ORIGINAL CONSTRUCTION AND MATHEMATICAL MODEL OF A PNEUMO-HYDRAULIC POSITIONING UNIT

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Abstract

The paper presents an original solution of pneumo-hydraulic positioning unit, as well as its mathematical model. The principle of the solution consists of the control of the position by using a hydraulic circuit governed by a proportional throttle. The unit contains a pneumatic cylinder with incorporated position sensor and a hydraulic cylinder whose rods are mounted in continuation. The load speed is thus rigorously controlled. Full stop is achieved by blocking the hydraulic circuit. The system is interfaced with a personal computer via specialized input-output electronic modules.

The dynamic behavior of the unit was simulated using SIMULINK. LabVIEW was used for developing control software. Experimental researches proved that the pneumo-hydraulic unit assured the positioning of the actuated load in any point of the working stroke with an accuracy of ±0.1mm.

Keywords: mechatronics, positioning system, pneumatics, control software

INTRODUCTION

Pneumatic positioning systems only in the past two decades have been associated with the accuracy concept. Until then, the main reasons of choosing such a positioning system were its simplicity, robustness, high reliability and the attractive price. In order to achieve a good positioning accuracy, the main disadvantages of the working fluid must be diminished though research: low viscosity and high compressibility. Those are the main barriers that stand against building advanced precise pneumatic systems.

Today more and more applications demand precise positioning, some only in specific points, and others in any point of the working range. There are different approaches for those situations. For the first category there are several existing solutions; all depends in how many stops are needed. The stop points can be the ends of the working range or intermediary points, marked by mechanical stop mechanisms. Once the stop points number rises, the positioning becomes more and more difficult. There are several special motors that can deal with this problem, but they had a low penetration into the market.

The problem that has not been solved yet properly is accurate positioning in any point of the working range. A way of overtaking this deadlock is the use of a hydraulic control circuit. In this situation, the unit becomes a pneumo-hydraulic unit.

In this context, the authors developed at the National Research and Development Center for Mechatronics, in the frame of "POLITEHNICA" University of Bucharest, an original solution of accurate pneumo-hydraulic unit, as part of a research project [1].

DESIGN AND CONSTRUCTION OF THE PNEUMO-HYDRAULIC UNIT

The principle of the solution consists of the control of the position by using a hydraulic circuit governed by a proportional throttle; in this way the liquid flow is accurately controlled. The unit speed is controlled rigorously and full stop is achieved by blocking the hydraulic circuit.

A series of general design considerations were observed during the design of the solution, among them being the following ones:

§ modular structure of the units;

§ use of actuators with integrated positioning sensors;

§ distribution equipment placed on the motor stator;
§ integrated control electronic circuitry;
§ PC-based electronic control system;
§ appropriate placement of various sensors in order to compensate the eventual perturbation factors.

The connection diagram of the proposed solution is presented in figure 1.

