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DYNAMIC BEHAVIOR OF A PNEUMATIC SERVOMECHANISM WITH SYMMETRIC PISTON'S AREA CYLINDER VERSUS ASYMMETRIC ONES

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In the work, the dynamic behavior of an electro-pneumatic servomechanism with linear pneumatic actuator controlled by a proportional distributor is studied through numerical simulations. The pneumatic actuator is considered in two constructive variants: with symmetrical piston and asymmetrical piston. The response of the two types of servomechanisms is compared to the same step signal considered as the reference. The servomechanism with an asymmetric actuator is faster than the symmetrical one.

Keywords: Asymmetric actuator, electro-pneumatic servomechanism, numerical simulation, pneumatic proportional control valves.

1. Introduction

Electro-pneumatic positioning servomechanisms have developed and improved their performance continuously so that they can be used in various applications. So are used in the military industry, food industry, pharmaceutical and medical equipment industry, automotive industry, and manufacturing industry. Practically, there is no industrial field with automated processes in which such systems do not find applications [1-3].

The performances of electropneumatic servomechanisms recommend these systems. Compared to hydraulic motors where the power-to-weight ratio is 0.5 to 1 kW/kg, pneumatic motors have a power-to-weight ratio of 0.3 to 0.4 kW/kg, which is much higher than electric motors, which have the ratio of 0.03 to 0.1 kW/kg [1]. Even in the case of the payload-to-weight ratio, pneumatic motors have a ratio of 11 compared to 3.5 N/kg, corresponding to electric motors. For hydraulic motors, the ratio between payload and weight is 20 N/kg [1].

The structure of an electro-pneumatic servomechanism generally consists of the pneumatic actuator (linear or rotary pneumatic motor), an electro-pneumatic amplifier for controlling the flow required to move the actuator, a position transducer (linear or angular) for closing the position adjustment loop (the

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feedback reaction) and an electronic device, the controller, for comparing the reference size in the system (input size) with the output size measured by the transducer as displacement of the actuator [4]. In addition, by connecting the parameters of the controller, the dynamic behavior of the system is adjusted to ensure its stability and the desired performances.

Usually, the electro-pneumatic amplifier used to command and control the actuator is a pneumatic servovalve [1-3]. The high purchase price of this type of pneumatic amplifier has encouraged the development of the equipment and control techniques [5-11] so that to approach the performance of the new pneumatic devices to servo valve performances. Thus, the development of the proportional technique [12-14] determined the use of proportional pneumatic distributors and even on-off solenoidal pneumatic distributors with PWM control (Pulse Width Modulation) in the structure of positioning servomechanisms [15], [16].

The purpose of the work is to compare the dynamic behavior of a linear pneumatic actuator having the piston with equal areas (called symmetric actuator) with the dynamics of a piston actuator with unequal areas (asymmetric actuator). The actuator, in both situations, is part of the structure of an electro-pneumatic positioning servomechanism and is controlled by the same type of proportional pneumatic control valve. This comparison is useful for the performance of each type of servo according to the technical application.

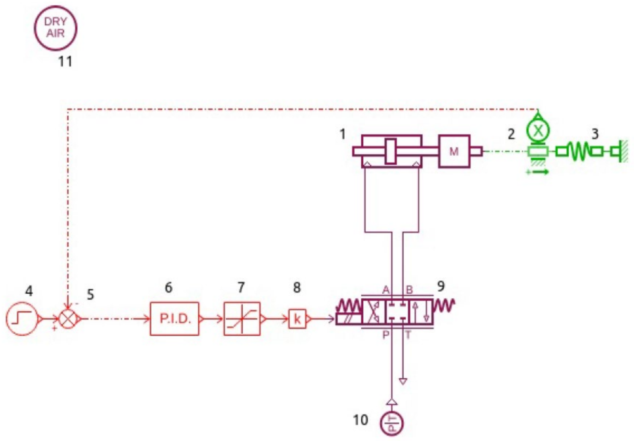
The paper presents the work methodology, the numerical simulation scheme and the obtained results.

