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An Innovative Approach for Accurate Hydraulic Actuating

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Abstract: The paper presents a linear hydraulic positioning system with a special proportional distributing device which is different from a classic one (proportional directional valve, servo-directional valve) by containing a mechanical adding system, as a result of many years of theoretical and experimental research, with many originality elements, some of them being patented. Two positioning systems with mechanical position feedback are proposed. The mathematical model of such a system and the results of the numerical simulation are presented. Two experimental models were built and tested and the experimental results are presented.

Keywords: actuating, hydraulic, mechanical feedback.

1. INTRODUCTION

Many industrial robots are built by combining different rotation or translation positioning systems as mentioned in (Korem, 1985). Similar but simpler systems may be also used in some low complexity applications, such as positioning fabrics, parts, tools etc. with an imposed accuracy. In these cases is more efficient to use such systems instead of using a robot. This is why manufacturers developed such systems capable to position the actuated load, conforming to (www.atos.com, 2015).

A hydraulic positioning system consists of a hydraulic motor (linear, oscillating, or rotational), one or more proportional devices, an electronic amplifier for controlling the proportional device and a position transducer. The system has a compact structure. The proportional device is often mounted on the fixed part of the motor, the control electronics is integrated in the mechanical structure and the position transducer is integrated in the motor structure as mentioned in (Vasiliu and Vasiliu, 2005).

A simpler structure for a hydraulic positioning system can be obtained without the position transducer, which is expensive and influences the positioning accuracy by its resolution. The absence of the position transducer also leads to a simpler electronic control system. Such a hydraulic positioning system may have a mechanical position feedback using a wire transmission as mentioned in (Kitagawa et al., 2002), a metallic tape transmission as in (Wu and Kitagawa, 2004), or an opposite spring as in (www.alphafluid.de, 2015).

Such a system can be also used in applications for specific environments like explosion danger environments or very high temperature environments, for example controlling the valves of internal combustion engines, as mentioned in (Ukpai, 2007), where the feedback is obtained with a lever.

The paper presents a linear hydraulic positioning system with a special proportional distributing device as mentioned in (Tudor et al., 2002; Demian et al., 2002; Avram, 1997; Demian et al., 1990), which is different from a classic one (proportional directional valve, servo-directional valve) by containing a mechanical adding system.

The authors approached the subject both theoretically and experimentally, going over the following steps:

- studying the actual stage of the domain;
- establishing the technical solution;
- theoretical analysis: mathematical modeling and numerical simulation of the functioning;
- designing and building the experimental model;
- experimental analysis: designing and building of some experimental stands, establishing the tests list, designing, developing and implementing the working programs etc.;
- comparative analysis of the theoretical and experimental results;
- constructive and functional optimization;
- integrating the systems within different actuating structures.

An important part of the developed solutions are patented by the authors.

In most cases the systems used to actuate robots must accomplish the positioning of the actuated load with an imposed accuracy in some points of the working stroke. This also implies a rigorous control of the speed of the load.

Positioning the load in some points of the working stroke as mentioned in (www.victorycontrols.com, 2015) is different from positioning the load in any point of the working stroke as mentioned in (www.atos.com, 2015). In the first case the positioning is discrete and in the second case the positioning is continuous.

2. HYDRAULIC PROPORTIONAL DIRECTIONAL VALVES WITH DIFFERENTIAL CONTROL

The goal of the authors was to develop a new type of hydraulic proportional device in order to obtain the following functions: controlling the flow circuits, controlling the flow rate, stopping the motor, like a proportional directional valve and, more than that, adding two mechanical control signals of the same type (linear or angular displacements).

Fig. 1 shows the principle scheme of such a hydraulic proportional directional valve with differential control as mentioned in (Avram, 2005).

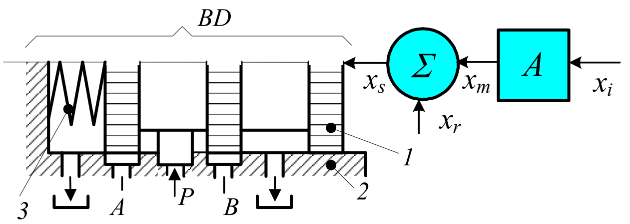

Fig. 1. The principle scheme of a hydraulic proportional directional valve with differential control.

The displacement x_m obtained at the output of the actuator A is not directly applied to the slide 1 as with the direct action proportional directional valves, but it is first added to the feedback growth x_r; the result x_s obtained at the output of the mechanical adding device Σ determines the flowing arias through the system.

Fig. 2 shows an example of a mechanical adding device.

