IOP Conference Series: Earth and Environmental Science

PAPER • **OPEN ACCESS**

Parameter tuning process for a closed-loop pneumatic actuator

To cite this article: V N Cococi *et al* 2021 *IOP Conf. Ser.: Earth Environ. Sci.* **664** 012030

IOP Conf. Series: Earth and Environmental Science **664** (2021) 012030 doi:10.1088/1755-1315/664/1/012030

Parameter tuning process for a closed-loop pneumatic actuator

V N Cococi[1], **C A Safta**[1] **and C Călinoiu**[1]

[1] University Politehnica of Bucharest, Power Engineering Faculty, 313 Splaiul Independentei, RO-060042, Bucharest, Romania

E-mail: carmen.safta@upb.ro

Abstract. Pneumatic drives are light weight, small-sizes, simple to install and to maintain. Pneumatic drives have a robust design and operation. They are clean and cheap, have no temperature limitation, and so, they are preferred in many branches of manufacturing and process industry. The well-known pneumatic actuators are widely used in closed-loop positioning systems because of their dynamic performance and mechanical flexibility in different electro-mechanical systems arrangements. The goal of the paper is to find the parameters of a closed-loop pneumatic actuator so that a good dynamic behaviour of the system is obtained. The parametric tuning process is developed by using numerical simulation technique. So, it was seen that using a proportional controller, the pneumatic actuator has an asymptotic stability for different values of the proportional gain K_p. Starting with $K_p = 8$ the system remains stable, but oscillations of low amplitude appears.

1. Introduction

In the framework of "Industry 4.0" and "cyber-physical systems" the pneumatic systems are continuously developed and changed from the "heavy metallic cylinders" to "pneutronic" systems [1-4]. The flexibility of electro-pneumatic systems, electronic embedded, digital pneumatics and new materials discovered make possible to integrate the pneumatic systems and devices in many industrial applications, bio-medical devices and even IoT (Internet of Things) technologies. So, new control algorithms were developed as microcontrollers with WIFI interface and the classic electro-pneumatic drives were developed and transformed to respond to the modern technologies [5-7].

The evolution of pneumatic control system (in open loop or closed-loop) must respond to the same nonlinearities identified in the research of these systems (as air compressibility, air-flow pressure relationship through an orifice, dead zone in the pneumatic servo valve behavior, friction force effects) [7, 8] even in the case of cyber-physical systems [2]. The performances of the pneumatic control system (accuracy of the pneumatic positioning system and stability) are difficult to be achieved in the presence of unknown disturbances and external perturbations.

For this reason, the important device in the closed-loop pneumatic actuator is the controller.

Ali *et al.* [9] reviewed the mathematical models and control methods of pneumatic actuators underlined the advanced control strategies: switching algorithm, programmable logic controller (PLC), neural network controller, fuzzy controller, adaptive controller for variable loads and uncertain disturbances, are only some examples. Usually, the controller design starts with the mathematical modelling of the pneumatic control system, numerical simulations (using specific procedures or commercial software) to find the pneumatic control system performances and experimental validation.

The complexity of pneumatic control systems requires simple models and modern computer technique to replace the old testing prototypes procedure used in industry [10].

In nowadays, the modern engineering methods of research as Model-in-the-Loop (MiL), Software-in-the-loop (SiL) and Hardware-in-the-Loop (HiL) help us to shorten the path between conception/design and prototype testing. LMS Imagine.Lab is such a platform. Simcenter Amesim, as part of the numerical platform, gives to the engineer opportunity to work with components that have the numerical procedures validated [11].

The aim of the paper is to design the controller of a continuous positioning pneumatic actuator in closed-loop. Using numerical simulation technique, it is studied the performances and the stability behavior of the linear pneumatic actuator controlled by a proportional valve and a proportional controller. The limits of controller proportional gain K_p for different step inputs are found. The stability of the pneumatic actuator is studied to different types of inputs and parameters.

2. Description of the pneumatic actuator in closed-loop

In paper [12] we demonstrate that a pneumatic proportional valve (PPV) used in an open-loop pneumatic actuator positioning system can be controlled using different electrical signal inputs. For this reason, the on/off solenoid valves can be replaced with PPV even if the position-controlled pneumatic actuator with pulse width modulation (PWM) algorithm applied to solenoids valves has a more reasonable cost than servo valves or proportional valves [13].