2. Methodology

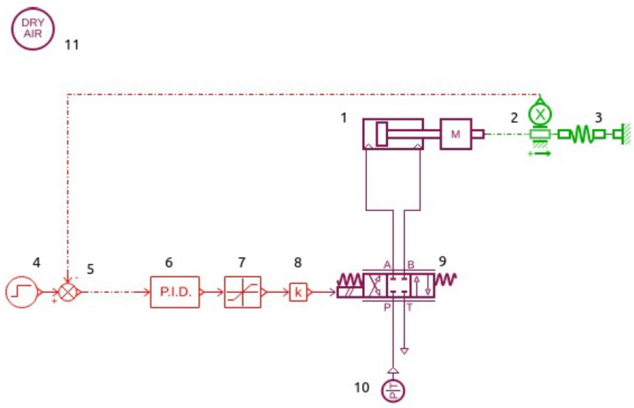
It was considered the driving scheme of a linear positioning servomechanism in which the pneumatic actuator is used in two constructive variants: actuator with equal piston areas and actuator with unequal piston areas. Using the work facilities offered by the Simcenter Amesim technical systems numerical simulation program, the dynamic behavior of the servomechanism considered in the two variants was studied by numerical simulation.

The Simcenter Amesim simulation environment (currently developed by Siemens https://plm.sw.siemens.com/en-US/simcenter/) provides a simulation platform for complex technical systems for virtual testing of their operation and performance [17]. Thus, the hydraulic and pneumatic driving schemes are numerically modeled based on the software blocks pre-generated by the program with the related device connections modeled according to the driving scheme. The scheme built in the Simcenter Amesim simulation program is identical to the graphical representation of the drive scheme. The predefined blocks are grouped into program libraries in the fields of mechanical, electrical, thermal, hydraulic, according to the engineering applications studied. The blocks and their connections are built based on the mathematical models that define the physical behavior of each technical element in the studied system. The programming language used is C.

In Fig. 1 is presented the driving pneumatic scheme of the servomechanism in both variants: actuator with equal piston areas (Fig. 1.a) and actuator with unequal piston areas (Fig. 1.b).



a)



b)

Fig. 1. Operation and numerical simulation scheme: 1-pneumatic cylinder (symmetric (a); asymmetric (b)) 2-displacement sensor; 3-resistive force; 4-command signal; 5-summing; 6-PID; 7-block saturation; 8- constant block; 9- proportional distributor; 10-gas supply with a pressure of 7barA; 11- type of gas used

The equipment parameters in the numerical simulation scheme have the parameters specified below.

The double pneumatic piston with symmetric areas piston (block 1 of Fig. 1.a) has a length stroke of 400 mm, a piston diameter of 80 mm and a rod diameter of 25 mm. The area on the charged stroke is 4535.67 mm^2, the same as on the uncharged stroke. The air temperature at the inlet port is the same as the outlet port of 293.15 K. The mass moved by the piston is 30 kg. It is not considered the coulomb friction force and the viscous friction coefficient is 100 N/(m/s). There are no leakage losses of air.

The double pneumatic piston with asymmetric areas piston (block 1 of Fig. 1.b) has the same parameters as described above, but on the charged stroke, the area is 5026.55 mm^2 and when the piston is uncharged, the area is 4535.67 mm^2.

The displacement sensor (block 2) has a gain coefficient of 25 V/m.

The perturbation force considered is an elastic force (block 3) with the elastic constant of 5000 N/m. Block 4 is the reference with the step size. Block 5 is a summing block, which compares the reference size with the size of the displacement measured by the transducer. The proportional integral derivative (PID) controller (block 6) has a proportional gain K_p= 1 and no integral gain and derivative gain. The saturation element (block 7) has a minimum permitted value of -10 V and a maximum value of 10V, and so the domain of the electric proportional pneumatic control valve is limited to [-10, 10] V, values recommended by the producer [18]. Block 8 is an amplifier with gain – 1 to change the command sign of the pneumatic proportional control valve with the symbol of Block 9. The pneumatic proportional control valve is assumed to be a 4/3 closed center valve, which is similar with 5/3, closed center pneumatic control valve [18].