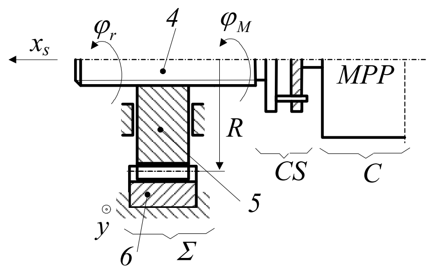

Fig. 2. An example of a mechanical adding device.

The mechanical adding device is a screw and nut subassembly and the actuator is a low power step by step electric motor. The feedback is transmitted to the adding device by a rack *6* which copies the movement *y* of the mobile assembly of the hydraulic motor controlled by the system, being engaged with the teeth machined on the outer surface of the feedback nut *5*.

The axial displacement of the feedback screw *4*, which is identical with the displacement of the slide *1,* is given by:

$$x_s = k_1 \cdot \varphi_M - k_2 \cdot y \tag{1}$$

where:

$$k_1 = P_s / (2 \cdot \pi) \tag{2}$$

$$k_2 = k_1 / R \tag{3}$$

After the actuator performed the programmed displacement φ_{M0} the slide must return into the initial position $x_s=0$ and the load displacement must be the programmed one, $y = y_0$.

This leads to:

$$k_1 / k_2 = y_0 / \varphi_{M0} \tag{4}$$

or:

$$R = \frac{y_0}{n_p} \cdot \frac{2 \cdot \pi}{N} \tag{5}$$

which is a condition that must be taken into account when the directional valve is designed.

Fig. 3 shows the variations in time of the linear displacements x_m and x_r as the adding device input data and x_s as the output of the adding device.

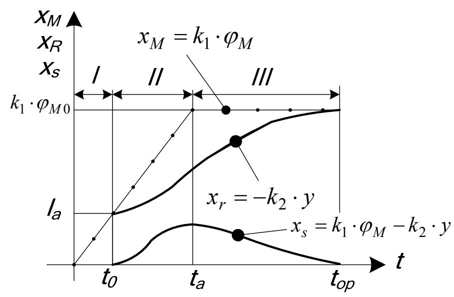

Fig. 3. The variations in time of the linear displacements.

There are three working stages of the system:

I - *the starting stage*: from the moment of the start command till the moment the slide displacement becomes equal with the overlap l_a ($t = t_0$); in this stage the feedback nut 5 is fixed because its lateral teeth are engaged with the feedback rack 6 which is fastened to the rod of the controlled hydraulic motor and now this is blocked (its chambers are blocked by the distribution block *BD*); a special coupling *CS* allows the axial displacement of the screw;

II - *the regime stage*: from the moment the hydraulic motor begins to move ($t = t_0$) till the moment the step by step motor stops ($t = t_a$ and $\varphi_M = \varphi_{M0}$, where φ_{M0} is the programmed angular position);

III - *the stopping stage*: from the moment the step by step motor stops till the moment the hydraulic motor stops ($t = t_{op}$).

Fig. 4,a shows the symbol of the system when the feedback rack and the actuator are placed on the same side and Fig. 4,b shows the symbol of the system when the feedback rack and the actuator are placed on opposite sides.

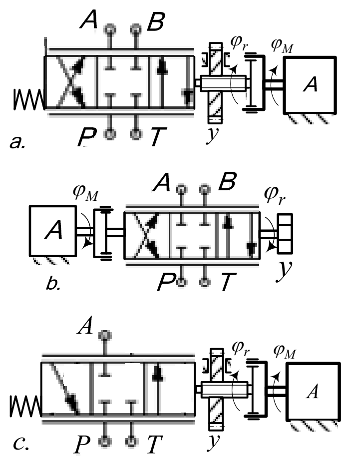

Fig. 4. The symbols of the system for: a. the feedback rack and the actuator are placed on the same side; b. the feedback rack and the actuator are placed on opposite sides; c. the device has only one consumer.

The variation low of the flow rates through the consumer orifices *A* and *B* can be:

- continuous, when the actuator is a *DC* servo-motor or a proportional electro-magnet, or
- incremental, when the actuator is a step by step motor.

A proportional hydraulic directional valve with differential control as in Fig. 4,a or Fig. 4,b, with two consumers A and B, implies very serious technological issues. It is much easier to make a device with one consumer like the one presented in Fig. 4,c.

3. POSITIONING SYSTEMS WITH MECHANICAL POSITION FEEDBACK

The positioning systems with mechanical feedback can be used in precision actuating. They were also used to develop some experimental models of positioning systems, further presented.

