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Hardware and Software Structure of a Pneumo-Hydraulic Positioning System

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Abstract: The paper presents the hardware and software structure of a linear unit for pneumo-hydraulic positioning, developed by the authors. The unit features two identical cylinders, one pneumatic and one hydraulic, mounted in parallel. The speed control is achieved by the use of two check valves of original construction. Mathematical model, simulation results and experimental results are also provided.

Keywords: mechatronics, pneumatics, hydraulics, positioning system, mathematical model.

1. INTRODUCTION

Pneumatic actuation and control of the driven load speed through a hydraulic control circuit are specific features of pneumo-hydraulic units. Even if their construction is more complex, it eliminates the shortcomings of the speed control of pneumatic units, caused by the high compressibility and low viscosity of the working fluid. The rigorous control of the load speed opens the way to the development of high accuracy positioning units.

2. THE EXPERIMENTAL MODEL

The principle scheme of the unit that is subject of this paper is presented in figure 1. The following equipments can be identified:

- *MLP-H* –linear pneumo-hydraulic motor;
- SPC_1 and SPC_2 – controllable check valves that can be unlocked if a proportional signal is applied;
- *DPC* – classical pneumatic direction control valve;
- *Tp* – position sensor.

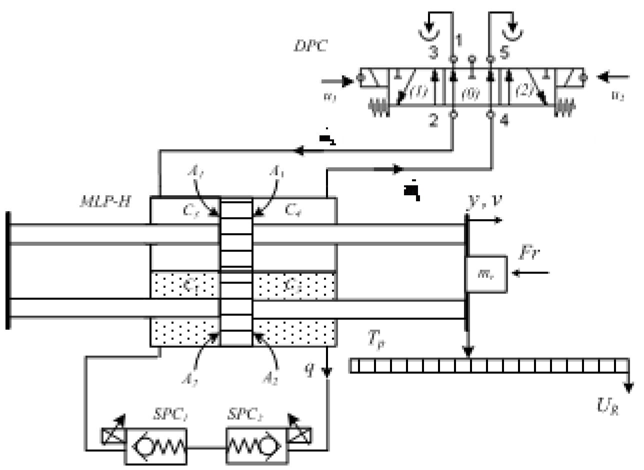

Fig. 1. The principle scheme of the unit.

The following features are specific to the proposed structure:

- the pneumo-hydraulic linear motor *MLP-H*, constituted by two identical cylinders with bilateral piston rods, mounted in parallel; their rods are joint by stiff clamps; the incremental position sensor is integrated in the construction;
- two identical equipments of original construction – SPC_1 and SPC_2 – are used in order to control the speed of the mobile unit; these equipments are in fact controlled check valves that can be unlocked if a proportional signal is applied; such an equipment allows the free flow of the fluid in one direction, through a section equal to the nominal section of the equipment; the flow is possible in the opposite direction only in the presence of an electric control signal; it is thus possible to control the fluid flow rate in the hydraulic circuit and therefore the control of the load speed; the valve SPC_1 must be unlocked in order to move the load from left to right (fig.1); the valve SPC_2 must be unlocked for the displacement of the load in the opposite direction.

Figure 2 presents the 3D model of the system, built using SolidWorks graphical environment. It can be seen also the air preparation unit *GPA*, needed for the good functioning of the system. The components of the system are mounted on the base plate *PB*.

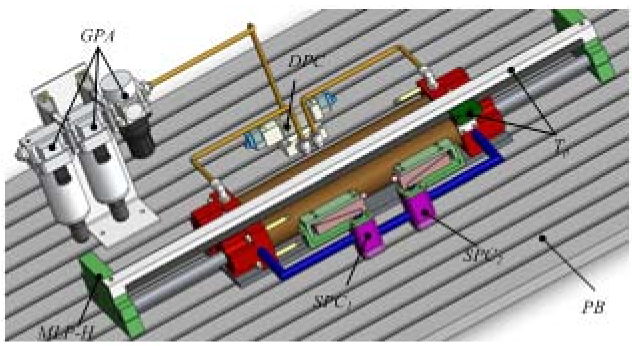

Fig. 2. Three-dimensional model of the unit.

The system is controlled by a personal computer *PC*, interfaced through specialized I/O modules.