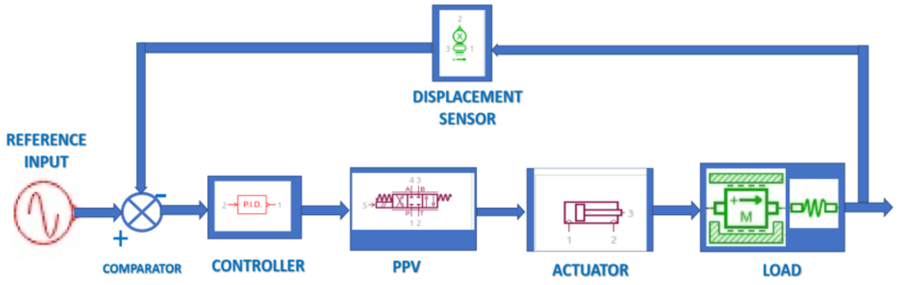
As a continuation of the pneumatic actuator in open-loop with results disseminated [12] we considered a double-acting pneumatic cylinder with the length stroke 400 mm and a proportional pneumatic valve (PPV) in closed-loop. The pneumatic actuator has the piston diameter 100 mm and the rod diameter 50 mm. The mathematical model which describe the actuator behaviour consider the viscous friction coefficient of 75 Ns/m and thermal exchange coefficient of 50 J/m^2/K/s with an external temperature of 293.15 K. The PPV has the area of control orifices of 7 mm^2 and the flow control coefficient of 0.72. The valve natural frequency is 80 Hz and the valve rate current 10 mA.

Using LMS Imagine.Lab Pneumatics the pneumatic actuator positioning system in closed-loop is numerical simulated for different signal inputs.

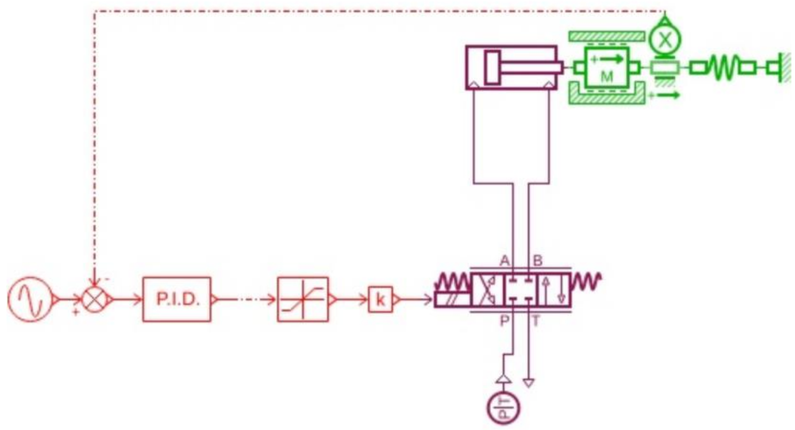
In Figure 1 is given the bloc diagram of the pneumatic actuator in closed loop and the numerical simulation scheme. In the bloc scheme is used a proportional controller and the load is giving by the inertial mass of 25 kg moved by the actuator and the variable technological force with a constant component of 2000N and a variable elastic force with the spring rate of 1000N/m (see block “LOAD” in Fig. 1.a). The displacement sensor has a zero offset and the gain 25 mA/m. The pressure of the compressed air supplied in the system is 7 barA.

The reference input signals are plotted in Figure 2, with amplitude of 10 mA for all signals and frequency of 0.05 Hz for sinus wave and triangular wave electrical signal.

Because the pneumatic control system proposed in this study is a single input single output system (SISO) with linear load and isothermal pneumatic actuator we used a proportional controller [13]. Keeping the same load value and changing the signal input we want to find the performances of the positioning pneumatic actuator and the value of proportional gain controller, too.

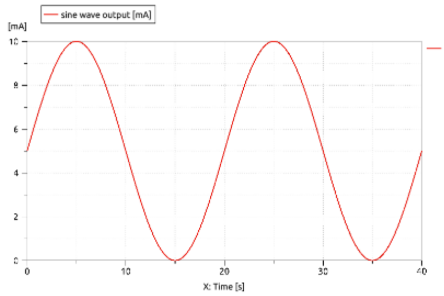


a)

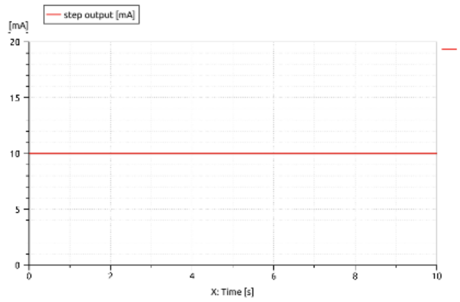


b)

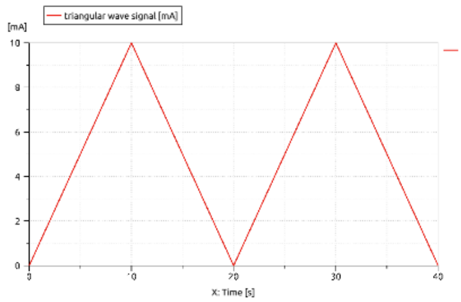
Figure 1. Bloc scheme (a) and numerical simulation scheme (b) of the pneumatic system.