The principle scheme of such a system, proposed by the authors, is shown in Fig. 5 and contains the following devices:

- the linear hydraulic motor *MHL*;
- the proportional hydraulic directional valve with differential control *DHP**;
- the locking-unlocking valves block *BS* (Fig. 6);
- the feedback rack *CR*.

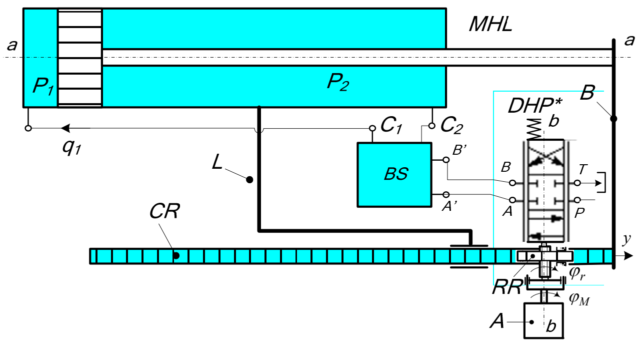

Fig. 5. The principle scheme of a positioning system with mechanical position feedback

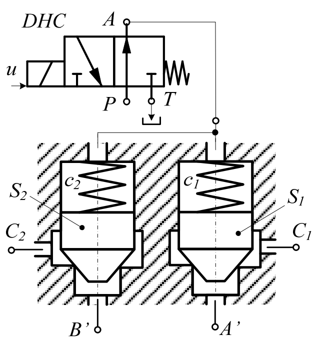

Fig. 6. The locking-unlocking valves block

The valves block *BS* consists of two identical conical valves S_1 and S_2 and a classic hydraulic directional valve *DHC*, and it has a protection role. It has two working stages:

- when the directional valve *DHC* is not controlled (the command voltage u=0), pressure is supplied to the upper chambers of the valves c_1 and c_2 and the valves S_1 and S_2 uncouple the active chambers of the motor; its mobile assembly and the load are blocked; this happens when shutting off the power of the control system, on purpose or accidentally;
- when the directional valve *DHC* is controlled (the command voltage $u \neq 0$), the upper chambers of the valves c_1 and c_2 are connected with the tank and the valves S_1 and S_2 couple the active chambers of the motor; its mobile assembly and the load are free; this happens when turning on the power of the control system.

As seen in Fig. 5, the longitudinal axis "*bb*" of the distribution device *DHP** is perpendicular to the axis "*aa*" of the linear hydraulic motor. A more compact structure is obtained if the two axes are parallel. This can be achieved using a conical gear transmission at the level of the adding device.

If the device has one consumer (Fig. 4,c) the principle scheme of the system becomes the one presented in Fig. 7. In this case, the chamber with the rod of the linear hydraulic motor *MHL* must be permanently connected to the pressure supply. The displacement of the mobile assembly occurs due to the different active areas.

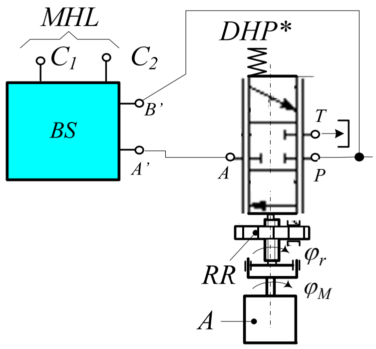

Fig. 7. The principle scheme of the system with one consumer.

The presented systems are linear. Using the same principle, rotational positioning systems with mechanical feedback can be developed.

Two constructive models of linear hydraulic systems with mechanical feedback as mentioned in Avram et al. (2011) will be presented further on.

Fig. 8 shows a view of the system having the principle scheme presented in Fig. 7 as mentioned in (Demian et al., 1990), with the following characteristics:

- the hydraulic linear motor *MHL* has four guiding ways *CG* on its body; the table *M* slides along these guidings by means of circulating rollers bearings; the feedback rack *CR* has one end articulated to the table *M* and it is pushed to the feedback wheel of the adding device ***Σ*** by means of a helicoidally spring mounted on the other end in order to compensate the backlash;
- the step by step motor has an integrated control electronic system, so only a clock signal and a

displacement direction signal are necessary; the angular step of the motor can be set as shown in table 1; for the built model a displacement of 1mm was imposed for an angular step of 1.8^0.