The principle scheme of check valve of original construction used for the control of the speed is presented in figure 3, Avram et al. (2010).

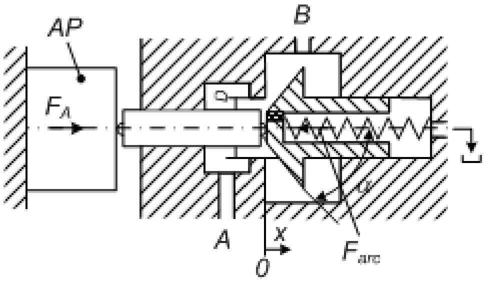

Fig. 3. The check valve of original construction.

If the piezoelectric actuator *AP* is not supplied, the equipment behaves as a classical check valve, namely the flow from *A* to *B* is free and the flow from *B* to *A* is blocked. If the actuator is supplied with a voltage *u*, it is possible for the fluid to flow from *B* to *A* through a flow section proportional to the voltage *u*.

3. MATHEMATICAL MODEL

In order to study the dynamic behavior of the pneumo-hydraulic unit, the authors developed its mathematical model starting from the scheme presented in figures 1 and 3. Table 1 presents the used notations.

Table 1. Notations used in the model of pneumo-hydraulic unit

Used notation	*Significance*
A_i	Active sections of the motor, i=1…4
B	Damping factor
c	Maximum travel of the rods
d_n	Nominal diameter of the supplying pipes
d	Piezoelectric coefficient
D	Diameter of the check valve seat
E	Oil elasticity module
f_0	Initial compression of the spring
F_{STACK}	Force generated by the piezoelectric stack
F_A	Force developed by the actuator A
F_c	Flowing force through the section controlled by the valve
F_r	External force
k	Equivalent stiffness of the piezoelectric stack
k_{arc}	Spring factor
k_m	Mechanical amplification factor
$\dot{m}_3$, $\dot{m}_4$	Mass flow rates in the chambers of the pneumatic cylinder
M	Mass of the mobile assembly piston – rod - load
P_a	Supply pressure
P_i	Pressures in the chambers of the motor; i=1…4
P_0	Atmospheric pressure
q	Oil flow rate through the check valve
R	Universal gas constant
S_c	Flow section through the check valve
T_a	Absolute temperature
V_i	Volumes of the motor chambers, , i=1…4
u	Control voltage
x	Displacement of the check valve seat
x_n	Nominal opening of the check valve
x_{STACK}	Displacement of the piezoelectric stack
y	Displacement of the two cylinder rods mounted in parallel
α	Angle of the valve cone
α_2	Shape factor
χ	Adiabatic factor
ρ	Oil density

In order to move the load in the direction depicted in figure 1, the classical pneumatic direction control valve *DPC* must perform the distribution scheme (1) and the check valve SPC_1 must be unlocked. The latest will generate a flow section proportional to the applied control voltage, section that allows the control of the load speed.

The model consists of the following equations:

- the equation that describes the movement of the load:

$$M \cdot \frac{d^2y}{dt^2} + B \cdot \frac{dy}{dt} + F_r = (P_3 - P_4) \cdot A_1 - (P_2 - P_1) \cdot A_2 \quad (1)$$

- the variations of the pressures in the chambers of the hydraulic motor:

$$\frac{dP_1}{dt} = \frac{E}{V_1(y)} \cdot \left(q - A_2 \cdot \frac{dy}{dt} \right) \quad (2)$$

$$\frac{dP_2}{dt} = \frac{E}{V_2(y)} \cdot \left(-q + A_2 \cdot \frac{dy}{dt} \right) \quad (3)$$

- the variations of the pressures in the chambers of the pneumatic motor:

$$\frac{dP_3}{dt} = \frac{\chi}{V_3(y)} \cdot \left(\dot{m}_3 \cdot R \cdot T_a - A_1 \cdot P_3 \cdot \frac{dy}{dt} \right) \quad (4)$$

$$\frac{dP_4}{dt} = \frac{\chi}{V_4(y)} \cdot \left(-\dot{m}_4 \cdot R \cdot T_a - A_1 \cdot P_4 \cdot \frac{dy}{dt} \right) \quad (5)$$

