EXPERIMENTAL RESEARCH FOCUSED THE MECHATRONIC POSITIONING SYSTEMS FOR REGULATING THE GEOMETRICAL VOLUME OF THE PUMPS WITH RADIAL PISTONS

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Pompele volumice rotative cu pistoane radiale folosite în domeniul acționărilor hidraulice au fost puțin cercetate în România, iar atunci când au fost abordate, problematica lor a fost tratată doar la nivel teoretic.

Lucrarea prezintă rezultatele cercetărilor experimentale efectuate de autor asupra unui servomecanism de poziționare destinat reglării volumului geometric al pompelor cu pistoane radiale.

Pentru efectuarea cercetărilor, autorul a conceput şi realizat atât modelul experimental al sistemului mecatronic de poziționare cât şi standul de probă.

The rotary volumetric pumps with radial pistons used in hydraulics have been very little studied in Romania and when the issue was approached it was only theoretically.

The paper presents the results of the experimental studies performed by the author on a positioning servo mechanism used for regulating the geometrical volume of the pumps with radial pistons.

For developing the research, the author has designed both and realized the experimental model of the mechatronic positioning system and the test stand.

Keywords: mechatronic system, positioning servo mechanism, volumetric pump, flow regulating

1. Introduction

The hydraulic power systems use as a working medium a fluid under pressure. The generators of hydrostatic energy, named currently volumetric pumps have the role of supplying the pressure power for the operational fluid. This is then used by the hydraulic motors for producing mechanical work for power mechanisms. A volumetric pump is energetically efficient if it supplies only the flow required by the system [1], which means that it must be capable to change continuously its geometric volume.

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At modern pumps the regulation of the geometric volume is achieved by means of mechatronic positioning systems.

In Fig. 1 a functional scheme of such a mechatronic system which operates with hydraulic power is shown [2].

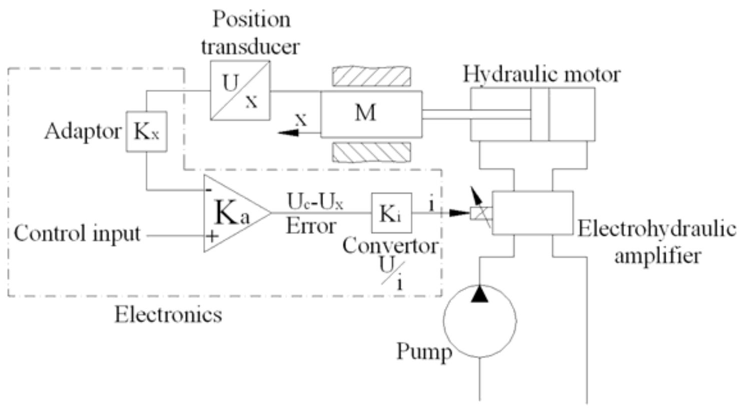


Fig. 1. Scheme of position control electro hydraulic servo system
M – load mass, x – piston displacement, k_x – calibration coefficient position information, k_i – conversion coefficient voltage / current, k_a - error amplification factor

The system includes the following components:

- linear hydraulic motor which may be a double or single-acting hydraulic cylinder;
- electrohydraulic amplifier which may be servo valve or proportional distributor;
- displacement or position transducer;
- electronics.

2. The experimental model of the servo mechanism

In Fig. 2 is presented the scheme of the functional model tested experimentally and in Fig. 3 is given a photo of the realized physical model.

The structure of the servo mechanism is the classic one used by all the manufacturers of pumps with blades or radial pistons. The experimental model was realized using the electrohydraulic amplifier D 930 MOOG with integrated electronics [3]. The mechanical components of the positioning system and the linear hydraulic motor were designed and realized by the author at the institute (INOE 2000 IHP).

The influence of the various parameters (the diameters of the pistons, clearances of the hydraulic motor, the constant of springs, the supply pressure etc.) on the dynamics of the system was determined by numerical simulation. The software package AMESim of the French company „Imagine" was used [4].