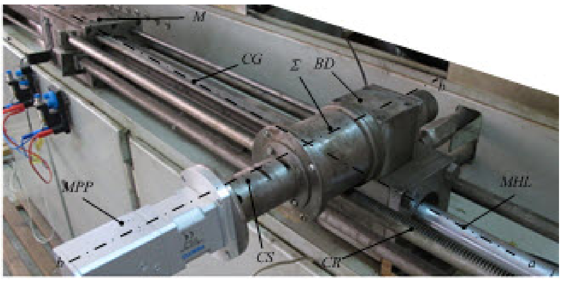

Fig. 8. A view of the system built according to the principle scheme presented in Fig. 7 (variant I)

In Table 1, column 3, the principal technical characteristics of the first built variant are presented.

Table 1. The technical characteristics of the built variants.

Technical characteristic	**Meas. unit**	***Variant I* (Fig. 8)**	***Variant II* (Fig. 9)**
Working stroke	mm	1200	600
Diameter of the cylinder	mm	50	40
Linear positioning increment	mm	0.1; 0.125; 0.2; 0.25; 0.5; **1**	1
Positioning accuracy	mm	±0.04	±0.04
Displacement speed	m/s	0.5...1	0.5...1
Displacement direction	-	forward-backward	forward-backward
Supply pressure	bar	100	100
Supply flow rate	l/min	40	40
Oil filter grade	µm	25	25
Max. working temperature	°C	+60	+60
Step by step motor couple	Nm	0.2	0.2
Angular step of the motor	°	0.18; 0.36; 0.45; 0.9; **1.8**	1.8

Fig. 9 shows a view of the system having the principle scheme presented in Fig. 7, with the following characteristics:

- the hydraulic linear motor *MHL* is a classic one;
- the feedback rack *CR* is fixed to the motor rod; in order to compensate the backlash the feedback wheel *RR* is made of two parts tensioned by two helicoidally springs;
- the step by step motor has an angular step of *1.8*°.

Fig. 9. A view of the system built according to the principle scheme presented in Fig. 7 (variant II).

In Table 1, column 4, the principal technical characteristics of the second built variant are presented.

4. THE MATHEMATICAL MODEL OF A POSITIONING SYSTEM WITH MECHANICAL POSITION FEEDBACK

In order to elaborate the mathematical model of the first variant of positioning system with mechanical position feedback (Fig. 8) the principle scheme presented in Fig. 10 and the characteristics presented in Table 1 were taken into account. In order to simplify the model only the displacement direction shown in Fig. 10 will be considered.

The considered system may be divided into two subsystems:

- a rotating subsystem consisting of the step by step motor, the special coupling and the feedback screw;
- a translating subsystem consisting of the piston of the linear hydraulic motor, the rod of the motor and the actuated load.

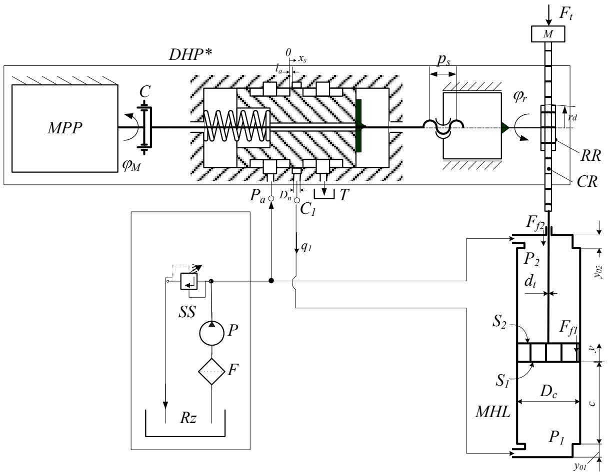

Fig. 10. The principle scheme for the mathematical model

The link between the two subsystems is made by the adding mechanism. In this case a very complex mathematical model is obtained as mentioned in (Avram, 1996).

A simpler model is obtained if the working equations of the step by step motor are neglected. Considering that the motor has the necessary couple to overcome the resistant couple once the number of steps n_p and the frequency f of the control signals are programmed, the variation in time of the rotation angle of the motor shaft φ_M is given by:

$$\varphi_M = \begin{cases} \dfrac{2\cdot\pi}{N}\cdot f\cdot t & \text{daca } t < t_a \\ \dfrac{2\cdot\pi}{N}\cdot n_p & \text{daca } t_a \geq t \end{cases} \tag{6}$$

where:

$$t_a = n_p / f \tag{7}$$

The following equations will be considered:

- the working equation of the adding device (1);
- the flow rate equation, given by:

$$q_1 = S_c \cdot w_c \tag{8}$$

where:

$$S_c = \begin{cases} 0 & \text{daca } 0 \le x_s \le l_a/2 \\ k_3 \cdot (x_s - l_a/2) & \text{daca } l_a/2 < x_s < x_n \\ S_n & \text{daca } x_n \le x_s \end{cases} \tag{9}$$

$$k_3 = \alpha_d \cdot \pi \cdot d_s \tag{10}$$

$$x_n = \frac{\pi \cdot D_n^2}{4 \cdot k_1} + l_a/2 \tag{11}$$

$$w_c = \sqrt{\frac{2}{\rho} \cdot (P_a - P_1)} \tag{12}$$

- the differential equation of the pressure in the working chamber C_1 of the cylinder, given by:

$$\frac{dP_1}{dt} = \frac{E}{V_1} \cdot (q_1 - S_1 \cdot \frac{dy}{dt}) \tag{13}$$

$$V_1 = S_1 \cdot (y_{10} + c + y) \tag{14}$$

$$S_1 = \pi \cdot D_c^2 / 4 \tag{15}$$

- the displacement equation of the mobile subassembly piston-rod-load, given by:

$$M_{red} \cdot \frac{d^2 y}{dt^2} + C_\eta \cdot \frac{dy}{dt} = P_1 \cdot S_1 - P_a \cdot S_2 - \\ - P_0 \cdot S_t - k_4 \cdot (P_a - P_1) - k_5 \cdot (P_a - P_0) - F \tag{16}$$

$$S_2 = \pi \cdot d_t^2 / 4 \tag{17}$$

$$S_t = S_1 - S_2 \tag{18}$$

$$k_4 = \pi \cdot \mu \cdot D_c \cdot b_c \tag{19}$$

$$k_5 = \pi \cdot \mu \cdot d_t \cdot b_t \tag{20}$$

where: C_η is the viscous friction coefficient, b_c and b_t are the breadth of the tightening fittings mounted on the piston and on the rod and μ is the friction coefficient at the tightening fittings level.

The mathematical model consists of nonlinear algebraically and differential equations of 1st and 2nd order; using substitutions as:

$$\frac{dy}{dt} = v \ , \tag{21}$$

the 2nd order differential equation is reduced to two 1st order differential equations.

Analytically solving of this system is not possible, so numerical solving methods are used, needing powerful computing systems.

The limit conditions are found out when studying the stabilized movement stage, characterized by:

$dy/dt = v_0$ and $d^2 y / dt^2 = 0$

where: v_0 is the speed of the mobile subassembly in the stabilized regime.

The displacement equation (16) becomes:

$$P_1 \cdot S_1 - P_a \cdot S_2 - P_0 \cdot S_t - k_4 \cdot (P_a - P_1) - \\ - k_5 \cdot (P_a - P_0) - F - C_\eta \cdot v_0 = 0 \tag{22}$$

This condition can not be accomplished if a variable force is applied to the rod, so further on the hypothesis of a constant force along the whole stroke is considered. This is possible only if the pressure P_1 is constant. In this case:

$$P_1 = P_{1,st} \ \text{and} \ \frac{dP_1}{dt} = 0 \tag{23}$$

The pressure in the active chamber C_1 in the stabilized regime is found from (22):

$$P_{1,st} = \frac{S_2 + k_4 + k_5}{S_1 + k_4} \cdot P_a - \frac{S_t - k_5}{S_1 + k_4} \cdot P_0 + \frac{F + C_\eta \cdot v_0}{S_1 + k_4} \tag{24}$$

If a large number of steps n_p is programmed, at a certain moment the movement will be stabilized and the flowing area of the directional valve remains constant at a value found from (13) and given by:

$$x_{s,st} = \frac{v_0 \cdot S_1}{k_1 \cdot \sqrt{\frac{2}{\rho} \cdot (P_a - P_{1,st})}} + \frac{l_a}{2} \tag{25}$$

Considering the initial conditions:

$$\begin{cases} t = t_a = n_p / f \\ y = \frac{k_1}{k_2} \cdot n_p \cdot \frac{2 \cdot \pi}{N} - \frac{1}{k_2} \cdot x_{s,st} \\ \frac{dy}{dt} = 0 \\ \frac{d^2 y}{dt^2} = 0 \\ P_1 = P_{1,st} \end{cases} \tag{26}$$

and the final conditions corresponding to $y = y_0$, the initial parameter $P_{1,i}$ for the evolution of the system can be determined.

The initial conditions for integration are the following:

$$\begin{cases} t = 0 \\ y = 0 \\ \frac{dy}{dt} = 0 \\ \frac{d^2 y}{dt^2} = 0 \\ P_1 = P_{1,i} \end{cases} \tag{27}$$

The pressure $P_{1,i}$ at the initial moment can also be measured, so the first integration can be skipped.

