Scientific Bulletin of the „Petru Maior" University of Tîrgu Mureş
Vol. 12 (XXIX) no. 1, 2015
ISSN-L 1841-9267 (Print), ISSN 2285-438X (Online), ISSN 2286-3184 (CD-ROM)

EXPERIMENTS OF PNEUMATIC LINEAR POSITIONING WITH CONVENTIONAL CONTROL SYSTEM

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Abstract

In a great number of industrial applications the pneumatic positioning systems are used often, because of good ratio between load and supplied power, in spite of fact that these systems present nonlinear behaviour.

In this paper, a pneumatically actuated linear positioning system is controlled using a classical PID. Simulations were performed using a software program available from the suppliers of the pneumatic unit to evaluate responses, and then the tests were repeated on the laboratory bench available. Software simulation results were compared with those obtained on real hardware.

Key words: pneumatic system, linear positioning, PID control

1. Introduction

The pneumatic systems have a lot of advantages like: simple design, economical to operate, high effectiveness, durability and reliability, adaptability to hostile environment, safe to use around flammable materials, environmental friendly and low cost of implementation.

A brief overview of pneumatic actuators, based on modelling and control strategies is very well developed in [13]. The theoretical and experimental aspects of a position control system using a pneumatic actuator and the modelling of an electro-pneumatic system, including valves, are studied in [1,4,7,15]. Also, a series of researches proposed a precise position control strategy for servo-pneumatic actuator systems, applied in combination with the PID controllers. Classical PID controllers and their tuning criteria are presented in [2,3,10]. Controllers, including the classical PID, tuned according to damping optimum criteria and the nonlinear PID, for a pneumatic positioning systems are proposed and tested in [12,14,15,16]. Other researches use the conventional controllers based on internal model or combined with fuzzy techniques [5,11].

In another research, the PID controllers are combined with a nonlinear gain, adjusted based on the generated errors feedback to the controller. The positioning accuracy required by the production task was achieved using such a control strategy.

This paper describes a study of linear positioning system, using classical PID controllers.

2. The automation of pneumatic process

Pneumatic actuators are used for industrial applications due to their high power rate. In order to obtain linear (or angular) movement, the pneumatic systems uses compressed and treated air or compressed inert gases.

The nonlinearities introduced due to high friction forces make accurate position control of a pneumatic actuator difficult to achieve.

For a nonlinear application system, which requires high accuracy in positioning and actuator rigidity, under different external loads, the classical PID controller law is not competitive enough.

However, in industrial applications that do not requires positioning with high accuracy, the conventional control algorithms, with the command signal, *u(t)*, can be used [2,3,10,17].

$$u(t) = k_p\left(1 + \frac{1}{T_i}\int_0^t \varepsilon(t)dt + T_d\,\frac{d\varepsilon}{dt}\right) \tag{1}$$

The linear pneumatic circuit, with the air source, service unit and proportional valve is presented in fig. 1. The generalized block diagram of the control system is presented in fig. 2.

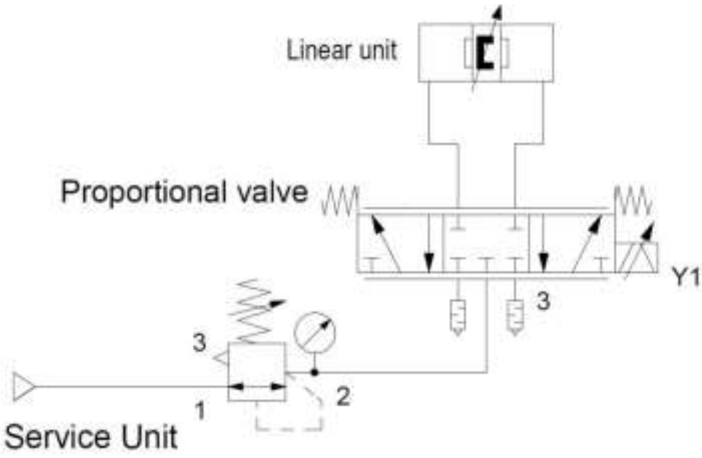

Fig. 1: The linear pneumatic circuit

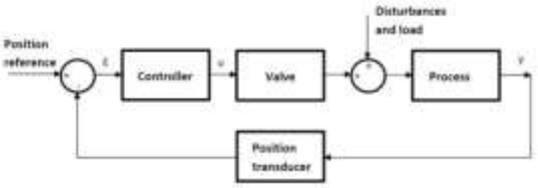

Fig. 2: The classical diagram of the position control loop

Consequently, in the last years, the performance of pneumatic positioning system is continuously improved. The main disturbance is the friction. The friction of moved mass is also considered. Therefore the friction should always be taken into consideration.