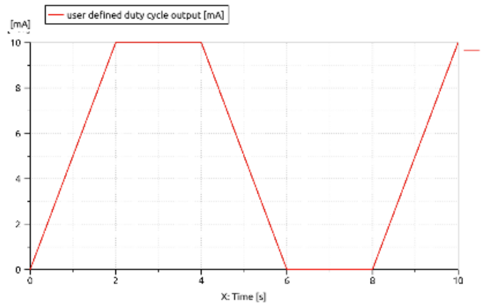
a)



b)



c)



d)

Figure 2. Electrical input signal type: a) sine wave, b) step, c) triangular wave, d) trapezoidal wave.

IOP Conf. Series: Earth and Environmental Science **664** (2021) 012030 doi:10.1088/1755-1315/664/1/012030

3. The steady-state and dynamic characteristics

3.1. Steady-state characteristics

The hysteresis or steady-state characteristics is plotted in Figure 3 for a triangular wave input with frequency of 0.05 Hz and the amplitude of 0.1mA, 1.0mA and 10mA. The hysteresis is obtained having in x-coordinate the amplitude of the input signal and in y-coordinate the rod displacement (or piston displacement).

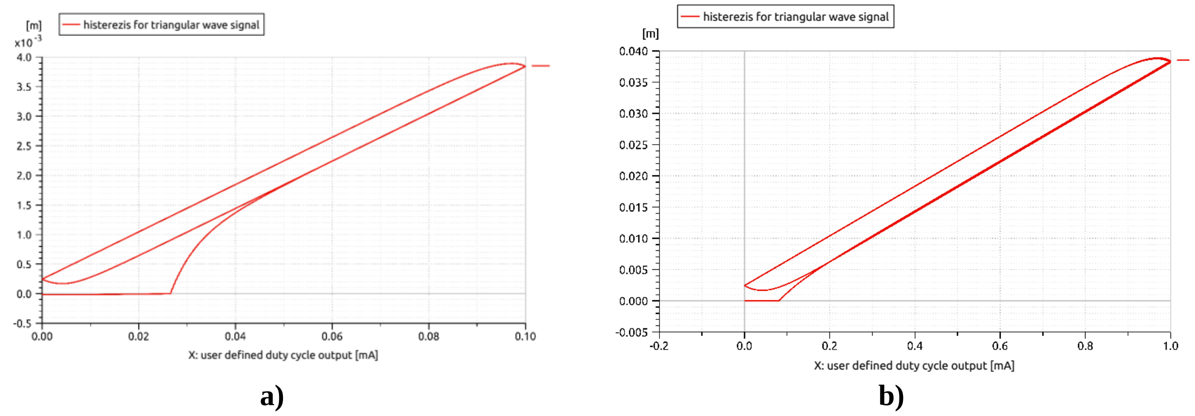
The absolute (error) hysteresis (ε_H) is expressed as a percentage of the difference between the upstroke (y_{up}) and downstroke (y_{dw}). The relative hysteresis (H) is expressed as:

$$H = \frac{y_{up} - y_{dw}}{y_{max} - y_{min}} \times 100\%. \tag{1}$$

In all three cases it is found the same value of *H*:

$$H = \frac{(2.6-2.2)\cdot 10^{-3}}{0.004} \times 100\% = \frac{0.026-0.022}{0.04} \times 100\% = \frac{0.25-0.21}{0.4} \times 100\% = 10\%, \tag{2}$$

with the values of y_{up} and y_{dw} (respectively, y_{max} and y_{min}) reading on Figure 3.



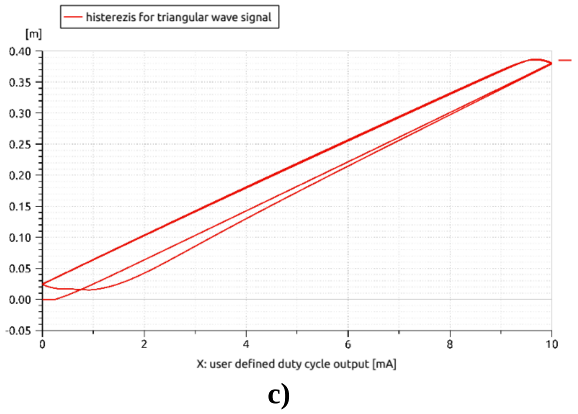
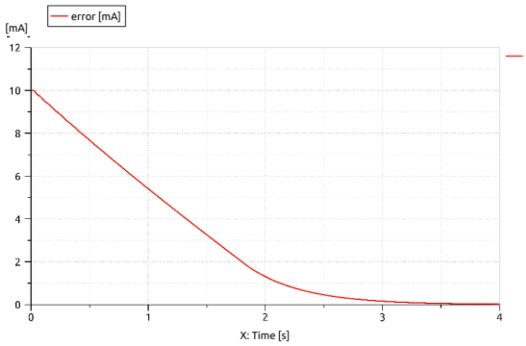
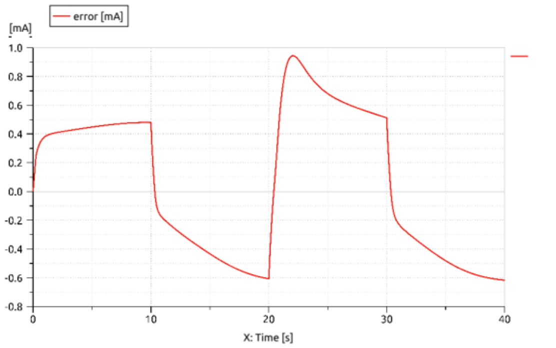


Figure 3. Static characteristic for triangular wave signal with amplitude 0.1 mA (a), 1.0 mA (b) and 10 mA (c) if the signal frequency is 0.05Hz.