The mathematical model was integrated using Matlab Simulink programming environment. A working window can be opened where the principle constructive and functional parameters can be set in order to obtain the desired dynamic behavior. Fig. 11 shows the results obtained by simulation for D_c=0.04m, d_t= 0.025mm, d_s=0.016m and D_n=0.016m and for the functional parameters F = 0 N, $P_a=40 \cdot 10^5$N/m^2, n_p=10, and f=100 Hz.

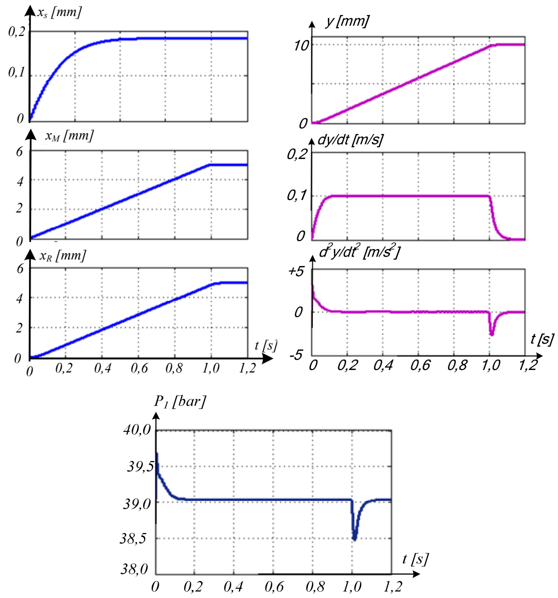

Fig. 11. The results of the simulation.

The results show that the programmed position is reached without override and with a positioning error depending on the hypothesis considered in order to simplify the mathematical model.

5. THE EXPERIMENTAL RESULTS

Special stands where designed in order to test the experimental models. As an example, Fig. 12 presents the functional scheme of the stand for the first variant of the system (Fig. 8). The interface of the experimental stand and the PC is performed by FieldPoint input/output modules *FP*.

A working program was developed using LabView in order to control the step by step motor (number of steps, direction of rotation and frequency of the impulses) and to acquire the signals from the transducers within the system.

The following parameters can be determined:

- the positioning accuracy;
- the repeatability;
- the dynamic behavior.

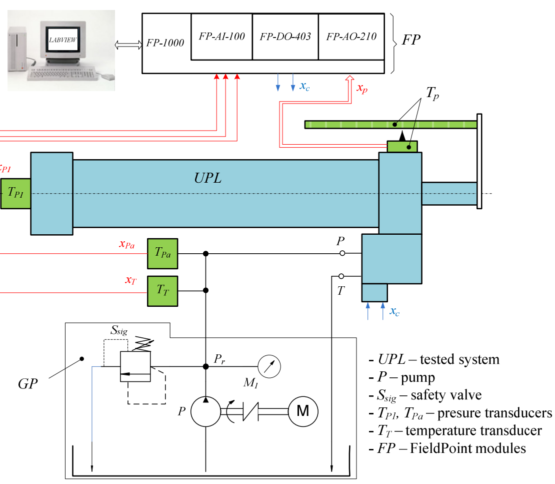

Fig. 12. The functional scheme of the stand for the first variant.

The results of the tests are displayed on the monitor in real time and they are also saved in text files, for further data processing.

5.1 Positioning accuracy determining

The actuated load can be positioned in a number of points $n = c_t / \Delta$ along the stroke, where Δ is the positioning increment. For every programmed position y_p the positioning error ε is determined by $\varepsilon = y_p - y_r$, where y_r represents the real position given by the position transducer *Tp*. Fig. 13 shows the results obtained for the first 200 points of the stroke. In this case the positioning error varies in the range of $[-0.05, +0{,}12]$mm. These errors are the consequences, on one hand, of the machining accuracy of the mechanical parts involved in signals transmitting and processing on the direct path, from the step by step motor to the screw of the adding device, and also on the feedback path, from the linear hydraulic motor to the feedback nut, and on the other hand, of the overlap with an adopted value of $20\ \mu m$.

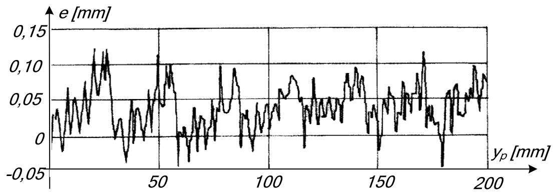

Fig. 13. The results obtained for the first 200 points of the stroke.

5.2 Repeatability determining

In order to determine the repeatability of the system, a number *i* of measurements are performed; for every measurement the errors $y_f - y_{f,r}$ and $y_i - y_{i,r}$ are computed.