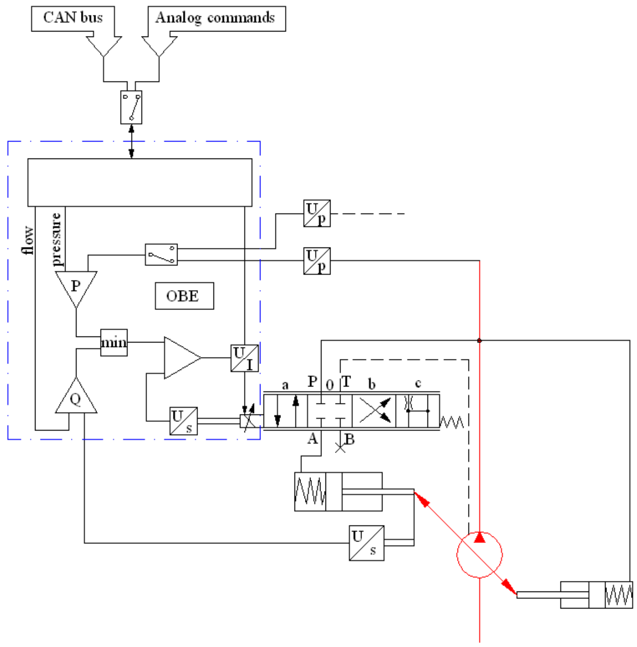


Fig. 2. Diagram of functional model of positioning servomechanism
CAN (Controller Area Network), OBE (Integrated (On Board) Control Electronics)

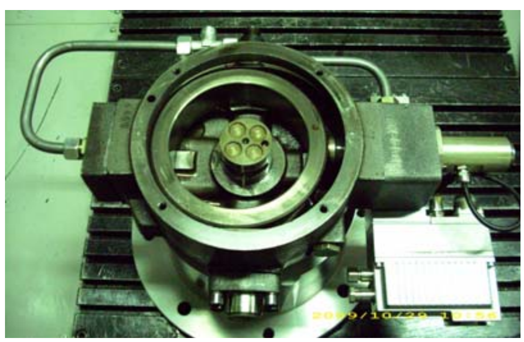


Fig. 3. Functional model of positioning servomechanism

3. Test stand

The scheme of the stand used for experimental research is shown in fig. 4 The source of hydraulic energy consists of: a pump (P) powered [5] by an electric motor, a safety valve used for regulating pressure (SS) and a high fineness filter (F).

The integrated control electronics compares and processes signals received from the stroke transducer and the generator of functions (GFA) by means of which the positioning tracks are programmed based on the electric power which supplies the proportional distributor. All the measured electric signals, processed in the system are collected by a data acquisition system (SAD) and send to a computer where they are memorized in a standard format system, as database.

The voltage source (STT) sends signals to the electronic amplifier to simulate the set pressure reaching by the pump on which is mounted the positioning device already tested.

Technical parameters:

a) The positioning mechanism:

- positioned mass $M = 3{,}05$ kg
- large area piston $D = 32$ mm $A = 804{,}25$ mm^2
- small area piston $d = 22$ mm $a = 380{,}13$ mm^2
- eccentricity range $e = -5 \ldots +5$ mm

b) Control circuit:

- flow 6 l/min
- pressure 20 bar

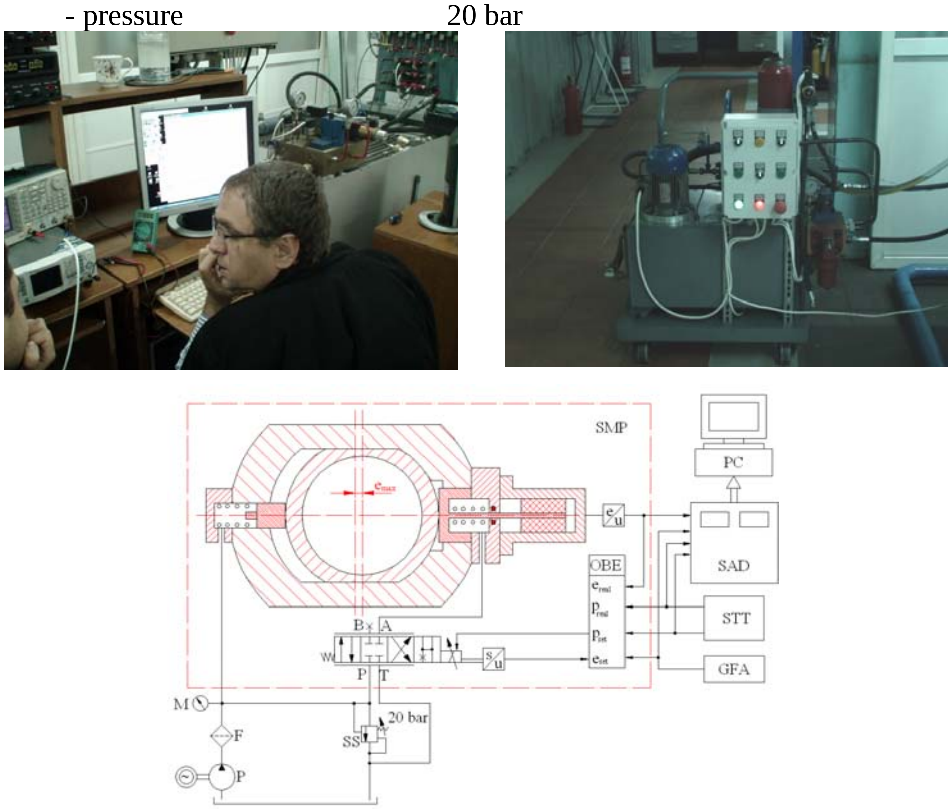


Fig. 4. Diagram of the experimental research stand

SMP-mechatronic positioning system
SAD-data acquisition system
STT-stabilized voltage source
GFA-generator of aleatory functions
SS-safety valve
OBE-on board electronics

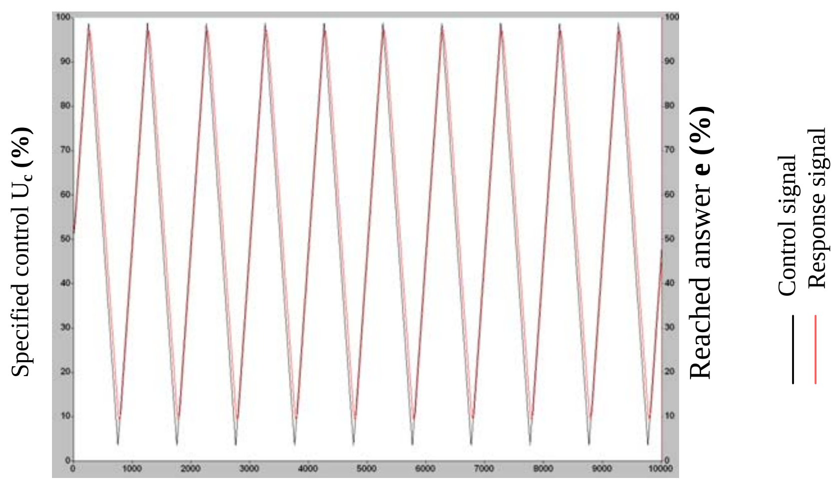
F-filter
M-manometer
P-pump
e_{set}-set eccentricity
e_{real}-reached eccentricity
p_{set}-set pressure
P_{real}-reached pressure

