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MECHATRONIC SYSTEM FOR PRESSURE CONTROL

The mechatronic system, that has been presented in this article, is a force adjustment system through pressure control. As a practical application of this system is the adjustment of the pressing force on a technological line for manufacturing rubber carpets on a textile support.

The main condition, required by the technologic process, is to maintain a constant value for the pressing force during the entire process, in order to maintain a constant thickness of the rubber carpet.

The mechatronic system is formed from two sub-systems, a hydraulic actuating sub-system and an electronic driving system (servo-controller). The electric signals from the transducers are sent to the servo-controller, for processing. The servo-controller compares the signals with pre-programmed values and drives the electromagnets from the proportional directional valve. By driving the proportional directional valve, the hydraulic system is driving the hydraulic cylinder, resulting the real pressing force, required by the parameters of the technological process.

1. INTRODUCTION

Electro-hydraulics systems for physical parameters adjustment (like force, torque, speed, pressure, flow control) can be found in hydraulic actuation systems, which are frequently used in making of mobile or stand-alone machinery in various fields of activity.

This article is referring to a mechatronic force adjustment system through pressure control.

A mechatronic system for force adjustment through pressure control can be found at a technological rubber band line with textile support (fig.1.b). A line like this is used to manufacture rubber carpets with textile insertions (with various uses in many technical fields).

Depending on the market requests, rubber carpets with textile insertions are manufactured in various sizes.

In order to obtain a constant thickness, the pressure force must be also constant.

A constant value for pressure force is obtained by adjusting the pressure in the pressing cylinder. Pressure control is made by a mechatronic force adjustment system thorough pressure control.

2. HYDRAULIC SYSTEM STRUCTURE

The hydraulic actuating system is represented in figure 1a. This system contains a pumping group, a proportional directional valve and a hydraulic cylinder. On the supply circuit is mounted an accumulator tank which smoothes the flow of the fluid and on the cylinder circuit is the group of valves for over-pressure discharging. On the same circuit we can find the pressure transducers (P_1 and P_2) and a movement transducer (x).