Fig. 14 shows the results for 20 working cycles. The reference position is $y_i = 200mm$ and the target position is $y_f = 280mm$. The obtained repeatability is $\pm 0{,}1mm$.

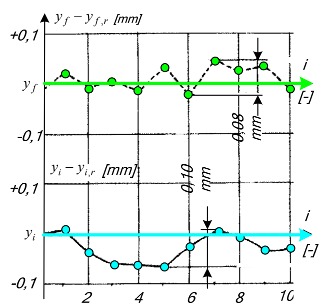

Fig. 14. The results for 20 working cycles

5.3 Dynamic behavior determining

In order to determine the dynamic behavior of the tested experimental model more sets of measurements were performed, modifying the input parameters. Figures 15 and 16 show two examples of the results obtained for different situations.

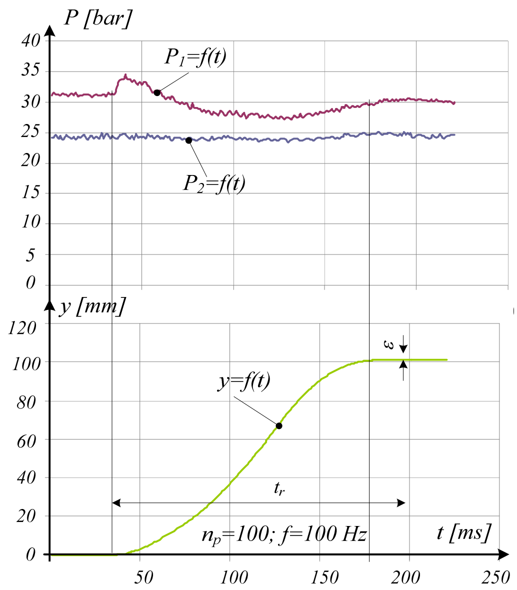

Fig. 15. The experimental results for y=100mm.

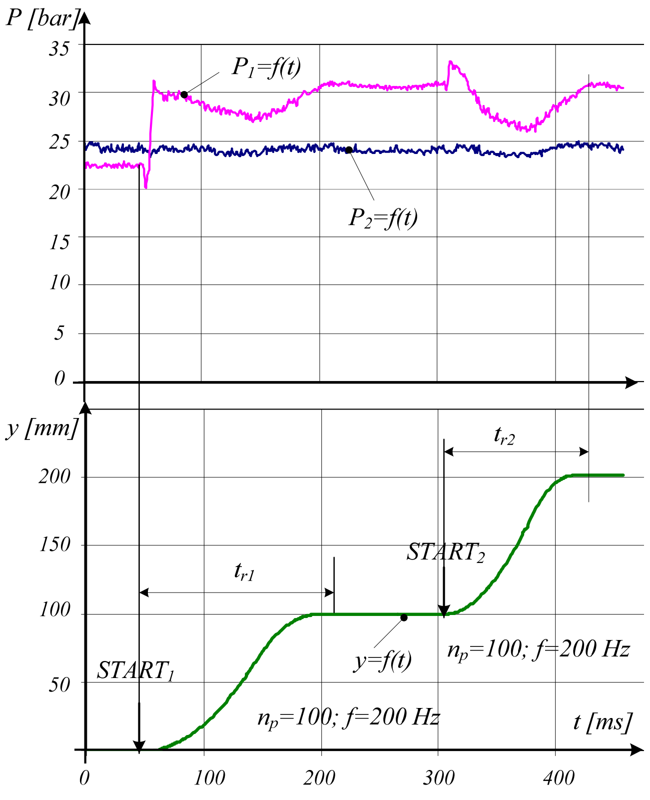

Fig. 16. The experimental results for y_1=100mm and y_2=200mm.

The results show that the positioning of the load is performed without override and the positioning error is less than $\pm 0.05mm$.

The comparative analysis of the theoretical and experimental results leads to the following conclusions:

- the experimental results are in concordance with the theoretical ones; the differences are the consequence of the simplifying hypothesis adopted for the mathematical model, the uncontrolled physical phenomena and the accuracy of machining and assembling (overlap, clearances, other factors that influence the accuracy of the control and feedback signals);
- the choosing of the working parameters (pressures, flow rates, frequencies, temperatures etc.) is of high importance, due to their major influence on the performances of the system; the optimum set of values is determined experimentally;
- a speed higher than the one corresponding to the limit frequency given by the step by step motor characteristic leads to loosing steps; this also happens when the needed torque is higher than the motor torque; in these cases the positioning is not possible;
- when positioning high weight loads the programmed position is outran; the connections of the distributing block are inverted, so the movement direction is inverted, this being an override that allows the desired positioning with the same errors.