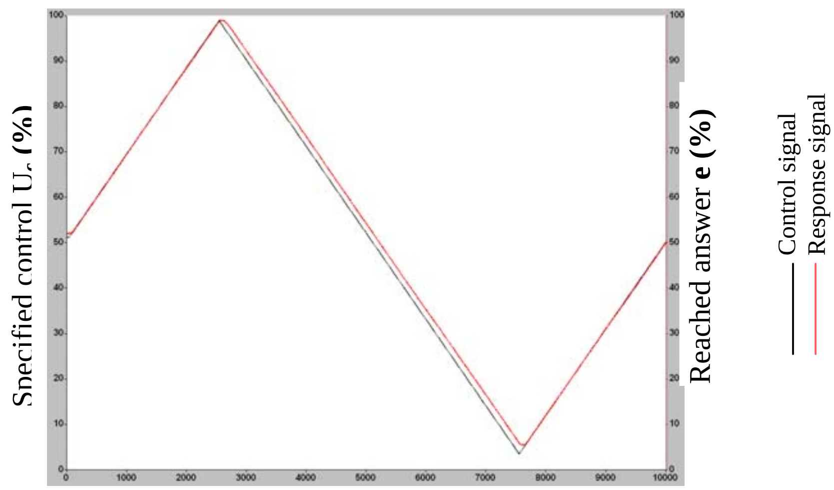
4. Quasi – static experimental tests

For finding the behavior in static operational state of the positioning system control signals U_c were applied in the range between 0...10 V increasing and decreasing, of various shape and a range of frequencies of 1; 0.7; 0.35 and 0.1 Hz.

4.1.The response to the signal in the increase/decrease range

The tests were performed for control signals with max. amplitude and frequencies of 1; 0.7; 0.35 and 0.1 Hz .The results for the values of 1 Hz and 0.1 Hz are shown in the diagrams from the Fig. 5. Table 1 is shown the delay between specified and reached for ups and downs at a certain moment chosen at random T.





Time (ms)   Frequency 0.1 Hz

Fig. 5. Response of the positioning system to increasing/decreasing ramp-type signal

Table 1

Difference between preset and obtained value in case of an increasing /decreasing ramp-type signal

Time (ms)	Sense	Specified (%)	Reached (%)	Difference (%)
1000	upward	49.7370986415577	46.9134193908708	2.82%
1500	downward	52.7455696644891	59.8680415163945	7.12%

Frequency 1 Hz

Time (ms)	Sense	Specified (%)	Reached (%)	Difference (%)
2000	upward	88.5573180303273	88.6701482377609	0.11%
4000	downward	71.1325538819081	73.2525682022705	2.12%

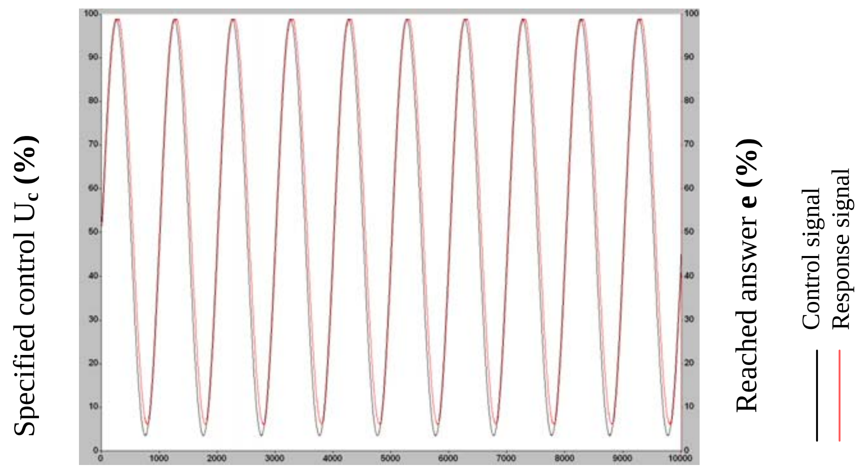
Frequency 0.1 Hz

It is found that the delay between specified and reached increases in the same time with the frequency and is higher downwards than upwards.