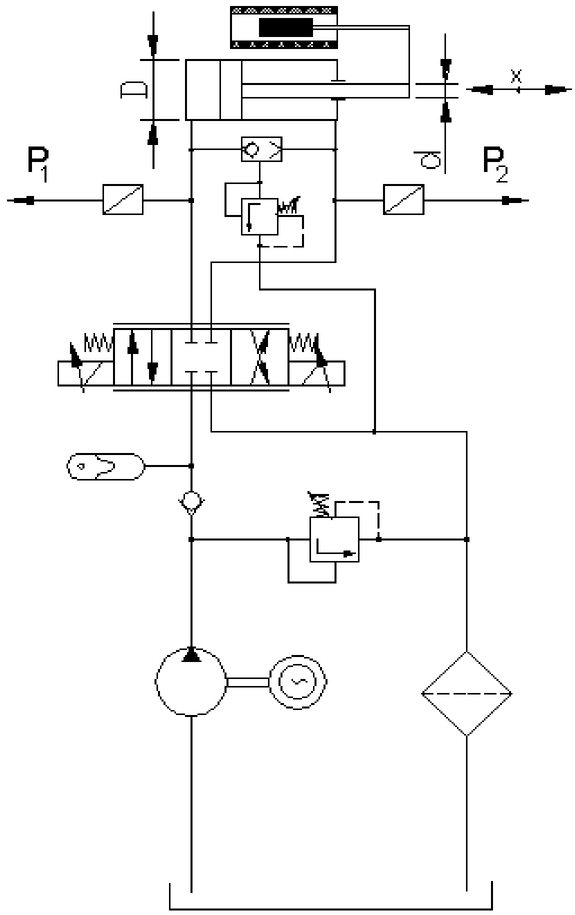

Fig. 1. a – Force adjustment hydraulic system

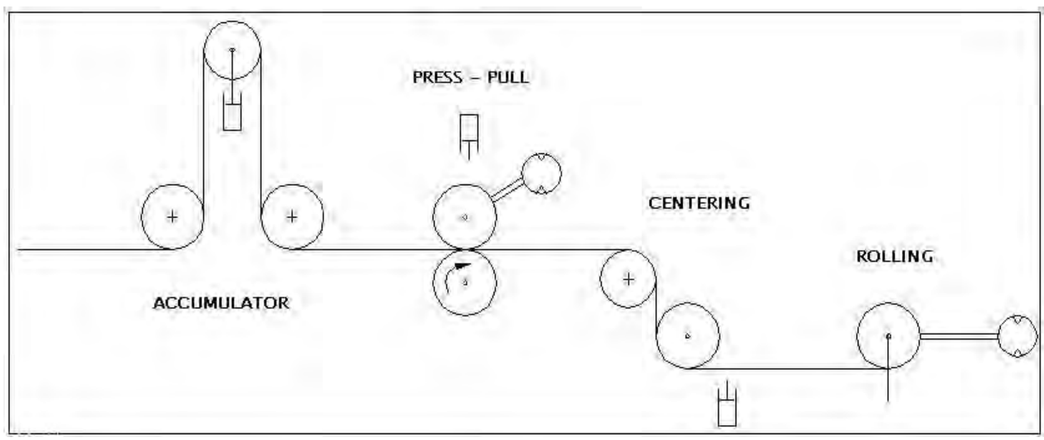

Fig.1. b – Technological rubber band line

Pumping group supplies hydraulic energy in order to operate the hydraulic cylinder. Pressure transducers inform the electronic system about the values of the two pressures from the cylinder's chambers. The movement transducer informs the electronic system about the relative positions of the piston rod. The electronic device of the system compares the values of the pressing and retracting pre-programmed forces and the values of the pressures P_1 and P_2. Furthermore the servo-controller is driving the electromagnets of the servo directional valve (proportional directional valve). In this way we can control the values of the forces through the pressure control.

Forces equations depending on the pressures are:

Formulas:

$$F = P \cdot A \tag{1}$$

$$A = \frac{\pi \cdot D^2}{4} \tag{2}$$

$$A = ct. \tag{3}$$

$$F = k \cdot \Delta p \tag{4}$$

$$F = p_1 A_1 - p_2 A_2 \tag{5}$$

$$A_1 = \frac{\pi \cdot D^2}{4}; \qquad A_2 = \frac{\pi\left(D^2 - d^2\right)}{4} \tag{6}$$

Where:

F – force	k – coefficient	D – piston diameter	
P – pressure	d – rod diameter	A – area	Δp – pressure drop

3. ELECTRONIC SYSTEM STRUCTURE

The electronic driving system is presented in the figure 2. This system has a force and movement programming device P, a movement comparator Cx, a force comparator CF, a proportional-integrator amplifier PI, a proportional-integrator-derivative amplifier PID, a polarity switch, two switch-mode converters U/I (PWM), two pressure transducers converters (P/U), a movement transducer converter (x/U) and an operational amplifier.

The programmer sends electric signals to the force and movement comparators. The electric signals from the pressure transducers are sent to the operational amplifier through the P/U converters. From this point, the electric signal is directed to the force comparator, CF. The signal from the movement transducer is sent to the movement comparator, through the x/U converter. The force value adjustment is given by the formula: $CF = PI - Cx$. From the Cx comparator, the electric signals are directed to the input of the PID amplifier and to the polarity switch. From this point the signals are sent to the U/I converters. In the end, the electric signals are directed to the electromagnets, which drive the proportional directional valve.