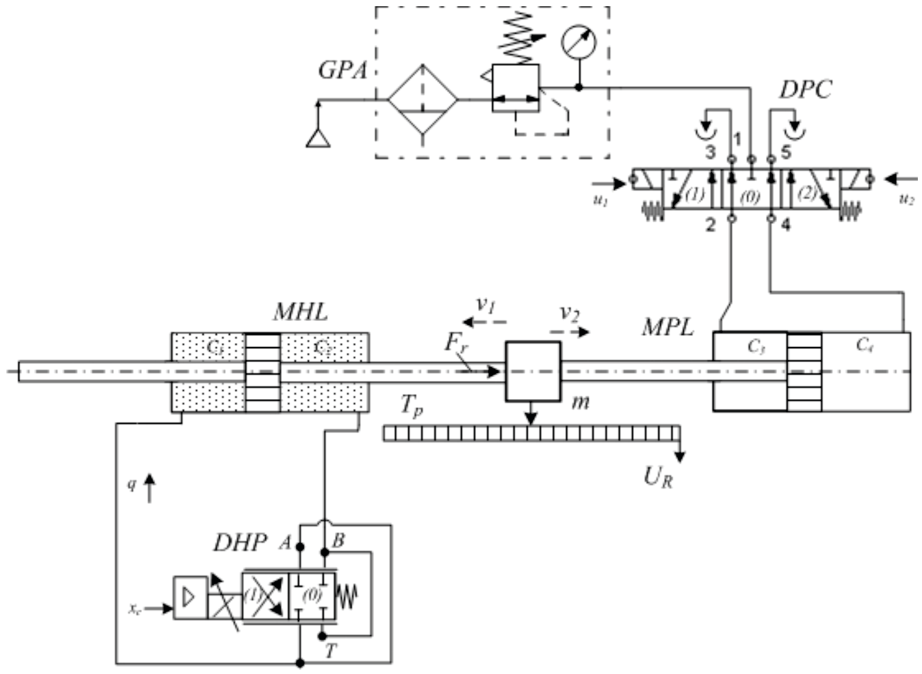

Fig. 1 The connection diagram

The following notations are used: MPL - Linear pneumatic motor; MHL - Linear hydraulic motor; DPC – Classical pneumatic direction control valve; DHP –Hydraulic proportional throttle; Tp - Position sensor; GPA - Modular air filter/regulator.

The construction of the pneumo-hydraulic unit is presented in more detail in [2].

Figure 2 presents the obtained physical model.

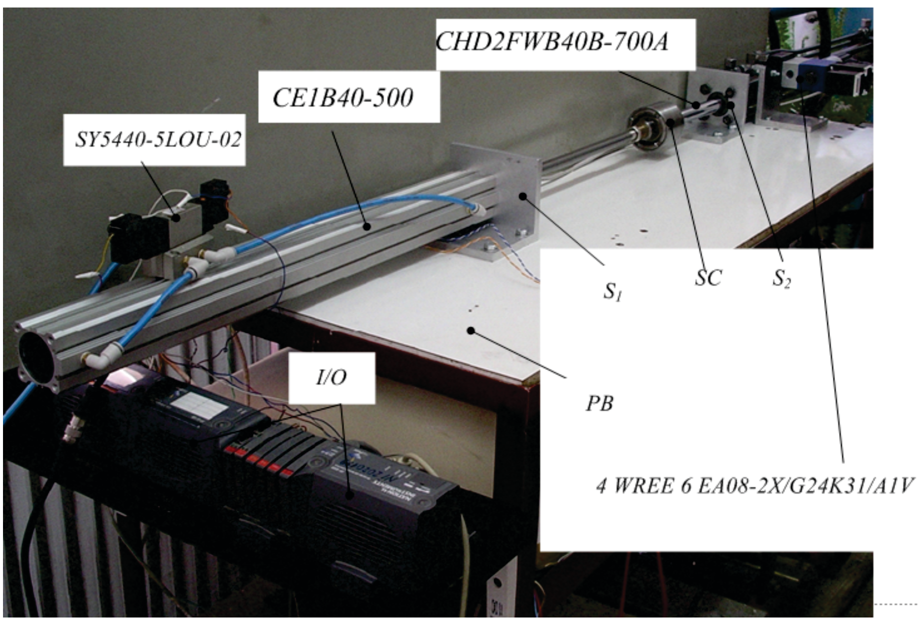

Fig.2 Physical model of the pneumo-hydraulic unit

The rods of the pneumatic cylinder with incorporated position sensor *CE1B40-500* and of the hydraulic cylinder *CHD2FWB40B-700A* are mounted in continuation and coupled via a special system *SC* that includes a spherical joint for the compensation of contingent deviations caused by non-coaxiality. The classical pneumatic direction control valve *SY5440-5LOU-02* is mounted directly on the sleeve of the pneumatic motor. The hydraulic control subassembly consists of the hydraulic motor with bilateral rod and the hydraulic proportional throttle *4 WREE 6 EA08-2X/G24K31/A1V*, mounted on the connection way between the two motor chambers. Other notations used in fig.2 are: S_1, S_2 – supports; *PB* – base plate; *I/O* - specialized input-output modules manufactured by National Instruments that allow the interfacing with the personal computer.

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 figure 1. Table 1 presents the used notations.

The movement of the mobile assembly will start if a flow section is established through the proportional throttle

The model consists of the following equations:

§ the equation that describes the movement of the load:

$$M \cdot \frac{d^2x}{dt^2} + B \cdot \frac{dx}{dt} + F_r = \\ = P_3 - P_4) \cdot S_1 - (P_2 - P_1) \cdot S_2 \qquad (1)$$

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

$$\frac{dP_1}{dt} = \frac{E}{V_1(x)} \cdot \left(\dot{q} - S_2 \cdot \frac{dx}{dt} \right) \qquad (2)$$

$$\frac{dP_2}{dt} = \frac{E}{V_2(x)} \cdot \left(-q + S_2 \cdot \frac{dx}{dt} \right) \qquad (3)$$

