic actuator system. In *Proc. of 35th IEEE Conf. on Decision and Control (CDC)*, volume 3, pages 2365–2370, Kobe, Japan, 1996.
9. **Hildebrandt, O. Sawodny, R. Neumann, and A. Hartmann.** Cascaded tracking control concept for pneumatic muscle actuators. In *Proc. of the European Control Conference (ECC)*, CD-ROM, Cambridge, 2003.
10. **Richer E., Hurmuzlu Y.** A high performance pneumatic force actuator system. Part1 – Nonlinear mathematical model // *ASME Journal of Dynamic Systems Measurement and Control*, September 2000. – Vol. 122. – No.3. – P. 416 – 425.– [was looked 2004-02-20]. Access to the internet:<http://cyborg.seas.smu.edu/syslab/papers/pneumatic_p1.pdf>.
11. **Richer E., Hurmuzlu Y.** A high performance pneumatic force actuator system. Part2 – Nonlinear mathematical model // *ASME Journal of Dynamic Systems Measurement and Control*, February 2001. – Vol. 122. – No.3. –P. 426 – 434. - [was looked 2004-02-20]. Access to the internet: http://cyborg.seas.smu.edu/syslab/papers/pneumatic_p2.pdf.
12. **Ichmann A.** Pneumatic cylinders: modeling and feedback force-control [was looked 2005-09-20]. Access to the internet: http://cyborg.seas.smu.edu/syslab/papers/pneumatic_p2.pdf.

Presented for publication 2006 02 13

V. A. Geleževičius, A. Grigaitis. Investigation of Dynamic Behaviour of Nonlinear Electro-Pneumatic Acting System // Electronics and Electrical Engineering.– Kaunas: Technologija, 2006. – No. 4(68).– P. 35–40.

The non-linear model of electro-pneumatic acting system considering essential nonlinearities of pressured air supply tract and pneumatic-mechanical power conversion process is developed and investigated in this paper. An influence of starting conditions such as initial position of piston of the cylinder and initial pressure in the working chambers of cylinder are investigated. Modeling results demonstrating remarkable dependence of the force generation of process in the pneumatic cylinder on initial piston position and pressure in working chambers are presented. Il. 8, bibl. 12 (in English; summaries in English, Russian and Lithuanian).

В. А. Гяляжявичус, А. Григайтис. Исследование динамических характеристик нелинейной электропневматической исполнительской системы // Электроника и электротехника – Каунас: Технология, 2006 –No. 4(68).– С. 35–40.

Представлено исследование модели нелинейной электропневматической исполнительской системы, в которой учтены основные нелинейности, связанные с зависимостью расхода воздушной массы от соотношения давлений на входе и выходе дроселирующего устройства и от пневмомеханического преобразования энергии распределителя. Исследованы влияние на динамику системы начальных условий таких, как начальное положение поршня и значения начального давления в рабочей камере. Результаты моделирования представляют зависимость развиваемого поршнем усилия от этих начальных условий. Ил. 8, библи. 12 (на английском языке; рефераты на английском, русском и литовском яз.).

V. A. Geleževičius, A. Grigaitis. Netiesinės elektrinės pneumatinės vykdyimo sistemos dinamiųjų charakteristikų tyrimas // Elektronika ir elektrotechnika. – Kaunas: Technologija, 2006 – No. 4(68). – P. 35–40.

Pateiktas netiesinės elektrinės pneumatinės vykdyimo sistemos, kurioje įvertinti esminiai netiesiškumai, susiję su oro masės debito priklausomybe nuo slėgių santykio droseliuojančio įtaiso įėjime ir išėjime ir pneumomechaniniu energijos virsmu, modelio tyrimas. Iširta pradinių sąlygų –pradinės stūmoklio padėties ir pradinių darbo kamerų slėgių – įtaka sistemos dinamikai. Modeliavimo rezultatai parodo stūmoklio generuojamos jėgos priklausomybę nuo šių pradinių sąlygų. Il. 8, bibl. 12 (anglų kalba; santraukos anglų, rusų ir lietuvių k.).