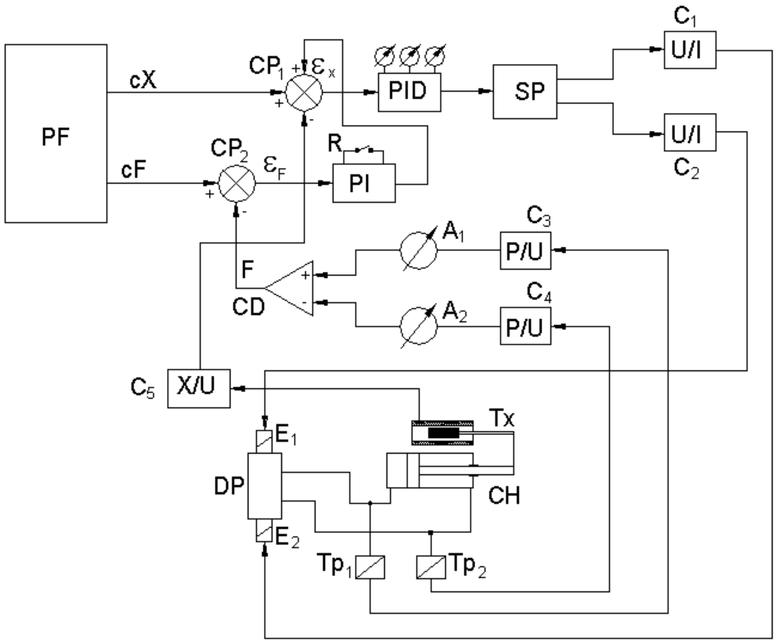

Fig. 2 - Electronic system structure

For system's analysis, figure 3 presents a mathematical model of the positioning system:

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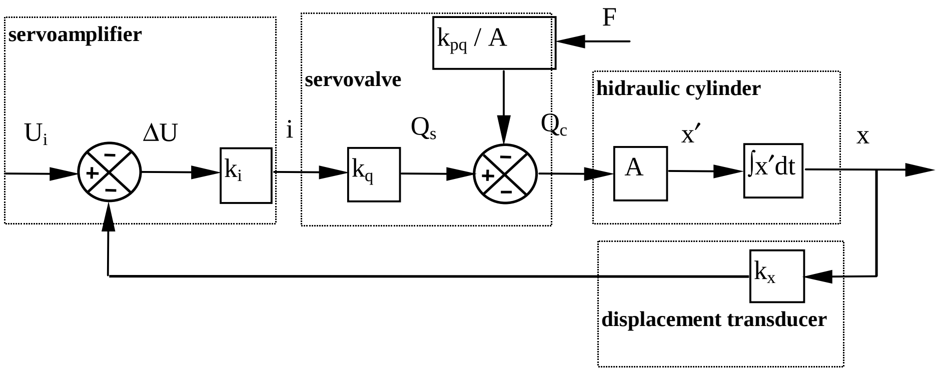

Fig. 3 - Mathematical model of the position control

Because both the proportional element and the hydraulic cylinder can be represented as second order execution elements, it results that the whole system is of fifth order. In practice because the self throb of the proportional element ω_v is much greater than that of the load ω_L , the system reduces at the third order with the following transfer function:

$$T(s) = \frac{1}{1 + \frac{1}{G_x} \cdot s(1 + \frac{2 \cdot \xi}{\omega_n} \cdot s + \frac{1}{\omega_n^2} \cdot s^2)} \tag{7}$$

where A is the piston area, ω_n is the natural throb, ξ is the attenuation coefficient, and G_x is the system's gain, respectively:

$$G_x = \frac{k_i k_q k_x}{A} [s^{-1}] \tag{8}$$

The systems precision is:

$$\Delta x = \frac{k_{pq}}{G_x A^2} F \tag{9}$$

These considerations lead to a first conclusion: in order to achieve a precise and rigid system has to be chosen the smallest value for the nominal flow and the greater piston area as possible. On the other side, as the positioning systems is conceived to remain stabile, the magnitude can not overlap a critical value G_x:

$$G_x < G_c = 2\xi\omega_n \tag{10}$$

The use of the time integral - absolute error method leads to the conclusion that the ratio G_x / ω_n is among 0.25 and 0.35 for typical values of the attenuation coefficient among 0.2 and 0.9. Therefore can be pointed out the second conclusion: the optimal magnitude is:

$$G_{xopt} = \frac{1}{3}\omega_n [\ s^{-1}] \tag{11}$$

From those pointed out until now results that the performances of a position regulation system (precision and stiffness) could not be raised infinitely for a precise system configuration.