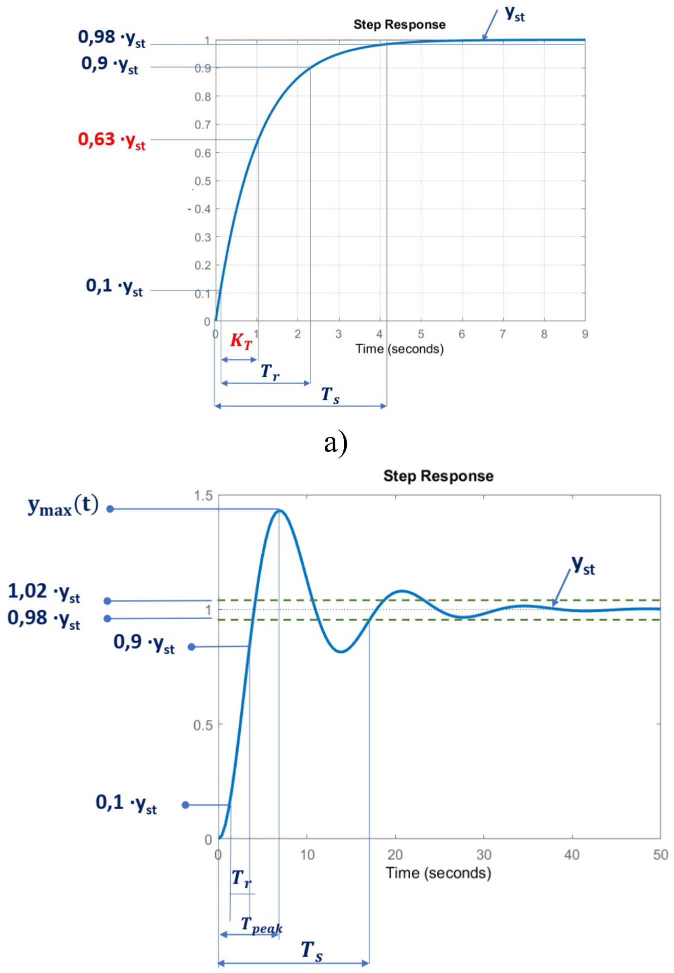
The valve natural frequency is considered to 70 Hz and a damping ratio of 0.8. The command orifice of the valve has an area of 7 mm^2 and is the same for all 4 ports. The flow coefficient of the valve is 0.72. The pneumatic pressure source (Block 10) has 7 barA and a temperature of 293.15 K. Block 11 defines the air properties. The air is a perfect gas with the density of 1.18 kg/m^3 for an operation temperature of 300 K and an absolute pressure of 1.013 barA. The absolute viscosity is $18.552 \cdot 10^{-6}$ Pa·s. Thermal conductivity is 0.026156 W/m/K, and the specific heat at constant pressure is 1004.815 J/kg/K.

All the specifications of the mentioned blocks from Fig. 1 are required and defined by the numerical simulation program used. For the two types of actuators used, the servomechanism elements remain the same, including the type of PID (Proportional Integrative Derivative) controller used (block 6), reference size (block 4) and the perturbation force (block 3).

3. Numerical simulations and results

Using the numerical simulation scheme (like the pneumatic driving system) in Fig. 1, for a step signal of 1 V, the behavior of the electro-pneumatic servomechanism was analyzed and compared in the two constructive variants.

Because the electro-pneumatic servomechanism is a control system the comparative results refer to specific sizes of the first and second order control systems regarding their response to a step signal. Figure 2 shows these sizes with the mention that K_T is the time constant of the response *y(t)*, T_r is the rise time and T_s is the settling time. Also, y_{st} is the steady-state response system.



b)

Fig. 2. Control system response to a unit step: a) first-order transfer function system; b) second-order transfer function syste.

The simulation results for a 1 V step input are given in figures 3 and 4, where rod displacement, velocity, acceleration, pressures at the actuator ports and the positioning error are plotted for both studied cases.