The stand can be used, with minor changes, for testing the second variant of the positioning unit (Fig. 9).

6. EXAMPLES OF ACTUATING SYSTEMS WITH TWO AND THREE AXES

Acording to the desired application, the linear positioning system as presented in Fig. 8 and Fig. 9 can be combined in order to obtain actuating system with two, three or more axis.

Fig. 17 shows a view of an actuating system with two axes, obtained by combining the two experimental models theoretically analyzed in §3 and experimented in §4:

- the *Ox* axis is a hydraulic positioning system *SPH_Ox*, like the one presented in Fig. 8;
- the *Oy* axis is a hydraulic positioning system *SPH_Oz*, like the one presented in Fig. 9.

The system has a third axis, *Oz*, which can position the load only in two points of the stroke (the stroke ends of the hydraulic motor) and a device *DOA* for gripping and orienting the load. Several experimental tests were performed using this structure and the results confirmed the functioning stability of the two systems in real working conditions.

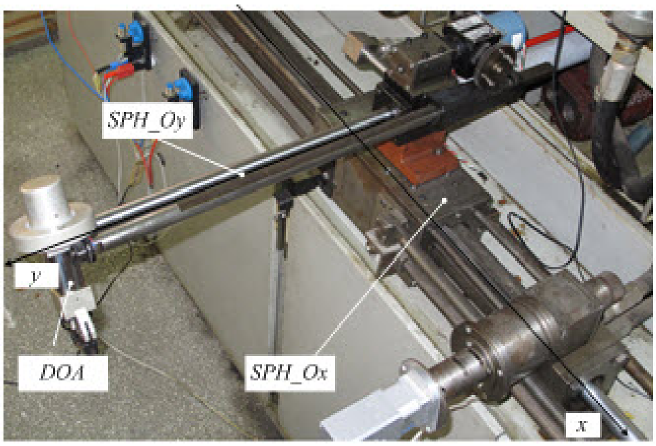

Fig. 17. A view of the actuating system with two axes.

Fig. 18 shows a view of a hydraulic robot as mentioned in (Demian et al., 1988).

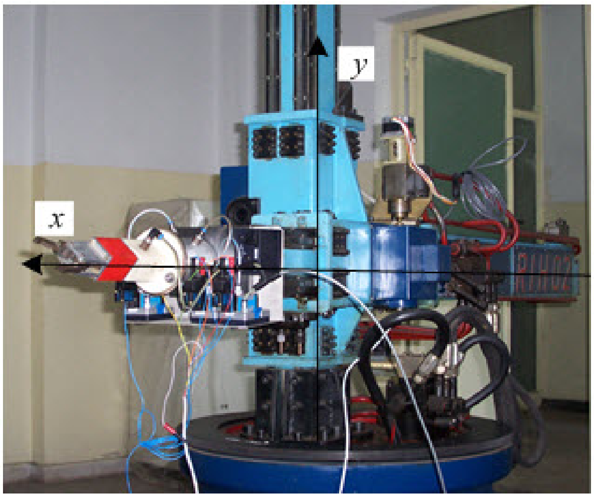

Fig. 18. A view of the hydraulic robot

It consists of three hydraulic positioning systems with mechanical feedback, one for each mobility axis.

The robot has two translation axes (along the *Ox* and the *Oy* axes) and a rotation (around the *Oy* axis). The rotation is obtained using a linear positioning system as described and a gear-rack mechanism. This time the three actuating systems are integrated within the structure of the robot.

The characteristics of the robot are the following:

- number of mobility degrees: 3;
- working space: $x = 1200mm$, $z = 800mm$, $\varphi_z = 360^o$;
- maximum actuated load: $25daN$;
- moving speed: translation $0{,}6m/s$, rotation $180^o/s$;
- positioning error: $\pm0{,}2mm$;
- displacement increment: 1 mm;
- weight: $500Kg$.

The control of the robot is performed using a working program developed using LabView environment.

7. CONCLUSIONS

The hydraulic positioning systems with mechanical feedback represent a viable variant to the systems with electric feedback. They have a simpler structure and needs a lower complexity control system. The control of these systems implies only the command of the actuator within the structure of the proportional hydraulic directional valve with differential control. According to the characteristic of the actuator – continuous or incremental - the positioning of the load can be performed in any point of the working stroke or in a finite number of points, imposed when designing the system. Such a system can be also used in applications for specific environments like explosion danger environments or very high temperature environments.

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