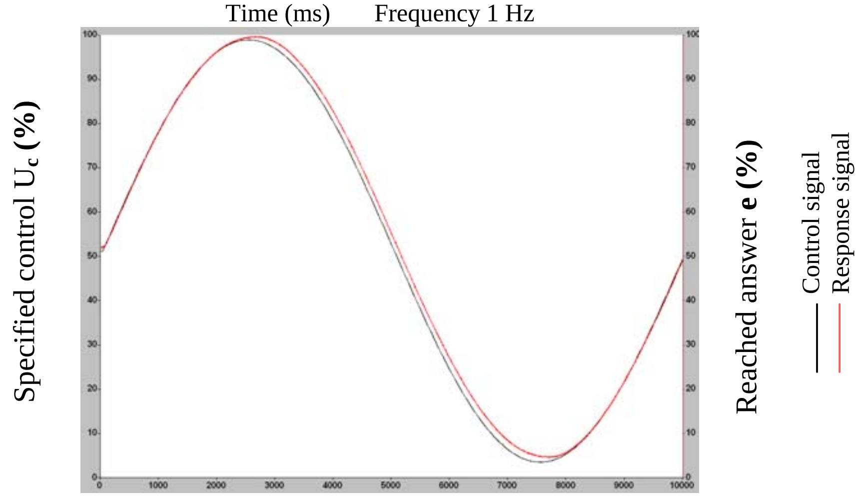
4.2. The response to the low frequency sinusoidal signal

The tests were made for prompts with max amplitude and frequencies of 1; 0.7; 0.35 and 0.1 Hz.

The results obtained for the values of 1 Hz and 0.1 Hz are shown in the diagram from 6. In Table 2 is presented the delay between specified and reached for ups and downs at a certain moment chosen at random T.



Time (ms) Frequency 1 Hz



Time (ms) Frequency 0.1 Hz

Fig. 6. Reponse of the positioning system to low frequency sinusoidal signal

Table 2

Difference between preset and obtained value in case of a low-frequency sinusoidal signal

Time (ms)	Sense	Specified (%)	Reached (%)	Difference (%)
1000	upward	48.3726078410567	44.0732337721616	4.3%
1500	downward	54.1412259937081	64.1677742408697	10.03%

Frequency 1 Hz

Time (ms)	Sense	Specified (%)	Reached(%)	Difference (%)
2000	upward	96.0911535185473	96.2432511385023	0.15%
4000	downward	80.428514170209	82.5280050154415	2.1%

Frequency 0.1 Hz

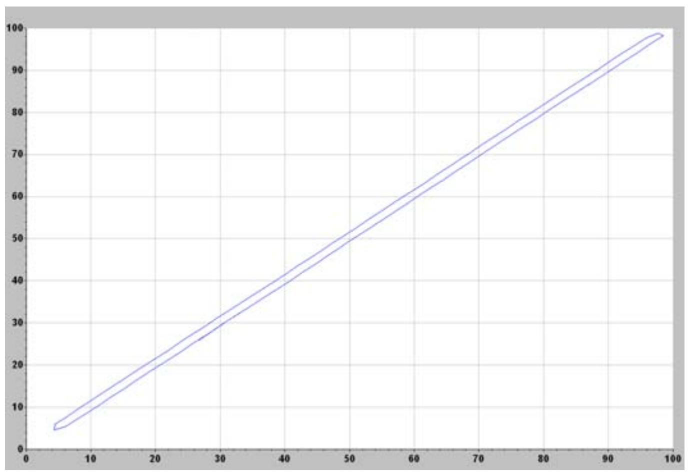
It is found that the delay between specified and reached increases in the same time with the frequency and it is higher downwards than upwards.

4.3.The static characteristic of the system

The relation between the specified prompt and the response of the system was determined for a control signal of an increasing decreasing range of frequencies between 0.1 and 0.0005 Hz (Fig. 9).