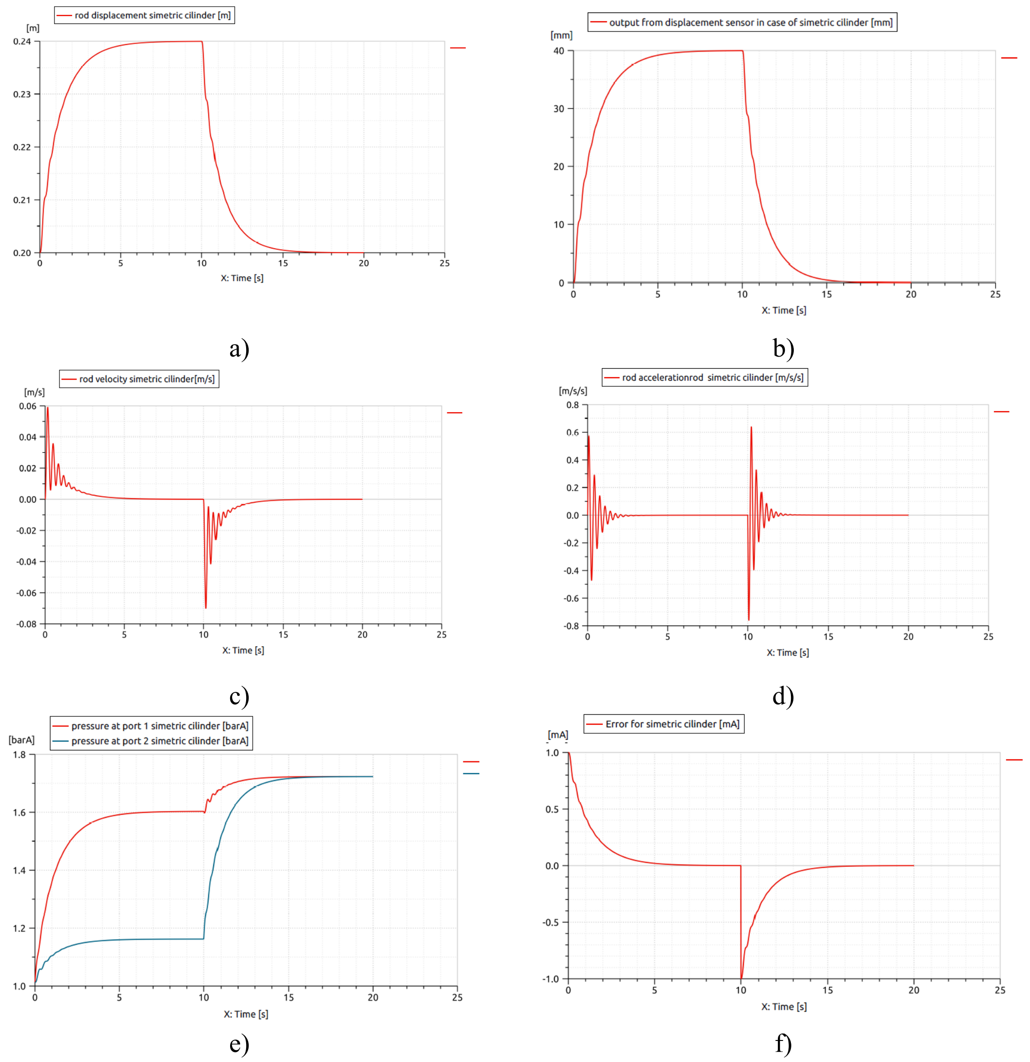


Fig. 3. Behavior of the pneumatic servomechanism having the actuator with equal piston areas: a) rod displacement; b) displacement sensor output c) rod velocity; d) rod acceleration; e) pressures at the actuator ports; f) positioning error.

It is considered that for the constant of the displacement transducer, at the step of 1 V, the displacement of the piston is 40 mm. Also, the two types of actuators will achieve the same movement, but the symmetrical actuator starts its movement from the middle of the stroke, while the asymmetrical actuator starts its movement from the end of the stroke.