The error of a positioning control system in closed loop is the difference between the input signal, the reference, and the feedback signal measured with the displacement sensor. In Figure 4, the error evolution is represented. It is observed that in the case of a step input of 10mA the error is zero after 4 seconds, unlike the case (b) with the triangular wave signal having 10mA amplitude. In this case the error follow-up the input signal and never will be zero.



a)



b)

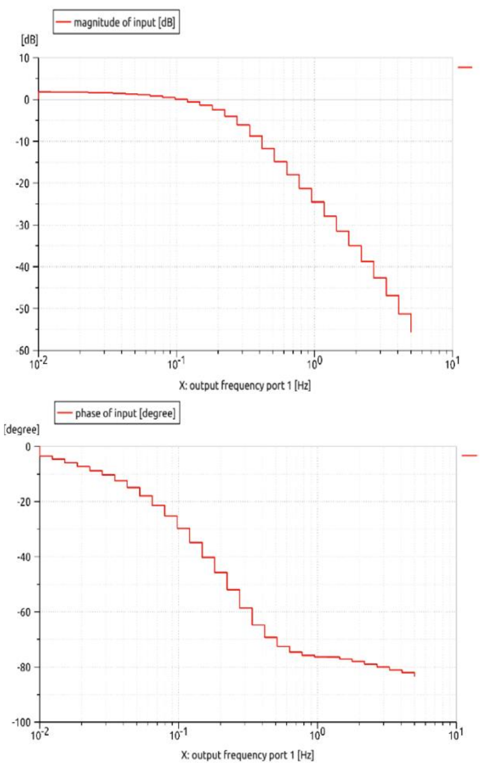
Figure 4. The error in rod displacement to different inputs: b) 10mA step input, c) 1mA triangular wave

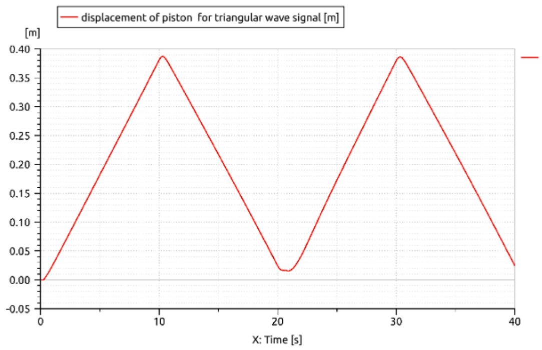
The relationship between the output of the controller $u(t)$ and the actuating error signal $e(t)$ is: $y(t) = K_p \cdot e(t)$, or Laplace transformed quantities: $K_p = \frac{U(s)}{E(s)}$ with K_p the proportional gain. The numerical simulation results in Figures 3 and 4 are for the same proportional gain value of 5.

3.2. Dynamic characteristics

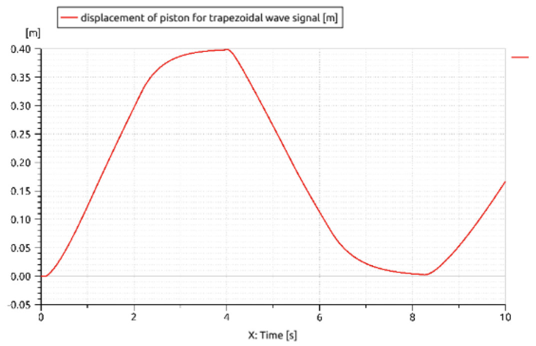
Frequency response analysis is the response of the system to a sinusoidal input. The sinus wave input was with 10mA amplitude and frequency of 0.05Hz and Bode diagrame is given in Figure 5.a.

The frequency range is from 0.01 to 1.5Hz and the magnitude at -3dB is founded the cut frequency of 0.2 Hz.





b)



a) c)

Figure 5. Dynamic characteristic: a) Bode diagram; b) rod displacement at triangular wave input; c) rod displacement at trapezoidal wave signal.

In Figure 5.b is plotted the rod piston displacement on the triangular input of 10 mA amplitude and is observed that the piston makes the stoke with an error of 0.01 mm and follow-up the input signal. In Figure 5.c at trapezoidal wave input (as in Figure 2.d), the piston makes the stroke with zero error. In both cases is observed a little time delay when a new cycle is beginning, see the time of 20 seconds for triangular input and time of 8.15 seconds for trapezoidal input.

4. Dynamic behaviour

It was studied the dynamic behaviour of the positioning pneumatic actuator in closed loop for the input signals presented in Figure 2 to see the evolution of stability.