Although there can be brought some improvements such as:

1. Introducing a supplementary speed loop, with the time constant $T = 1/\omega_n$ makes possible the optimal gain increasing and therefore of the stiffness with maximum 50 % (commonly 30-40 %).

2. Using a PD regulator leads to the possibility of enlarging the gain with 100%, if ω_D=(0.5..0.7) ω_c. It is though necessary the use of a filter low pass with cutting frequency of ω_Y=5 ω_D.

3. Introducing an integrative component into the regulator makes, in theory, possible obtaining an unlimited precision and stiffness, for a very low load resonance throb, but there can appear magnitude oscillation in the system.

Another applications require a force control of the hydraulic system. Comparatively with the position control system, this system is easy to optimise.

In figure 4 it present simplified mathematical model of the force control system.

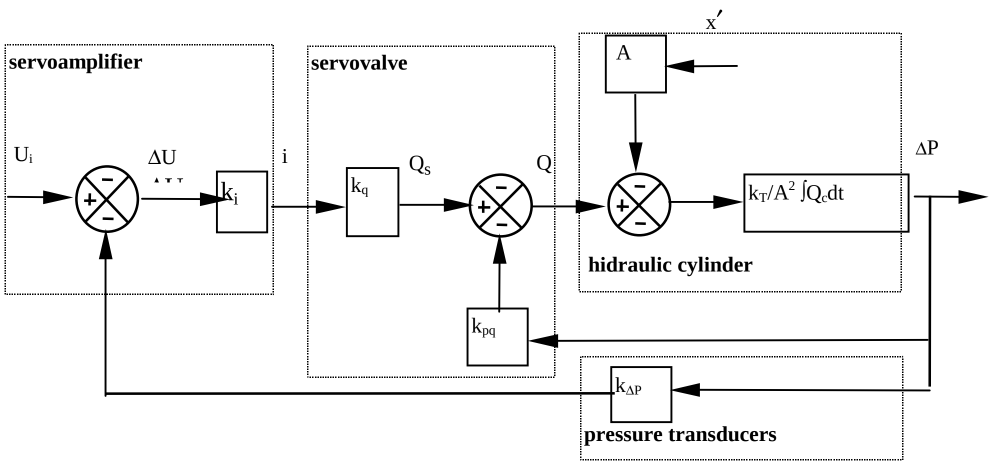

Fig.4. - Mathematical model of force control

4. TESTING

The authors made in the institute’s testing laboratory a series of experiments on the testing desk showed in figure 1.

The electronic module, designed and constructed by them, admits adjust of the position loop to obtain an optimal frequency response (see fig. 5). This is possible due to use a PID position error amplifier. The maximum speed at step command is limited by the ratio of maximum flow and cylinder’s active area (see fig.6).

The system’s force control is possible in two mode, without disconnect position control, used a secondary PI force error amplifier. So, if the load is a rigid bar, the error signal corrects the position reference signal to obtain the desired force (see fig. 7).

If the load is a shock absorber, the error signal corrects the slope of position reference signal to obtain the desired force (fig. 8).

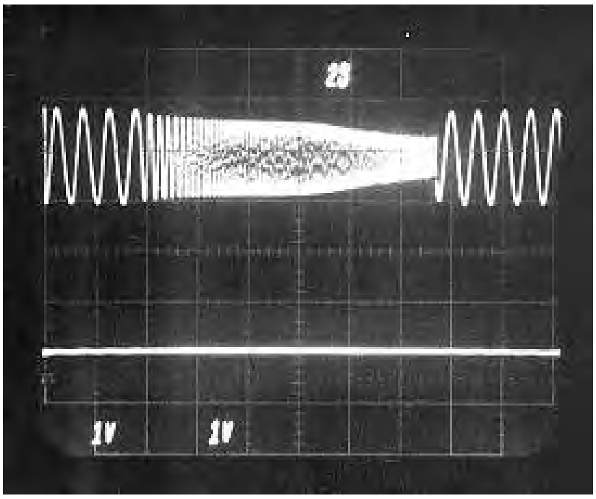

Fig.5 - Frequency response (1 to 30Hz) of the positioning system, without load (up –displacement, down – force)