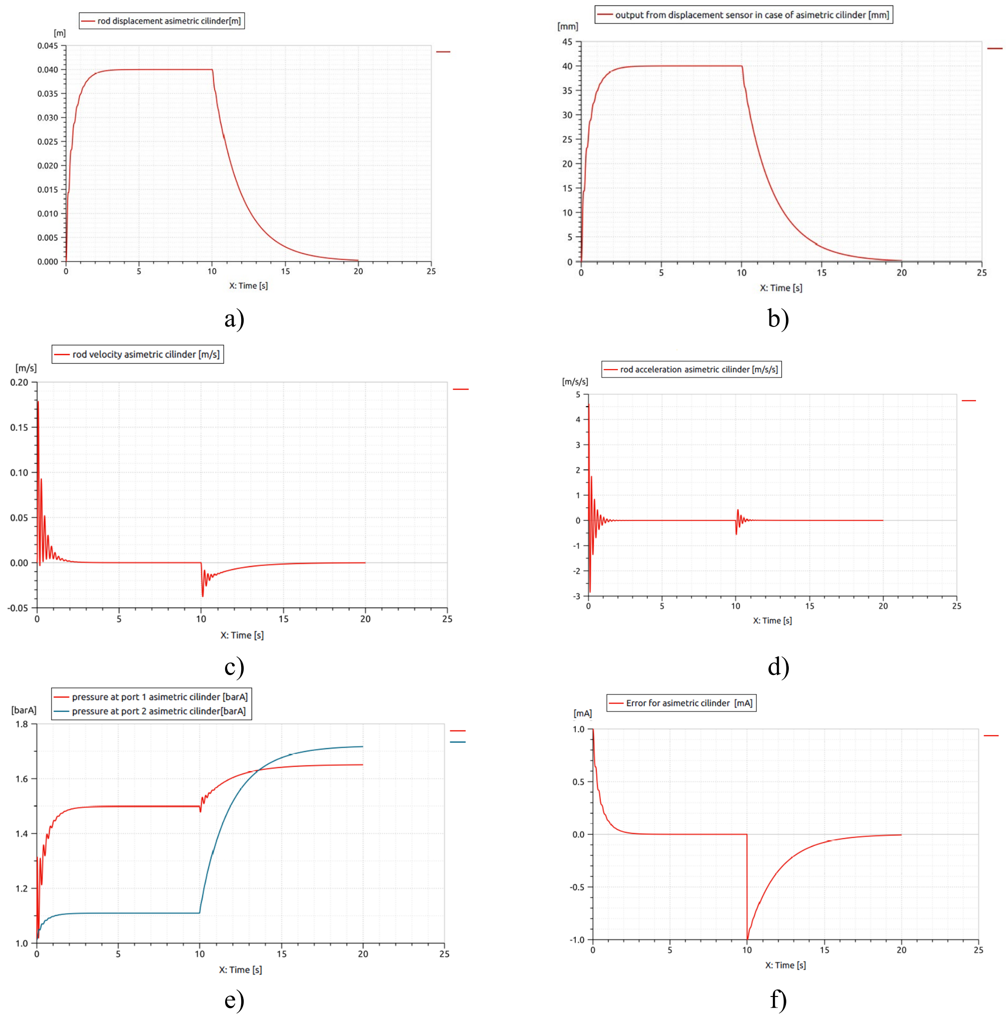


Fig. 4. Behavior of the pneumatic servomechanism having the actuator with unequal piston areas: : a) rod displacement; b) displacement sensor output c) rod velocity; d) rod acceleration; e) pressures at the actuator ports; f) positioning error.

From the above graphic representations, it can be observed that with the step signal applied as a reference value of 1 V, the servomechanism behaves as an over-damped system in the two constructive cases studied. To compare the behavior of the two types of servomechanisms, the positioning error, the rise time, and the damping time were monitored. In addition, the stationary time of the piston when changing the direction of movement with the change of commutation on the proportional pneumatic distributor was also considered, Tabel 1.

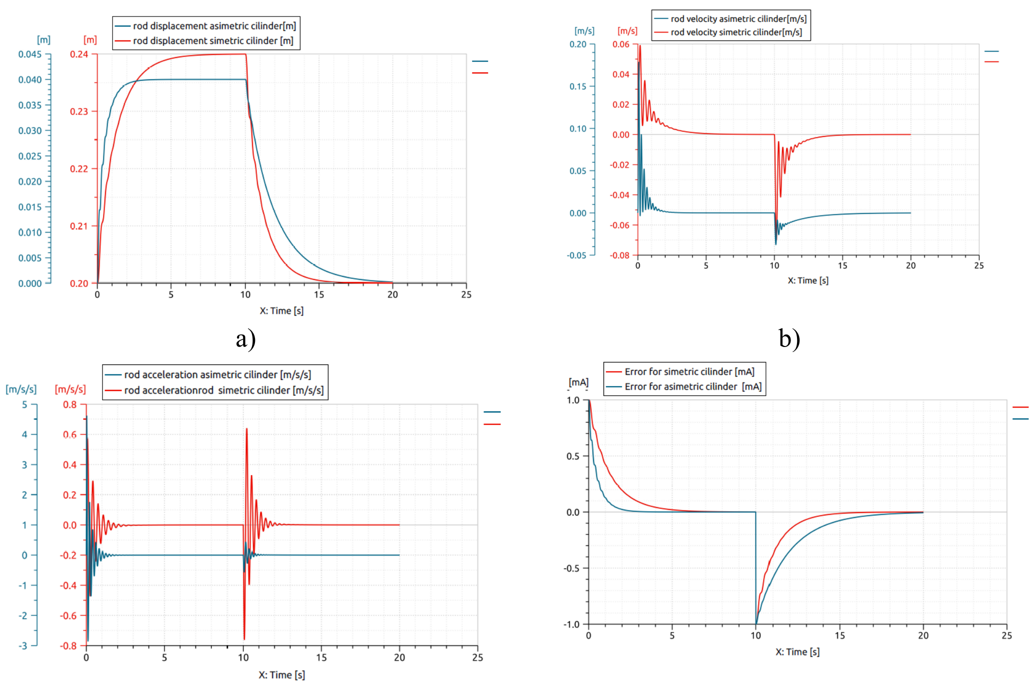
From figures 3.f and 4.f, it is observed that the positioning error is zero both for the servomechanism with symmetrical piston and in the case with asymmetrical piston.