Therefore, the dynamics, *x(t)*, of the positioning system (only mechanical part) is modelled based on Newton's second law [8,9,12,16]:

$$M\frac{d^2x}{dt^2}+\gamma\frac{dx}{dt}+Fx(t)=A\Delta p(t) \qquad (2)$$

where: M is the mass of the piston-load assembly; γ - the viscous friction coefficient; F - the load resistant force; A - the cross-section area; Δp – the differential pressure in the chambers of the pneumatic positioning system.

From equation (2) results the transfer function of the mechanical part, neglecting the term s^2, for very small values of M.

$$H(s)=\frac{\frac{A}{F}}{\frac{M}{F}s^2+\frac{\gamma}{F}s+1}\cong\frac{k}{Ts+1} \qquad (3)$$

Due to the compressibility of air, which requires processing before use, these systems work with dead-time, uneven moving speed and relatively low accuracy [6,16].

PID controller can be adjusted to tune the coefficients of k_p - proportional gain, T_i - integrative time and T_d - derivative time with the purpose to compensate the feedback error [3].

According to the Ziegler-Nichols criteria, ultimate gain can be obtained by increasing the gain constant in proportional mode until a sustained oscillation takes place. Hawing the ultimate gain and oscillation period, the Ziegler-Nichols method can be applied to obtain the tuning parameters of the PID controller.

3. Experimental tests

Before running tests on the physical system, some simulations were made in dedicated software, using a pneumatic model. To obtain more realistic results during simulation, physical parameters of the real process must be introduced in the simulation software. Some of them are the air pressure after the service unit, nominal air flow trough the proportional valve, moved mass by the linear unit. The main purpose of this simulation was to find starting gains for the PID controller.

The controller is an analogue PID, with access to each k_p, T_i and T_d through potentiometers. All electrical components from the simulator and physical system, are using and delivering analogue control voltage between (0-10) V.

Ziegler Nichol tuning method is implemented by inserting the calculated parameters into PID block.

The simulated system was brought to sustained oscillations by increasing the ultimate gain K_c up to 11. The period of the oscillations was measured and found T_u = 0.21 seconds.

The response with sustained oscillations of simulated circuit is shown in fig. 3.

The real system response with sustained oscillations is shown in fig. 4. K_c in this case was 10.8 and T_u = 1.54 seconds.

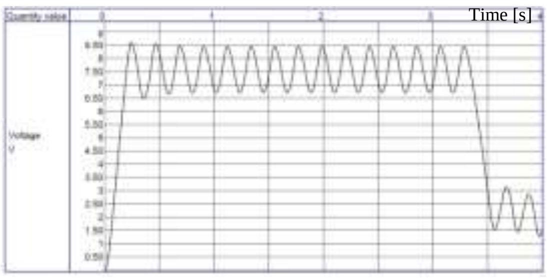

Fig. 3: The simulated system in sustained oscillations

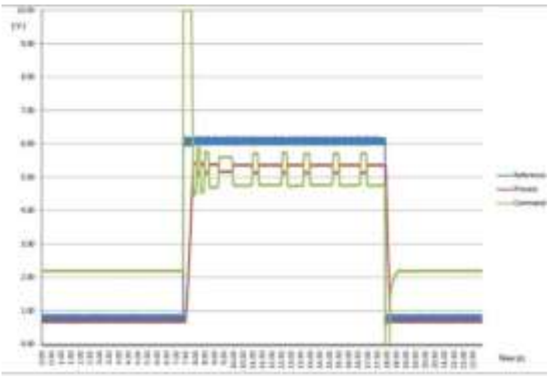

Fig. 4: The physical system in sustained oscillations

Data capture was performed using a data acquisition boards on a 12-bit resolution with speeds up to 10 ks/s.

After calculating the approximate parameters with the Ziegler-Nichols method, for the physical system, the following PI gains were achieved: $k_p = 4.86$; $T_i = 3.78$ seconds. The same tests were performed on the simulated model and on the real equipment, in order to compare the results. The behaviour was similar with a significant difference of the oscillation period because friction of real system could not be measured exactly. The performance of the system is significantly enhanced with the respect of system robustness against the load changes.