The system response to all step inputs is the response of an overdamped control system which can be approximated with a first order control system. So, for input step of 10 mA amplitude applied as reference, the constant time (T) is 1.397 seconds, the rise time is $(T_r = 2.13\ \mathrm{s})$ and time settling is $(T_s = 4.08\ \mathrm{s})$, see Figure 6. Time constant is 63.2% of the response system to the step signal and it describes the speed with which the control system responds to the reference input. Time rise is 90% of the response system and time settling is 100% of the response system.

The positioning error is zero and the proportional gain controller is the same of 5.

For the parameters of the pneumatic actuator presented in chapter 2 of the paper, the control system response is stable for all input signals applied with amplitudes of 10mA. In Figure 7 are given the time evolution of piston displacement, velocity, pressures in the cylinder chambers and force variation at the rod piston given by the external variable load. In Figure 7.b it can be observed that at the beginning of movement the velocity has an oscillatory behavior with small amplitudes which indicates the presence of friction force in the pneumatic servomechanism numerical model. Madziak et al. [14] underline that the non-linear frictional resistance is the cause of irregular linear motion at low speeds.

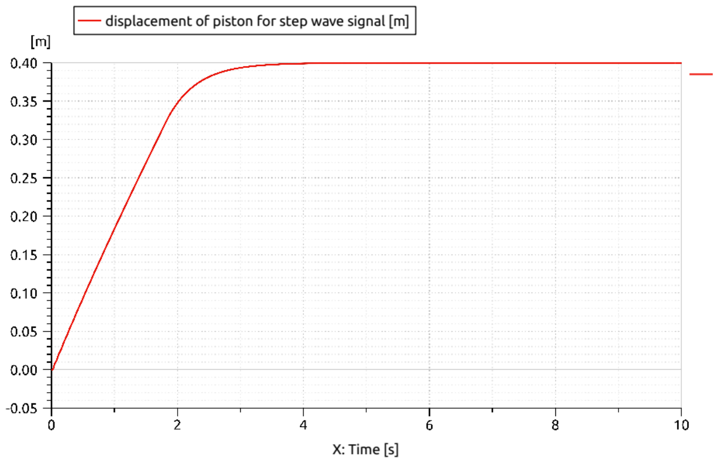


Figure 6. Dynamic behaviour to step signa with 10 mA amplitude.