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

$$\frac{dP_3}{dt} = \frac{\chi}{V_3(x)} \cdot \left(\dot{m}_3 \cdot R \cdot T_a - S_1 \cdot P_3 \cdot \frac{dx}{dt} \right) \qquad (4)$$

$$\frac{dP_4}{dt} = \frac{\chi}{V_4(x)} \cdot \left(-\dot{m}_4 \cdot R \cdot T_a - S_1 \cdot P_4 \cdot \frac{dx}{dt} \right) \qquad (5)$$

Used notation	*Significance*
B	Damping factor
c	Maximum travel of the rods
d_n	Nominal diameter
D_s	Diameter of the throttle slide valve
E	Oil elasticity module
F_r	External force
K	Constant 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
Pa	Supply pressure
P_i	Pressures in the chambers of the motor; *i=1…4*
P_0	Atmospheric pressure
q	Oil flow rate through the throttle
R	Universal gas constant
S_i	Active sections of the motor, *i=1…4*
S_c	Flow section through the throttle
T_a	Absolute temperature
V_i	Volumes of the motor chambers, , *i=1…4*
x	Displacement of the two cylinder rods mounted in continuation
y	Displacement of the throttle slide valve
y_n	Nominal opening of the throttle
α_1	Shape factor
χ	Adiabatic factor
ρ	Oil density

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

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

$$V_1(x) = V_{10} + x \cdot S_2 \qquad (6)$$

$$V_2(x) = c \cdot S_2 + V_{20} - x \cdot S_2 \qquad (7)$$

$$V_3(x) = V_{30} + x \cdot S_1 \qquad (8)$$

$$V_4(x) = c \cdot S_1 + V_{40} - x \cdot S_1 \qquad (9)$$

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

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

The flow section through the throttle is equal to:

$$S_c=2\cdot\alpha_1\cdot\pi\cdot D_s\cdot y \tag{12}$$

The variation of the position of the throttle slide valve can be deduced from the characteristic diagram offered by the manufacturer [4], presented in figure 3:

$$y=\begin{cases} \dfrac{0.5}{0.04}\cdot y_n\cdot t & \text{if } 0\le t\le 0.04 \\ 0.5\cdot y_n & \text{if } t>0.04 \end{cases} \tag{13}$$

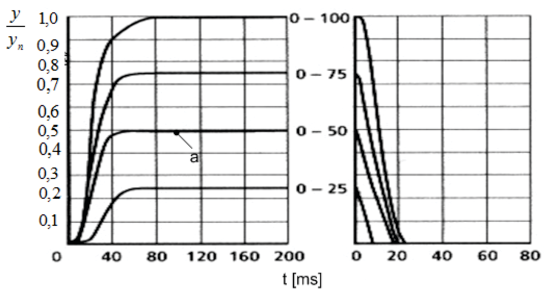

Fig. 3 Characteristic diagram of the throttle (available on the manufacturer's site)

The nominal opening yn that appears in (13) can be deduced from the continuity of the flow section:

$$\frac{\pi\cdot d_n^2}{4}=\pi\cdot D_s\cdot y_n \tag{14}$$

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 proportional hydraulic throttle DHP (fig. 1). Therefore the initial conditions of the model become, as shown in figure 4:

$$\begin{cases} P_3=P_a \\ P_4=P_1=P_0 \\ P_2=P_1+\dfrac{(P_3-P_4)\cdot S_1-F_r}{S_2} \end{cases}$$