It is found that the hysteresis decreases with the frequency.

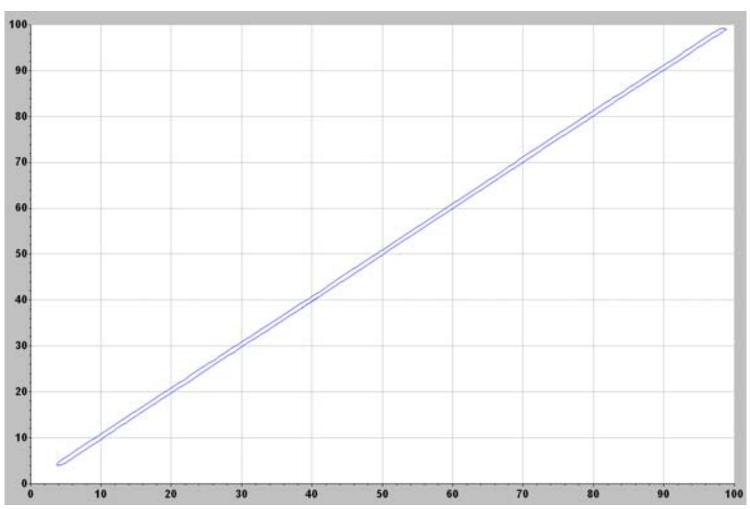
Reached response **e (%)**



Frequency 0.1 Hz

Specified control U_c **(%)**

Reached response **e (%)**



Frequency 0.03 Hz

Specified control U_c **(%)**

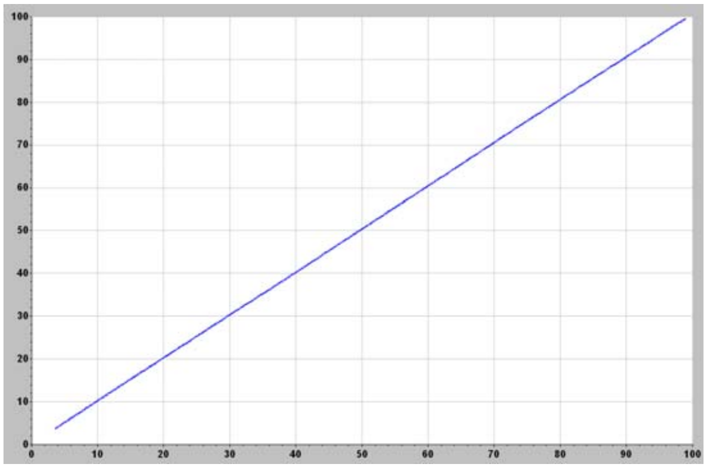


Fig. 7. Static characteristic of positioning system

5. Experimental tests in dynamic operational status

5.1.The response to the signal step of the mechatronic positioning system

The tests were performed for a broad range of amplitudes for finding the relation between the response times and this parameter. At each amplitude, the system was stimulated with a range of signal steps of various frequencies for finding if the response remains the same when the frequency changes.

Response of positioning system to step-type signal with amplitude of 100% and frequency of 0.1 Hz is shown as a diagram in Fig. 8, and its values - in Table 3.

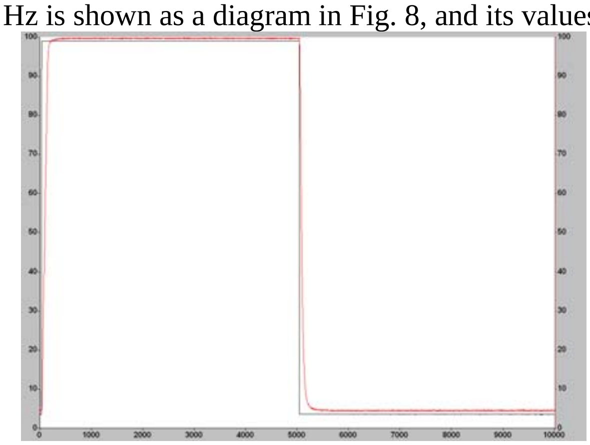


Fig. 8. Response of positioning system to step-type signal

Table 3

Response values of positioning system to step-type signal

Time (ms)	....	96	98	100	....	158	160	162
Specified (%)	....	98.960714	98.955287	98.967967	....	98.957436	98.973233	98.951848
Reached (%)	....	49.754011	51.186666	52.812876	....	94.402089	95.000497	95.281231