The variable volumes that appear in equations (2)…(5) have the following expressions:

$$V_1(y) = V_{10} + A_2 \cdot y \quad (6)$$

$$V_2(y) = c \cdot A_2 + V_{20} - A_2 \cdot y \quad (7)$$

$$V_3(y) = V_{30} + A_1 \cdot y \quad (8)$$

$$V_4(y) = c \cdot A_1 + V_{40} - A_1 \cdot y \quad (9)$$

The oil flow rate between the two chambers of the hydraulic cylinder can be computed as:

$$q(x)=\begin{cases} S_c(x)\cdot\sqrt{\dfrac{2}{\rho}\cdot(P_2-P_1)} & if \;\; P_2>P_1 \\ 0 & if \;\; P_2=P_1 \\ -S_c(x)\cdot\sqrt{\dfrac{2}{\rho}\cdot(P_1-P_2)} & if \;\; P_2<P_1 \end{cases} \qquad (10)$$

The flow section through the valve is equal to:

$$S_c(x)=k_1\cdot x-k_2\cdot x^2 \qquad (11)$$

where k_1 and k_2 are constants:

$k_1=\pi\cdot D\cdot\sin\alpha$,

$k_2=\pi/2\cdot\sin\alpha\cdot\sin 2\alpha$.

In the relations presented before, D is the diameter of the seat and α is the angle of the valve cone.

- the equation that describes the movement of the valve seat:

$$m\cdot\frac{d^2x}{dt^2}+B\cdot\frac{dx}{dt}= \\ =F_A+(P_1-P_0)\cdot\frac{\pi}{4}\cdot D^2-k_{arc}(f_0+x)-F_c \qquad (12)$$

where:

F_c – flow force through the section controlled by the check valve, that can be computed as:

$$F_c=2\cdot\cos\alpha\cdot A_c(x)\cdot(P_2-P_1)/\rho \qquad (13)$$

F_A – force developed by the actuator A if it is supplied with the voltage u; the construction of the actuator A must be considered in order to establish the expression of the force; the actuator, of type P-287, is endowed with a piezoelectric stack integrated in a mechanical structure that achieves a high resolution, frictionless amplification of the displacement; the elastic structure is manufactured by wire electro-erosion; for this model, the amplification factor is $k_m=12[-]$; the stack parameters are similar to the model P-007.40, for which:

$$\begin{cases} k=19[N/\mu m] \\ d=500\cdot 10^{-12}[m/V] \end{cases},$$

where k denotes the equivalent stiffness of the stack and d the piezoelectric coefficient.

In the case of a piezoelectric stack, the relation that gives the force is, Belmut (2007):

$$F_{STACK}=k\cdot(x_{STACK}-d\cdot u)$$

Thus the force developed by the actuator A for a supply voltage u will be equal to:

$$F_A=\frac{k}{k_m}\cdot\left(\frac{1}{k_m}x-d\cdot u\right) \qquad (14)$$

The model can be simplified if it is supposed that there is a delay between the actuation of the direction control valve *DPC* and of the check valve *SPC* (fig.1). Therefore the initial conditions of the model become, as shown in figure 4:

$$\begin{cases} P_3=P_a \\ P_4=P_1=P_0 \\ P_2=P_1+[(P_3-P_4)\cdot S_1-F]/S_2 \end{cases}$$

Consequently, the flow sections through the pneumatic direction control valve *DPC* can be computed as:

$$A_1=A_2=A_n=\alpha_2\cdot\pi\cdot D_n^2/4 \qquad (15)$$

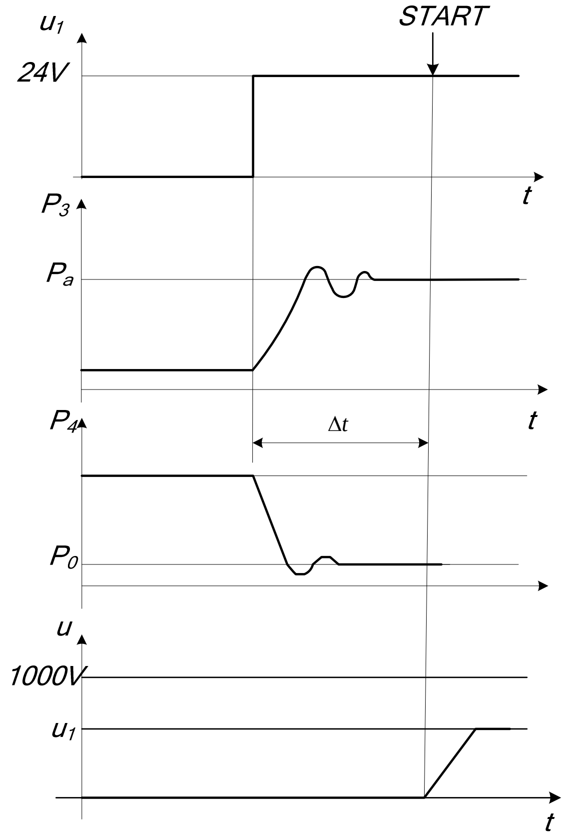

Fig. 4. Initial conditions.

The flow rates controlled by the *DPC* are equal to:

$$\dot{m}_3=\begin{cases} \dfrac{K\cdot P_a}{\sqrt{T_a}}\cdot A_n\cdot N\left(\dfrac{P_3}{P_a}\right) & if \quad 0<\dfrac{P_3}{P_a}<1 \\ 0 & if \quad \dfrac{P_3}{P_a}=1 \\ -\dfrac{K\cdot P_3}{\sqrt{T_a}}\cdot A_n\cdot N\left(\dfrac{P_a}{P_3}\right) & if \quad 1<\dfrac{P_3}{P_a} \end{cases} \qquad (16)$$

$$\dot{m}_4 = \begin{cases} \dfrac{K \cdot P_4}{\sqrt{T_a}} \cdot A_n \cdot N\left(\dfrac{P_0}{P_4}\right) & if \;\; 0 < \dfrac{P_0}{P_4} < 1 \\ 0 & if \;\; \dfrac{P_0}{P_4} = 1 \\ -\dfrac{K \cdot P_0}{\sqrt{T_a}} \cdot A_n \cdot N\left(\dfrac{P_4}{P_0}\right) & if \;\; 1 < \dfrac{P_0}{P_4} \end{cases} \tag{17}$$

where:

$$N(x) = \begin{cases} 1 & \text{if} \;\; 0 \le x \le 0.528 \\ 3.8 \cdot \left[x^{\frac{2}{\chi}} - x^{\frac{\chi+1}{\chi}} \right]^{\frac{1}{2}} & \text{if } 0.528 < x < 1 \end{cases} \tag{18}$$

Starting from this mathematical model, the dynamic behavior of the unit was simulated using SIMULINK. Figure 5 presents the software simulation scheme. Some relevant diagrams obtained from the simulation are presented in figures 6, 7, and 8.

4. THE EXPERIMENTAL SETUP

The conceived and developed experimental setup represents, by its hardware and software structure, a complex test system where all the experiments are aided by the computer. The hardware structure of the stand integrates high performance equipments. The application programs were developed using software specific to data acquisition and control (LabVIEW and FieldPoint).

One of the facilities of the system consists in the real time visualisation of the experimental results on the computer monitor, that is a high advantage for the experimental research. In the same time the results are stored in computer's memory as text files, for further processing and printing, without intervention of the human operator.

The experimental stand concretes the functional scheme presented in figure 1. Its construction is presented in figure 9.

The software application for the control of the system was developed using LabVIEW. Figure 10 presents the front panel of the application.