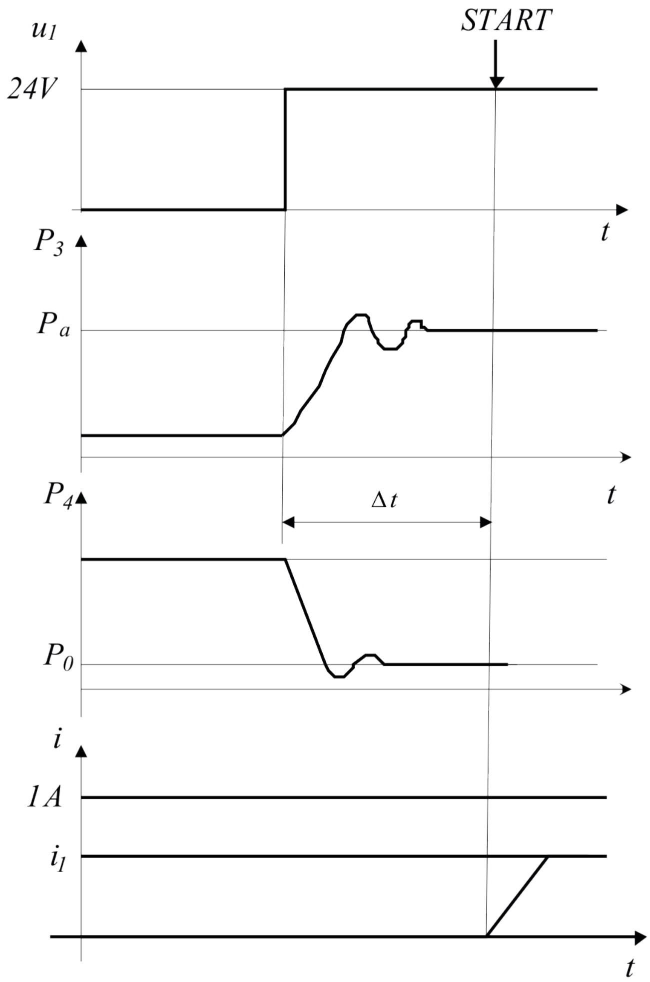

Fig. 4 Initial conditions

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

$$A_1=A_2=A_n=\alpha_2\cdot\frac{\pi\cdot D_n^2}{4} \tag{15}$$

The flow rates controlled by the DPC are equal to:

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

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

where:

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

CONTROL SOFTWARE

In order to perform experimental researches of the pneumo-hydraulic unit, the authors developed a series of four original subroutines using LabVIEW 7.1, each useful either in determining certain parameters of the unit or actually positioning it. The algorithms used as starting point for the routines are presented in [3].

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 and 7.

It can be noticed from figure 6 that, after a transitory period, the acceleration decreases to zero, leading to a constant-speed translation of the two cylinder rods mounted in continuation.

Figure 7 presents the variation of the pressures installed in the two chambers of the hydraulic cylinder.

Figure 8 shows the front panel of the application. For ease of use, the panel is divided into several areas. The left area consists of a Menu Ring list that allows the user to select which subroutine is to be used. The subroutines are as follows