Table 1

Dynamic behavior of step input of 1 V

	Time constant (s)	Rise Time (s)	Settling Time (s)	Error (mm)
Symmetric Actuator	1.13	2.45	4.8	0
Asymmetric Actuator	0.51	1.15	1.92	0

In Fig. 5 are placed on the same graphic the response of the servomechanisms to the 1 V step signal to better highlight the differences that appear in the dynamic behavior.



c) d)

Fig. 5. Comparison of results at 1V step signal for pneumatic servomechanism with symmetrical piston actuator vs. asymmetric piston: a) displacement; b) velocity; c) acceleration; error.

The two pistons make their displacement in accordance with 1 V input size, Fig. 3.a and Fig. 4.a, with the observation that the piston of the symmetrical actuator starts from the middle of the stroke, while the piston of the asymmetrical actuator starts from the begining of the stroke. Also, it can be observed that the rise time of the asymmetric actuator is lower than that of the symmetrical actuator, for the loading stroke Fig. 5. a. The stationary time at the end of the stroke is lower in the case of the symmetrical actuator because the rise time, in this case, is longer than for the asymmetrical actuator and the total time of maintaining the step signal is 10 s, the same for both types of servomechanisms.

The numerical simulations show that the servomechanisms have a stable dynamic behavior and good positioning performance. The response time of the servomechanism with asymmetric actuator is better than in the case of the symmetrical actuator, which is why it is recommended in the case of applications that need fast positioning servomechanisms.

4. Conclusions

In the present paper, the dynamic behavior of an electro-pneumatic servomechanism with a linear pneumatic actuator controlled by a proportional pneumatic distributor was studied through numerical simulations. The pneumatic actuator was considered in two constructive versions: with a piston having equal areas (symmetrical) and a piston having unequal areas (asymmetrical).

For the same constructive parameters specific to the system elements and for the same values of the disturbance force, the actuated mass and the input signal, the stability of the system, the positioning error and the response time were monitored. From the obtained numerical simulations, it was observed that the servomechanism is stable, the positioning error is zero and the asymmetrical piston servomechanism is faster on the loading stroke than the symmetrical one.

Considering the current trends in the design of industrial robotics and the costs of the driving equipment, it is important to develop optimal technological solutions from an economic point of view, but without diminishing the performance of the driving systems. A wider study of the structure of the error compensator should be added.

The advantages of the clean electropneumatic servosystems should be developed with some new extensions of the industrial robotics. The subject of this paper will be continued with the development of a positioning servomechanism structure with proportional pneumatic control valve for which a new type of controller, suitable for the application, will be designed by the authors.

R E F E R E N C E S

[1]. *I. L. Krivts, G. V. Krejnin*, Pneumatic actuating systems for automatic equipment: structure and design, Ed. Taylor & Francis Group, 2006.

[2]. *P. Beater*, Pneumatic Drives. System Design, Modelling and Control, Ed. Springer, 2007.

[3]. *L. Zhao, Y. Xia, H. Yang, J. Zhang,* Pneumatic Servo Systems Analysis. Control and Application in Robotic Systems, Ed. Springers, 2022.

[4]. *V. N. Cococi, C.-A. Safta, C. Călinoiu,* Parameter tuning process for a closed-loop pneumatic actuator, IOP Conf. Series. Earth and Env. Science; Bristol **Vol. 664**, Iss. 1, 2021, DOI:10.1088/1755-1315/664/1/012030.