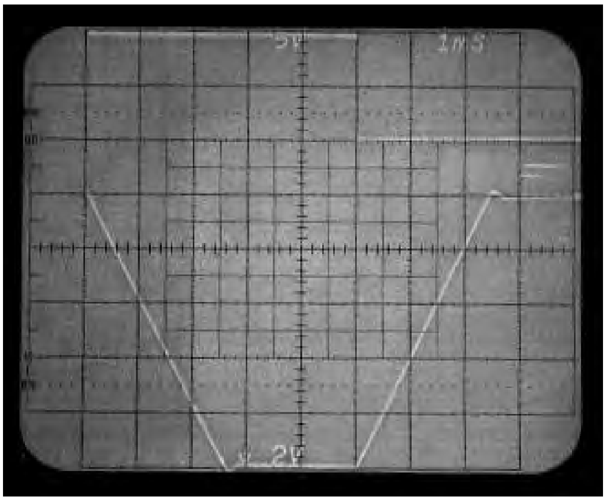

Fig.6 - Position loop response, at maximum system’s speed, without load (up-command, down-displacement)

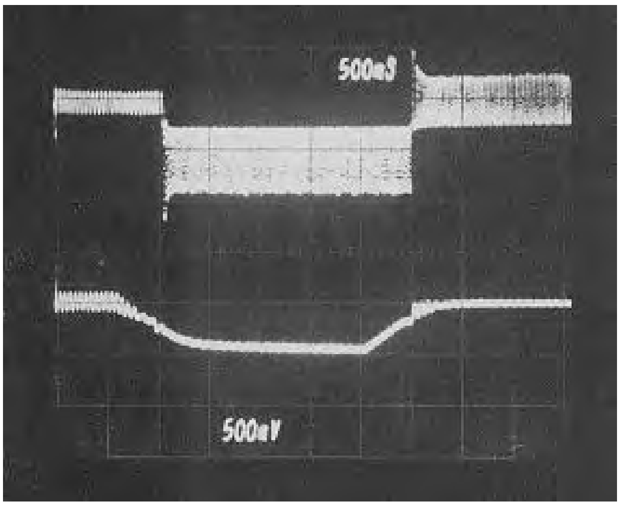

Fig. 7 - Force control system’s response with rigid load (up – force, down – deformation)

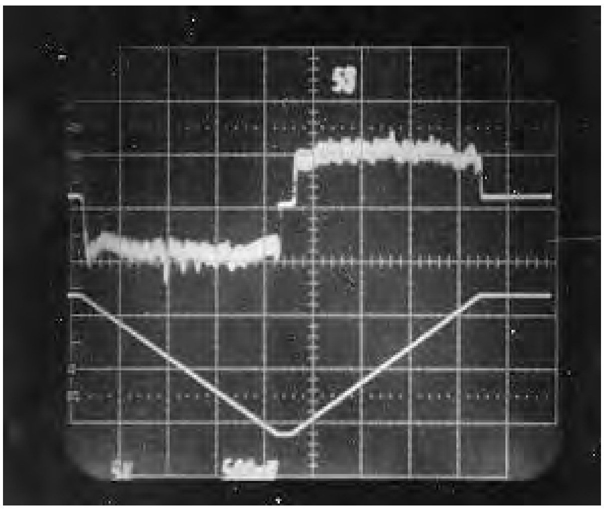

Fig. 8 - System’s response with force and position control (up – force, down – displacement)

5. CONCLUSIONS

The testing showed that a system, complex enough to tune, can be rapidly and optimal adjusted (regulated), based yet on the results of a long research and testing activity developed by a group of field specialists. With this module was realized a very good and shock less transfer of position control into force control. It is taken into account that in the near future, the modules to be achieved by modern technologies - for example SMT (surface mounting technology) - that can lead to small dimensions low price, high performances. At the same time this is a start in order to achieve local intelligence systems, of position regulation, by using micro-controllers having implemented a regulation-control software based, among others, on the presently paper result.

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