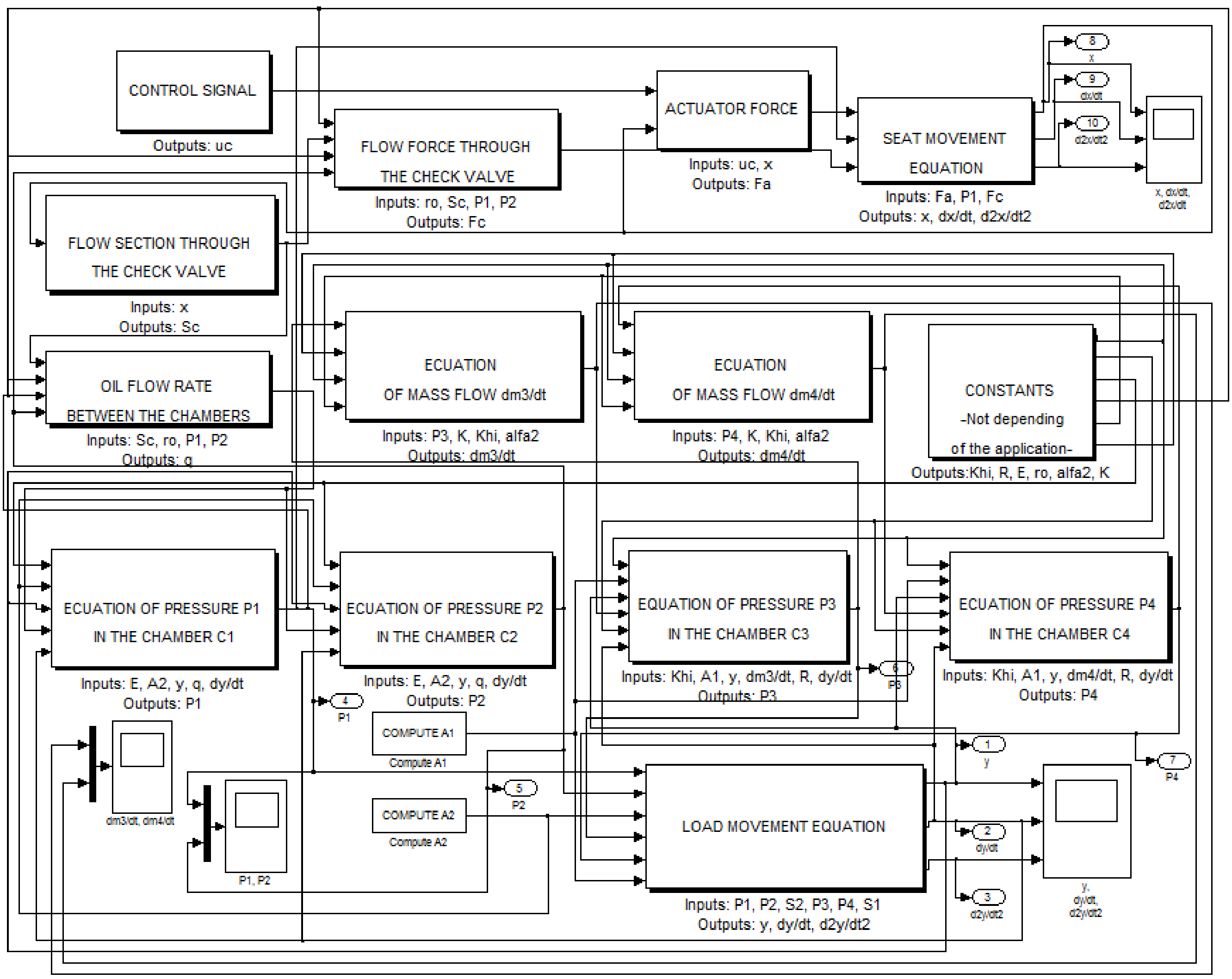

Fig. 5. Software simulation scheme.