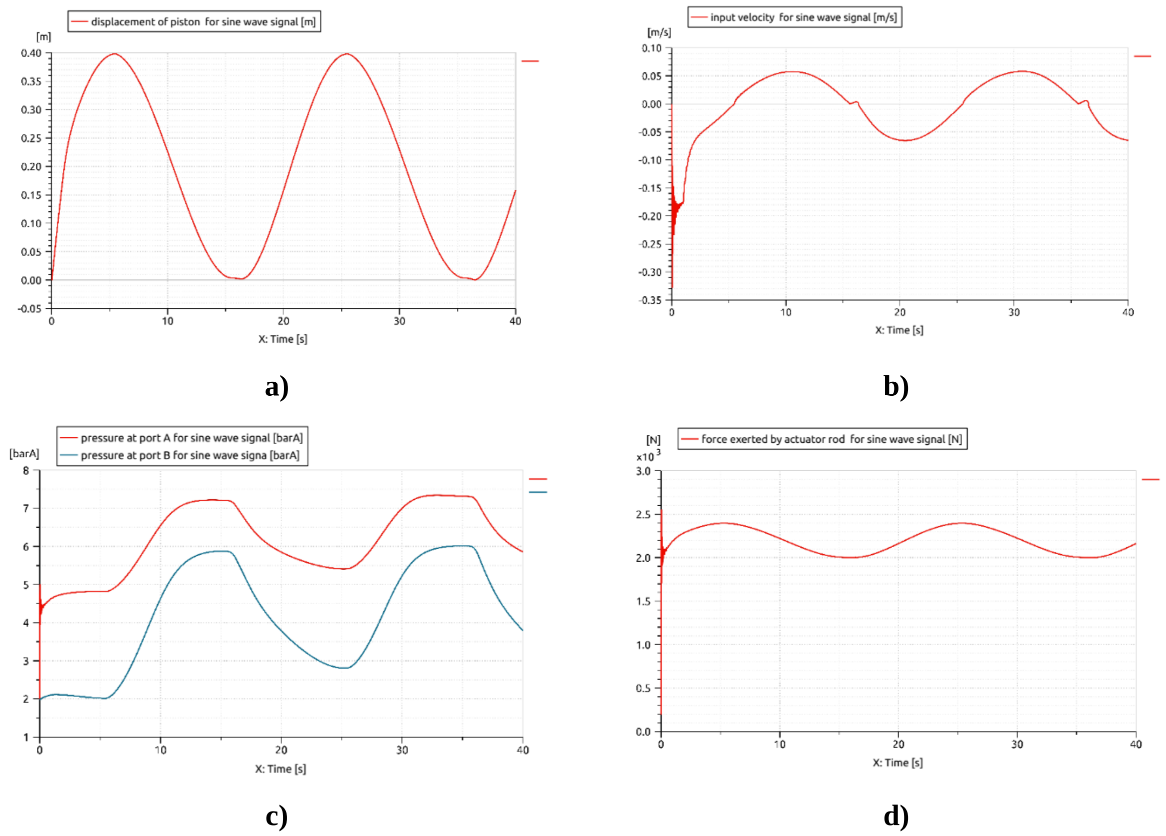


Figure 7. Dynamic behaviour to sine wave signal with 10mA amplitude and 50Hz frequency: a) rod displacement, b) rod velocity, c) pressures in cylinder chambers, d) force at the rod.

5. The influence of K_p coefficient on system dynamics

This paper has two goals: the performance of the positioning pneumatic actuator in closed-loop having a proportional pneumatic valve as pneumatic amplifier, and to find the limits of proportional gain of the controller, K_p, so that the pneumatic control system preserves its stability.

The method used to find the K_p limits was Ziegler-Nichols tuning method simplified for a proportional controller. For a given step signal, the proportional gain K_p is increased until the stability of the system is compromise or the system reaches periodic oscillations.

In Figure 8 is presented the response to 0.1 mA step input if the $K_p = 5$ (a) and the error (b). It is observed that the response has a delay time of 0.31 seconds, the time rise is 1.38 seconds and the settling time is 3.19 seconds. Regarding Figure 8.c and 8.d it is observed that when K_p is increasing starting from $K_p = 2$ the time delay is diminished from 0.86 seconds to 0.26 seconds for $K_p = 6$ and the system became a little bit faster in time response, $T_s = 2.91$ seconds for $K_p = 6$.

If $K_p = 12$ periodic oscillations appear and the quality of the positioning action of the pneumatic actuator is compromised, even if the system is faster. The stability is preserved but the periodic oscillation for the first seconds of the movement cannot be accepted.

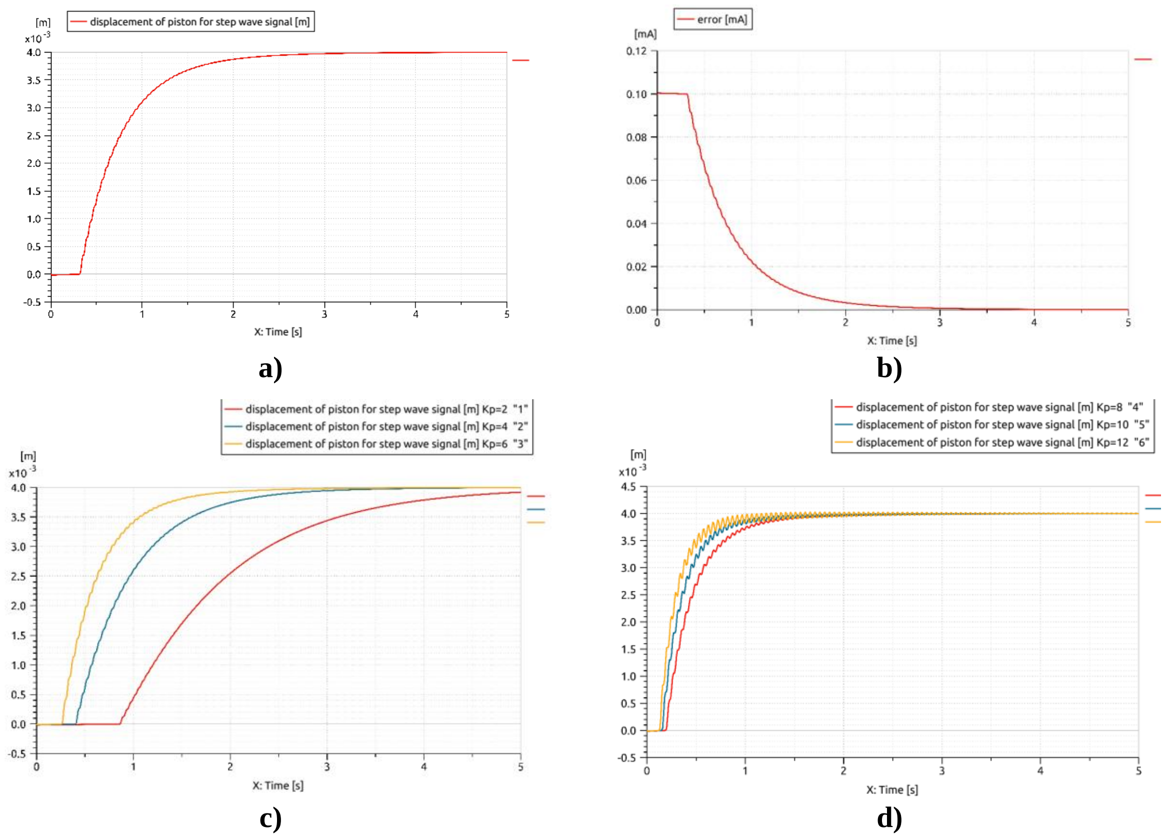


Figure 8. Influence of K_p coefficient of the P controller for step signal with amplitude of 0.1 mA: a) rod displacement, b) the error, c) and d) the dynamics for different K_p values.