[5]. *M. I. P. Azahar, A. Irawan, R. M. T. Raja Ismail*, Adjustable Convergence Rate Prescribed Performance with Fractional-Order PID Controller for Servo Pneumatic Actuated Robot Positioning, Cognitive Robotics, **Vol. 3**, 2023, pp 93–106, https://doi.org/10.1016/j.cogr.2023.04.004

[6]. *A. Irawan, M. S. Ramli, M. H. Sulaiman, M. I. P. Azahar, A. H. Adom*, Optimal Pneumatic Actuator Positioning and Dynamic Stability using Prescribed Performance Control with Particle Swarm Optimization: A Simulation Study, International Journal of Robotics and Control Systems, **Vol. 3**, No. 3, 2023, pp. 364-379.

[7]. *S. Ning, and G. M. Bone,* High Steady-State Accuracy Pneumatic Servo Positioning System with PVA/PV Control and Friction Compensation, Proceedings of the 2002 IEEE International Conference on Robotics and Automation, Washington, DC, pp. 2824-2829, 2002. DOI: 10.1109/ROBOT.2002.1013660

[8]. [4.10] *I. Ramírez*, Design of a tracking controller of a siso system of pneumatic servopositioning Ingeniería y Desarrollo, **Vol. 36**, no. 1, 2018, pp. 74-96.

[9]. *X. Brun, M. Belgharbi, S. Sesmat, D. Thomasset, S. Scavarda*, Control of an electropneumatic actuator: comparison between some linear and non-linear control laws. Proceedings of the Institution of Mechanical Engineers, Part I: Journal of Systems and Control Engineering, SAGE Publications, 213 (5), pp.387-406, 1999.

[10]. *M. F. Rahmat, S. N. S. Salim, N. H. Sunar, A. 'A. M. Faudzi, Z. H. Ismail and K. Huda,* Identification and non-linear control strategy for industrial pneumatic actuator, International Journal of the Physical Sciences Vol. 7(17), pp. 2565 - 2579, 23 April, 2012, DOI: 10.5897/IJPS12.030

[11]. *M. F. Rahmat, N. H Sunar, S. N. S. Salim, M. S. Z. Abidin, A. A M. Fauzi, Z. H. Ismail,* Review on Modeling and Controller Design in Pneumatic Actuator Control System, Int. J. on Smart Sensing and Intelligent Systems, **Vol. 4**, no. 4, 2011.

[12]. *F. Ning, Y. Shi, M. Cai, Y. Wang, W. Xu,* Research Progress of Related Technologies of Electric-Pneumatic Pressure Proportional Valves, *Appl. Sci.* 2017, *7*, 1074; doi:10.3390/app7101074

[13]. *B. M. Y. Nouri, M. B. Y. Saudi,* Experimental Modelling and Identification of Compressible Flow through Proportional Directional Control Valves, Universal Journal of Control and Automation 2(1): 4-13, 2014, DOI: 10.13189/ujca.2014.020102

[14]. *H-T. Lin*, A Novel Real-Time Path Servo Control of a Hardware-in-the-Loop for a Large-Stroke Asymmetric Rod-Less Pneumatic System under Variable Loads, *Sensors* 2017, *17*, 1283; doi:10.3390/s17061283.

[15]. *Z. Lin, T. Zhang, Q. Xie, Q. Wei,* Electro-pneumatic position tracking control system based on an intelligent phase-change PWM strategy, Journal of the Brazilian Society of Mechanical Sciences and Engineering (2018) 40:512, https://doi.org/10.1007/s40430-018-1431-y

[16]. *B. Najjari, S. M. Barakati, A. Mohammadi, M. J. Fotuhi, S. Farahat, and M. Bostanian,* Modelling and Controller Design of Electro-Pneumatic Actuator Based on PWM, International Journal of Robotics and Automation (IJRA), **Vol. 1**, No. 3, September 2012, pp. 125~136.

[17]. *Vasiliu N., Vasiliu D., Călinoiu C., Puhalschi R.,* Simulation of the Fluid Power Systems with Simcenter Amesim, CRC Press, Taylor & Francis, Boca Raton, FL, USA, 2018.

[18]. https://www.festo.com/ro/ro/a/161979/?q=mpye%7E%3AsortByCoreRangeAndSp2020