The test equipment is presented in the fig. 5, where the following components can be identified: the process, (1) is a linear actuator; the component (2) represents the position feedback transducer which is a liner potentiometer; the proportional pneumatic valve (3); the component (4) is the reference voltage generator. The component (5) is an analogue PID controller with direct access to P, I and D gain adjustment and (6) is the power supply unit.

In fig. 6 the following components are identified: the position transducer (between 0-1 connection points), the reference generator (0-2), the PID controller (0-3), the valve (0-4), which is a proportional valve that converts the electrical signal in pneumatic signal.

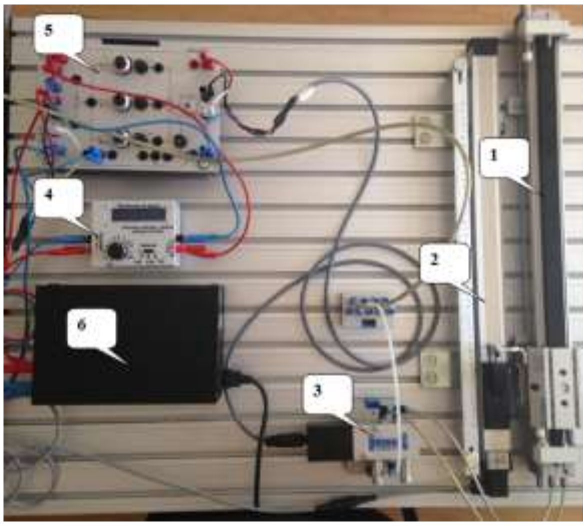

Fig. 5: Test equipment of positioning system

4. Conclusions

The behaviour of the simulated system was compared to the behaviour of the physical process. The simulated response is shown in fig. 7 and the response of the real equipment is shown in fig. 8.

It was observed that, considering the step reference, the proposed algorithm leads to a response with an admissible error and with oscillations around imposed position. Also, the closed loop response of the positioning system, points out a delay time.

In these conditions, the efficiency of the PID proposed controller degrades rapidly.

It is obvious that the problems come from the nonlinear behaviour, associated to compressibility and dry-friction forces. These features limit the utility of the PID controller in practical applications.

However:

- high precision positioning can be achieved for a part of the cylinder stroke;
- a precise mathematical model of the process provides a convenient method to control a nonlinear system by using linear controllers, tuned based on well-known rules.