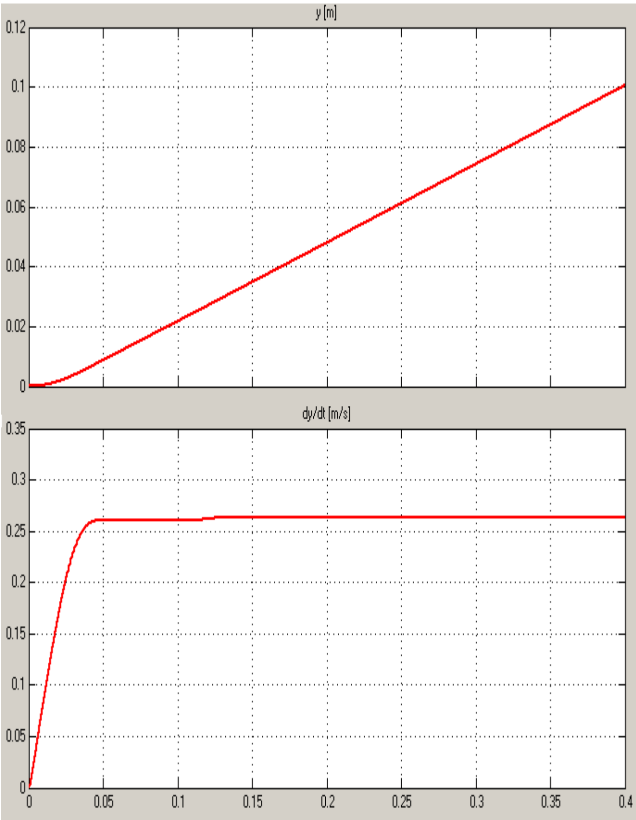

Fig. 6. Variation of load position and speed.

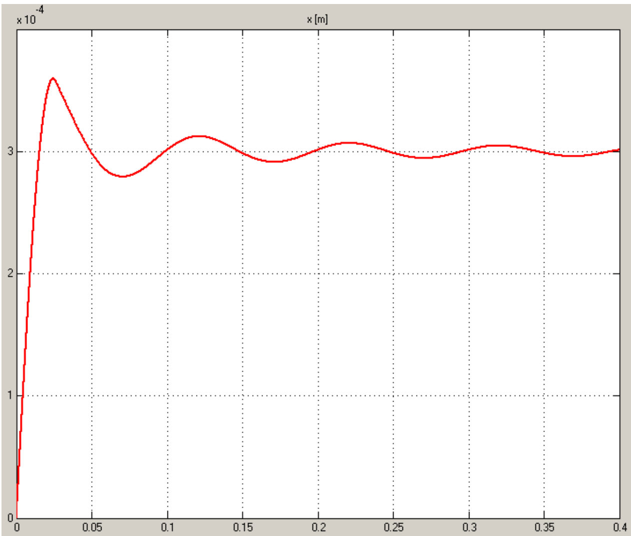

Fig. 7. Variation of check valve seat position.

The application allows the positioning of the load in any point of the work travel, with an imposed error. Besides, a matrix that contains the coordinates of the positions where the load has to be moved can be built and subsequently invoked. Each row of the matrix represents a position where the mobile unit has to be moved.

Using this application, the following tests could be performed: determination of positioning accuracy, determination of repeatability, study of the dynamic behaviour of the unit.

The first stage of the determination of the positioning accuracy consisted of the choice of points where the positioning error should be computed. In the example presented in figure 11, the positioning in the points y = 50, 100, 150 and 200 mm was intended. It has to be commented that the experiment could be performed for any number of points on the work travel.

The determination of the repeatability referred to the behaviour of the unit when it has to position the load in the same point for a number of times. An initial point and a final point of the load position were established.

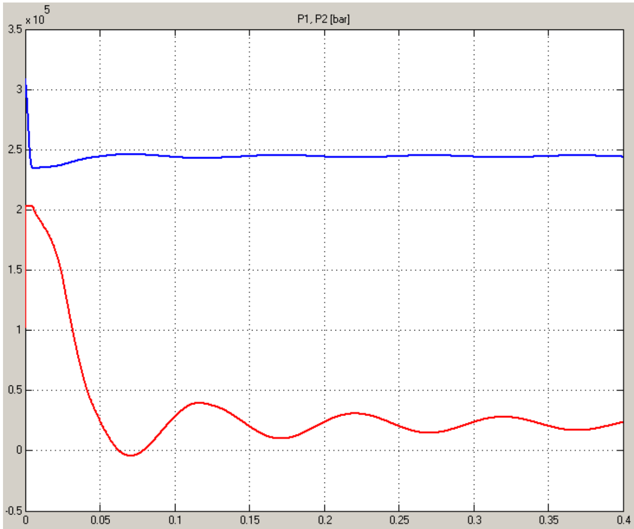

Fig. 8. Variation of the pressures in the chambers of the hydraulic cylinder.