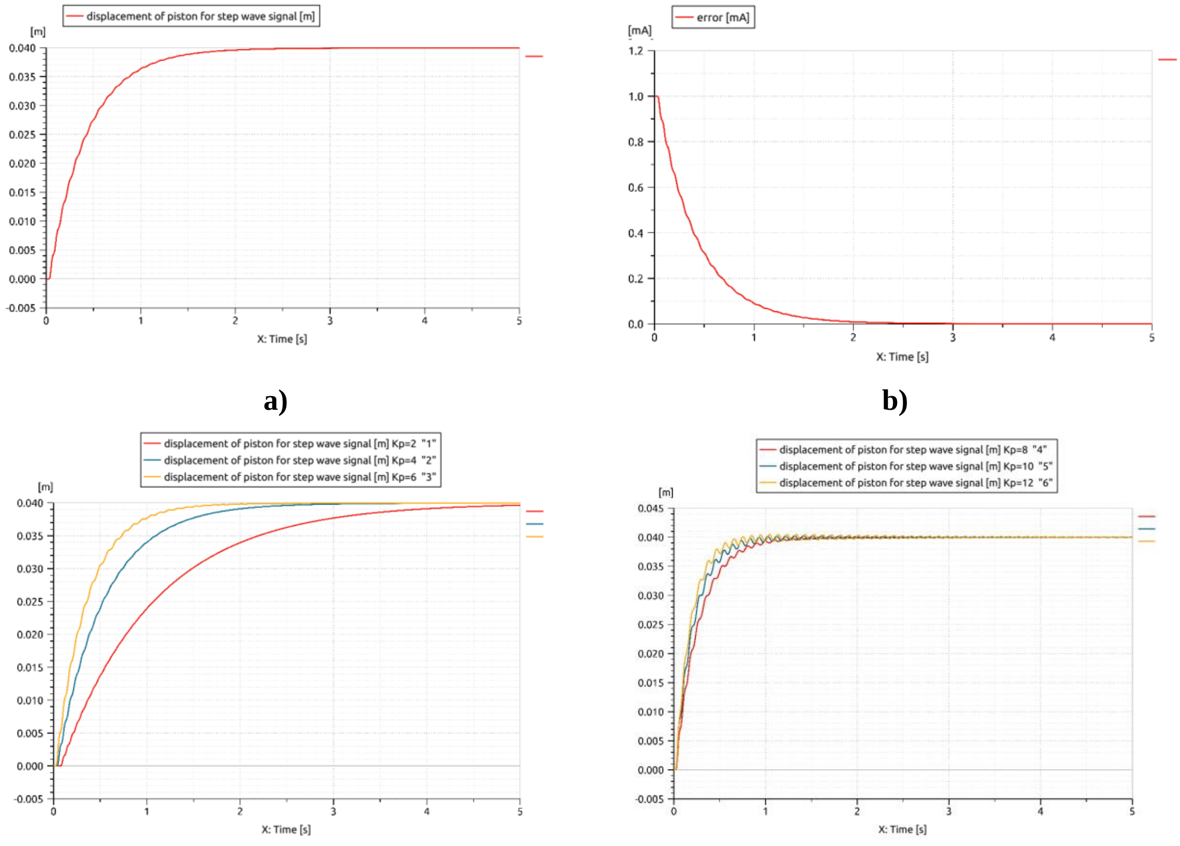
In Figure 9 is presented the system behaviour for 1 mA step input.

In Table 1 are presented the values of time delay (T_d), rise time and settling time for different K_p values when the step input is 0.1 mA and 10 mA.

It is observed that for 1 mA step input the system has a time delay with low values even for K_p with $K_p = 5$, Fig. 9.a. Also, the quality of the pneumatic control system parameters is comprised for $K_p > 6$ although the frequency of the oscillations is lower.

Table 1. Tuning K_p controller and positioning pneumatic actuator performances

K_p (-)	Input step 0.1 mA				Input step 1 mA				Input step 10 mA			
	T (s)	T_d (s)	T_r (s)	T_s (s)	T (s)	T_d (s)	T_r (s)	T_s (s)	T (s)	T_d (s)	T_r (s)	T_s (s)
2	1,979	0,86	3,342	6,81	1,083	0,08	2,431	6,5	1,397	0,02	3	6,8
4	0,974	0,4	1,734	3,3	0,534	0,04	1,215	3,5	1,397	0,02	2,21	4,11
5	0,769	0,31	1,388	3,19	0,436	0,03	0,982	3,02	1,397	0,02	2,13	4,08
6	0,631	0,26	1,156	2,91	0,366	0,03	0,816	2,49	1,397	0,02	2,08	3,38
8	0,471	0,19	0,859	3,5	0,258	0,02	0,594	2,03	1,397	0,02	2,04	3,02
10	0,375	0,15	0,722	2,51	0,226	0,02	0,508	2,51	1,397	0,02	2,04	2,74
12	0,316	0,12	0,535	3,5	0,18	0,02	0,428	3,51	1,397	0,02	2,04	2,58



a) **b)** **c)** **d)**

Figure 9. Influence of K_p coefficient of the P controller for step signal with amplitude of 1 mA: a) rod displacement, b) the error, c) and d) the dynamics for different K_p values.