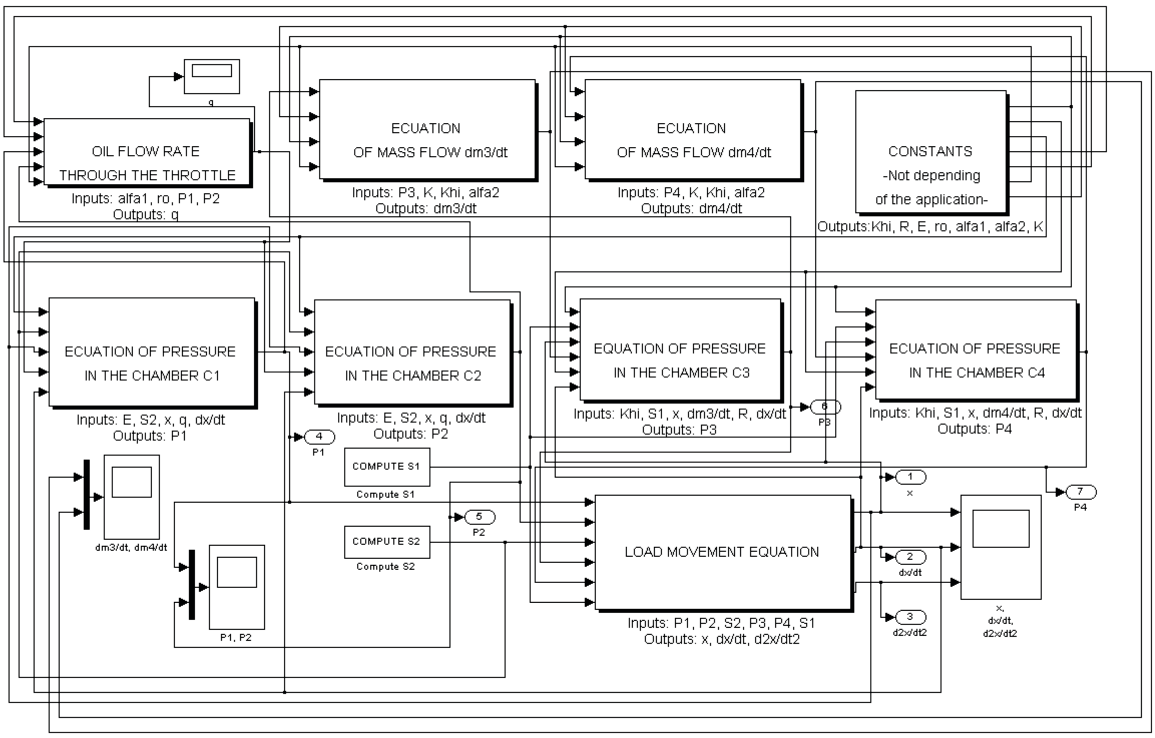

Fig. 5 Simulation scheme

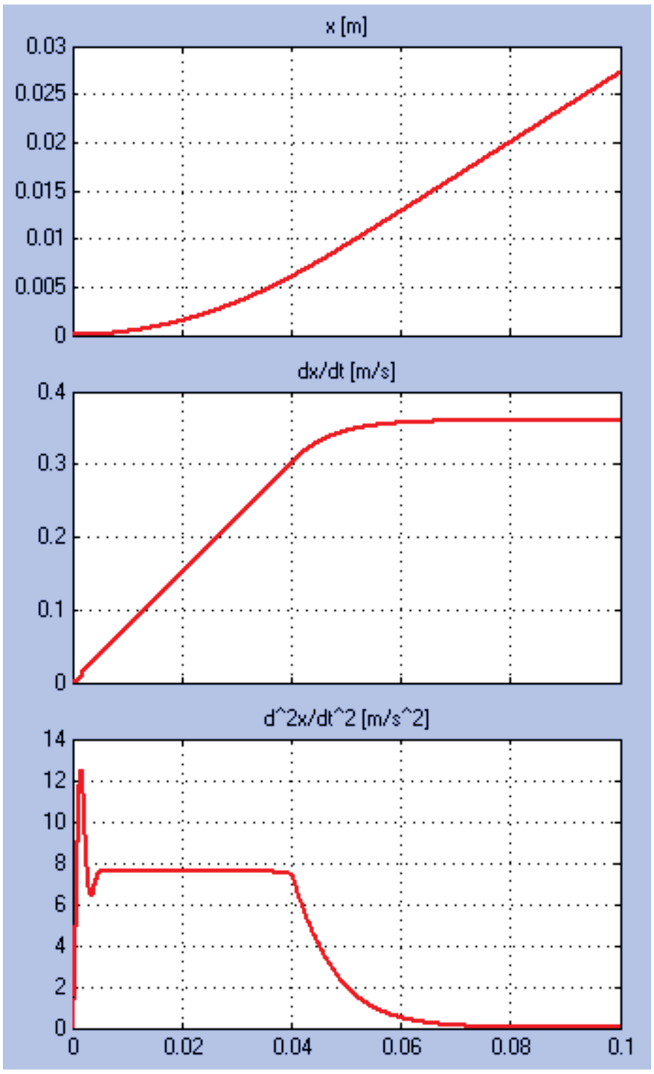

Fig. 6 Displacement, speed and acceleration of the two cylinder rods mounted in continuation

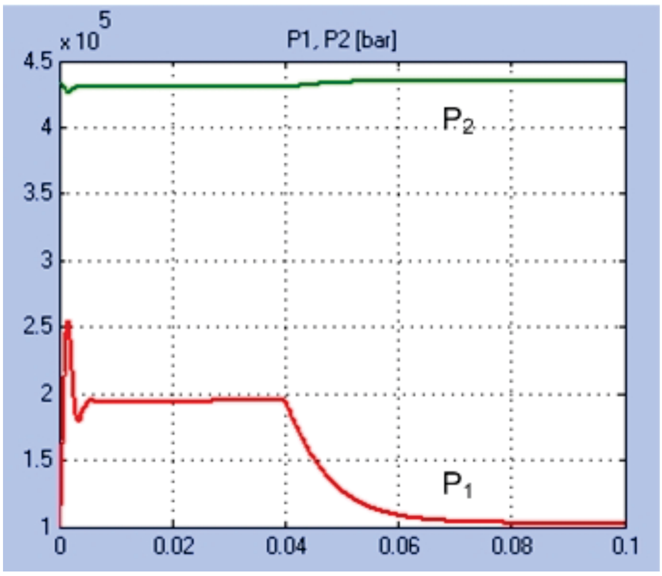

Fig. 7 Variation of the pressures in the two chambers of the hydraulic cylinder.

§ "Manual"; in this case, the central area will feature a three position toggle switch: advance – right position, backwards – left selector and stop – center position

§ "Return to 0"; the center area will display a LED indicator that turns on when the proximity sensor is activated.