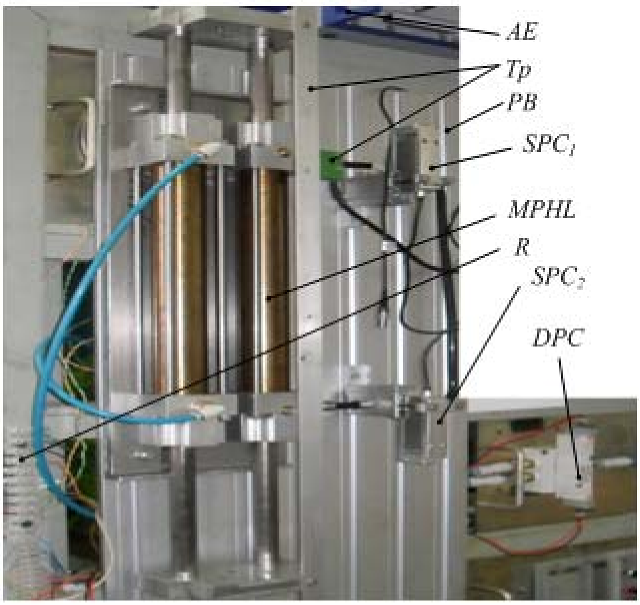

Fig. 9. The experimental setup.

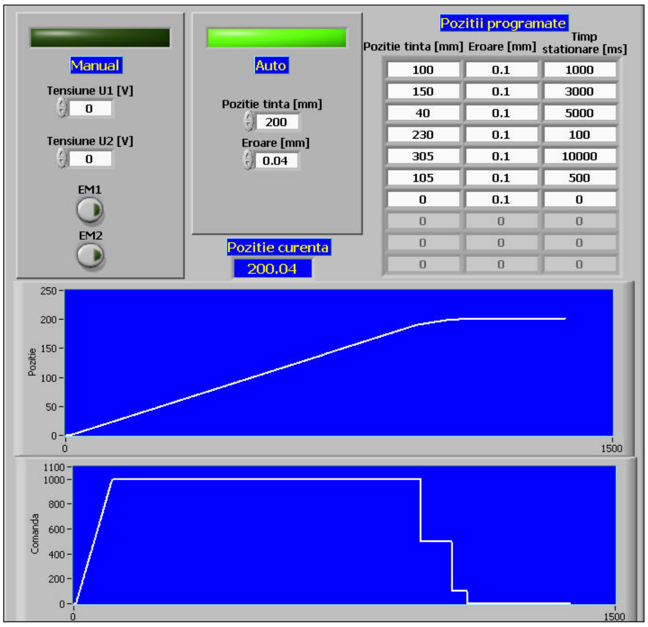

Fig. 10. The control panel of the application.

A target value of the programmed position (fig.11 – position *y* = 200mm) was established in order to study the dynamic behaviour of the unit. Initially, the unit was brought to the 0 position. The positioning of the load in the target position was achieved with an error of 0.04mm.

If only the target position was changed during similar experiments, there were situations when the positioning was achieved with overshoot. After some oscillations round the programmed position, the stop was achieved with the error imposed by the program.

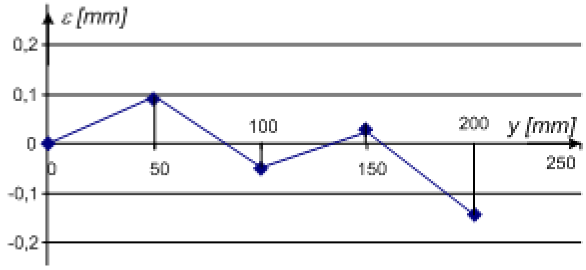

Fig. 11. Positioning accuracy of the unit in a number of given points.

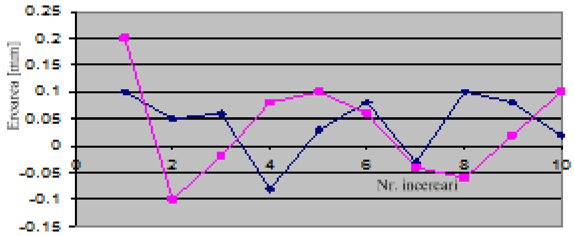

Fig. 12. Results of test performed in order to establish the precision of the unit.

REFERENCES

Avram, M., Bucşan, C., Bogatu, L., Constantin, V., Duminică, D. (2010). *Unit for Linear Pneumo-hydraulic Positioning*. Patent no RO125708-A0.

Belmut, O.G. (2007). *Design, Modeling, Identification and Control of Mechatronic Systems*. Doctoral Thesis. Barcelona.