6. Conclusion and future work

The paper aimed to study the dynamic behavior of a positioning pneumatic control system in closed-loop and to find the range of the controller proportional gain so that the pneumatic control stays stabile and accurate.

It was considered that the pneumatic actuator uses a proportional pneumatic valve as pneumatic amplifier and a proportional controller. The external load is variable. The steady-state characteristics and the dynamic behavior were determined for different input signals. Using numerical simulations of the pneumatic control system it is found that positioning accuracy depends on the input signal type, the input parameters values and the controller, not only by the actuator and pneumatic amplifier.

It was found a range of K_p variation with maximum value of 8 so that the pneumatic control system remains stable and accurate. Numerical simulations show that the system can be adapted at the external perturbations by modifying the reference signal and the proportional gain K_p.

This paper, together with [12] which is referred to the same positioning pneumatic system in open-loop, has underlined (at least theoretically) that the pneumatic linear positioning systems with proportional valve (PPV) are easy to be controlled and represent an alternative to the pneumatic positioning systems with servo valve.

References

[1] Davis S 2018 Pneumatic Actuators *Actuators* **7** 62 doi:10.3390/act7030062
[2] Foit K, Banaś W and Ćwikła G 2018 The pneumatic and electropneumatic systems in the context of 4th industrial revolution *IOP Conf Ser Mater Sci Eng* **400** 022024
[3] Avram M and Constantin V 2011 Controlling of a Pneutronic Positioning System without Proportional Equipment *Romanian Review Precision Mechanics, Optics & Mechatronics* **39** pp 82-5
[4] Vladu I C, Roșca D, Bîzdoacă N and Stoian V 2016 Dynamic control of pneumatic actuator system *Proc. 18th Int. Conf. on Scientific Research and Education in the Air Force -AFASES* pp 399-404 doi: 10.19062/2247-3173.2016.18.1.54
[5] Stoychitch M Y and Knezevic B Z 2019 *IOP Conf. Ser Mater Sci Eng* **477** 012034
[6] Kobayashi W, Kato N, Dohta S and Akagi T 2017 Position control of flexible pneumatic cylinder using tiny embedded controller with disturbance observer, *Int J Mech Eng Rob Res* **6** (4) pp 318-21
[7] Ahmad 'Athif Mohd Faudzi A A M, bin Osman K, Rahmat M F, Mustafa N D, Azman M A and Suzumorid K 2012 Controller Design for Simulation Control of Intelligent Pneumatic Actuators (IPA) *System Procedia Engineering* **41** 593 – 99
[8] Valdiero A C, Ritter C S, Rios C F and Rafikov M 2011 Non Linear Mathematical Modeling in Pneumatic Servo Position Applications *Hindawi Publishing Corporation Mathematical Problems in Engineering* Article ID 472903 doi:10.1155/2011/47290
[9] Ali H I, Noor S B B M, Bashi S M and Marhaban M H A 2009 Review of Pneumatic Actuators (Modeling and Control) *Australian Journal of Basic and Applied Sciences* **3**(2) 440-54
[10] Dragne F D, Marc Alirand M, Ioan Mircea Oprean I M and Vasiliu N 2009 ABS valve model reduction AMESIM, *Proc. of the Romanian Academy* A **10**/2 pp189-96
[11] Vasiliu N, Vasiliu D, Călinoiu C, Puhalschi R 2018 *Simulation of fluid power systems with Simcenter Amesim*, (CRC Press Taylor and Francis Group) pp 11-40
[12] Cococi V N, Safta C A and Calinoiu C 2020 Dynamic behaviour of pneumatic actuators in open-loop controlled by proportional valves *Proc. 9th Int. Conf. on Thermal Equipment, Renewable Energy and Rural Development (TE-RE-RD 2020)* Vol 180 04012 https://doi.org/10.1051/e3sconf/202018004012
[13] Faudzi A A M, bin Osman K, Rahmat M F, Din Mustafa N, Asyraf Azman M, Suzumori K 2012 Controller Design for Simulation Control of Intelligent Pneumatic Actuators (IPA) System *Procedia Engineering* **41** 593 – 99
[14] Magdziak Ł, Malujda I, Wilczyński D, Wojtkowiak D 2017 Concept of improving positioning of pneumatic drive as drive of manipulator *Procedia Engineering* **177** 331 – 38