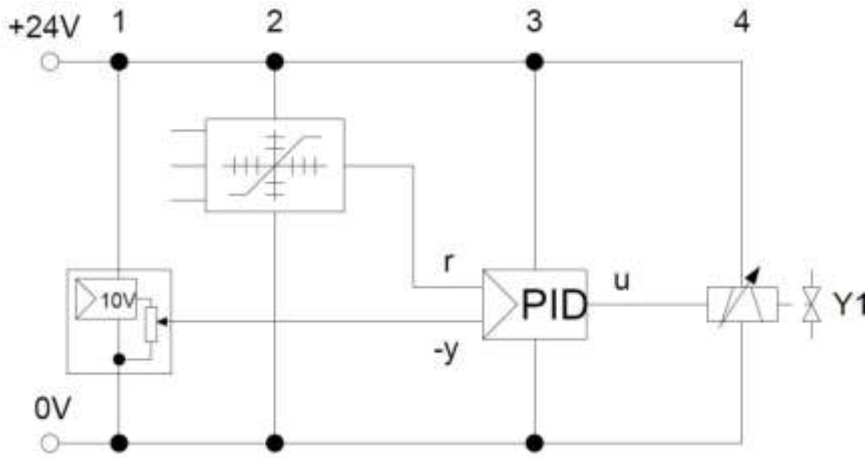

Fig. 6: Electrical connections of pneumatic components

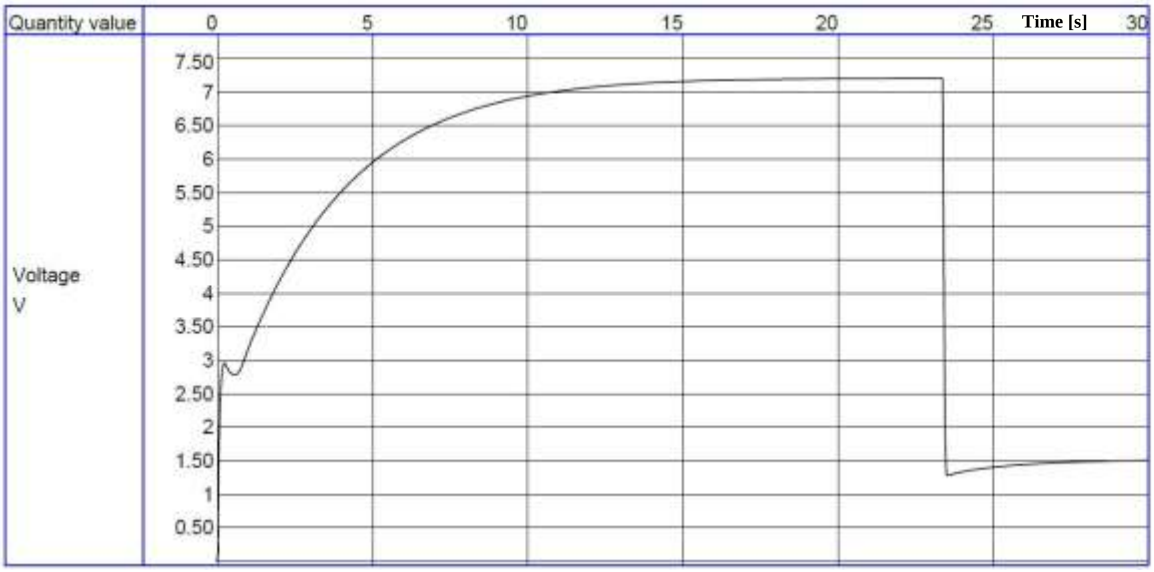

Fig. 7: Simulated response after PID tuning

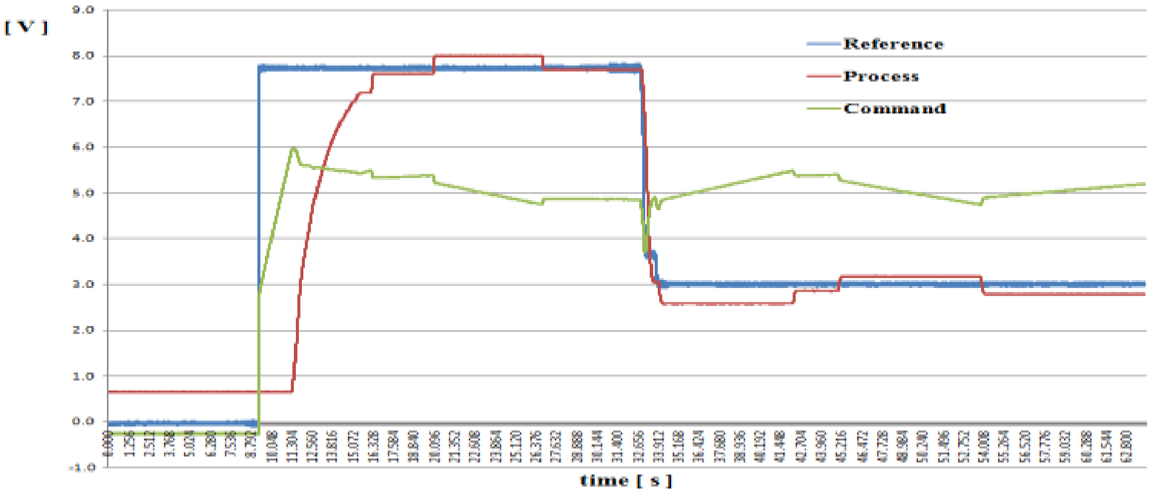

Fig. 8: Data capture of the real process

The purpose of this paper is to present a simplified way to control the pneumatic positioning system, with the conventional PID algorithm.

The tuning parameters influence the dynamic and need to be determined with accuracy.

This research is useful for choosing linear positioning systems depending on the application, and obtaining the tuning parameters using simulation software packets, ready to be transferred in hardware equipment.

Future research will focus on:

- determine a PID controller, combined with a nonlinear amplifier;
- a stability analysis of the closed loop system;
- the robustness approach.

Acknowledgement

This paper is based upon work supported by the Energy and Electrotechnologies Management research center.

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