§ "Optimum speed test"; for this option, the central area will reveal the controls relevant to the braking algorithm; the front panel will feature several numeric controls allowing the operator to set the input parameters for this subroutine: target position, the decrement for the command voltage per unsuccessfull attempt, the reference point for a new attempt (input is a percentage of the total distance)

§ "Auto"; upon launching this routine the program will perform the actual positioning algorithm; the central area of the front panel will feature several numeric controls (input parameters for the algorithm), a numeric indicator for the current position of the unit and a LED indicator activated when an appropriate control voltage has been found inside the results.txt file. Also, the area features a indicator that reveals the current state of the search progress of the results.txt file, file that contains the results of the "Optimum speed test" subroutine.

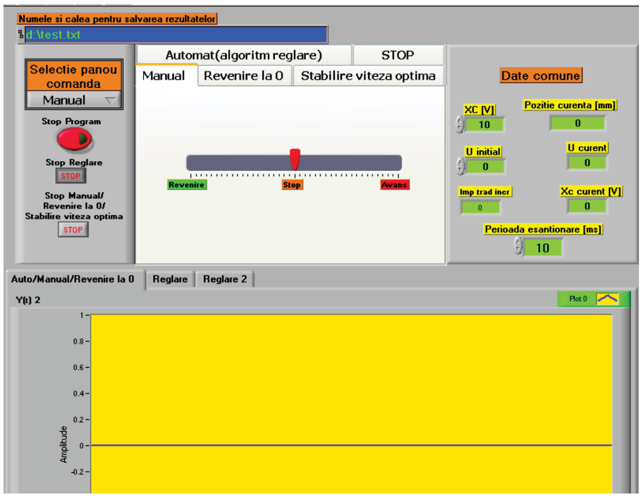

Fig. 8 Front panel of the application.

The left area also contains three on/off toggle switches:

§ "Stop program" that halts the program and all equipments involved in the process;

§ "Stop Auto"; this button is used only for stopping the "Auto" algorithm;

§ "Stop Manual/Return to 0/Optimum speed test"; the button is used for stopping any of the three algorithms, provided they are currently in use.

The right area contains three numeric controls used for setting the initial values of the maximum voltage for the hydraulic proportional control valve, the initial state of the two electromagnets that are part of the pneumatic way valve and the number of samples/second for the FieldPoint modules. Also, the area features four numeric indicators that show the current value of the control voltage for the proportional electromagnet, the current state of the electromagnets that control the pneumatic way valve, current position in mm.

Figure 9 shows a chart depicting the position of the unit during the "Optimum speed test" subroutine. Figure 10 features the movement of the load over time.

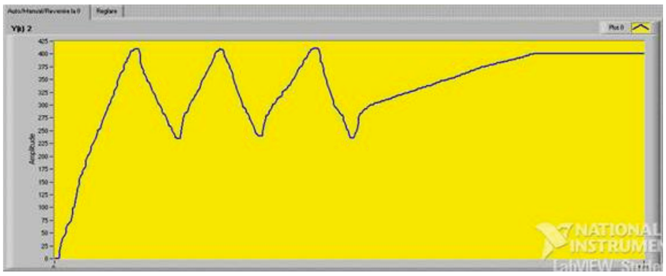

Fig. 9 Optimum speed test position chart.

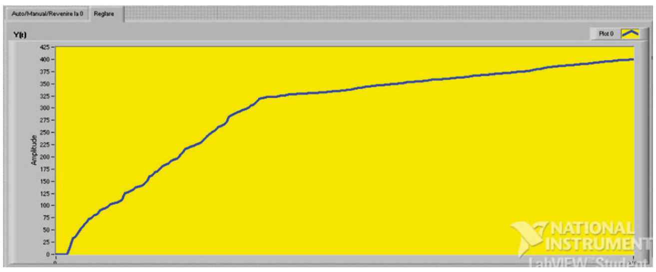

Fig. 10 Position chart for Auto subroutine.

For this test, the desired position was 400 mm and the initial position was 0 mm. The desired tolerance was mm5.0± and the positioning was done within mm2.0± of the target.

CONCLUSIONS

The experimental researches proved that the pneumo-hydraulic unit assured the positioning of the actuated load in any point of the working stroke with an accuracy of ±0.1mm. Further increase of the positioning accuracy can be obtained by developing appropriate control algorithms and software.

ACKNOWLEDGMENTS

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