From this test it results:

- The time of delay: 97 ms
- The stabilization time: 160 ms
- overregulation: zero

In Fig. 9 is shown the response for signal steps with frequencies of 0.5 and 1 Hz.

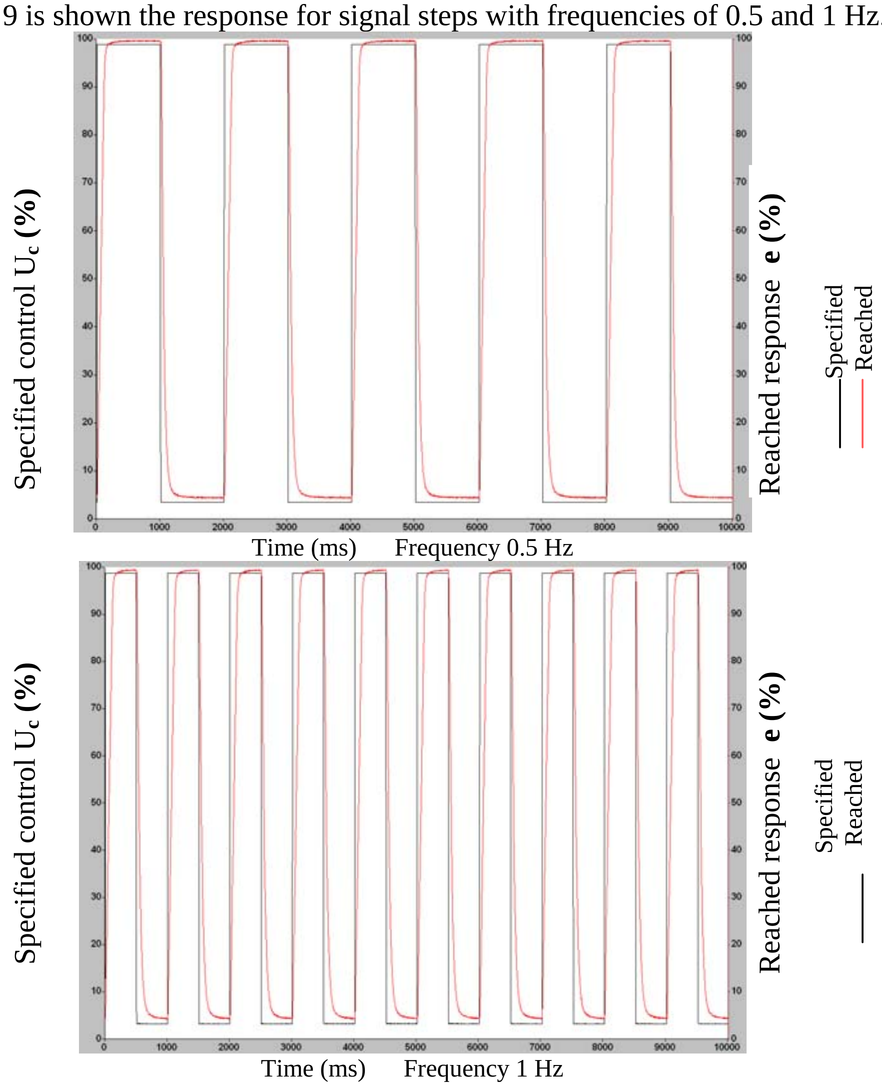
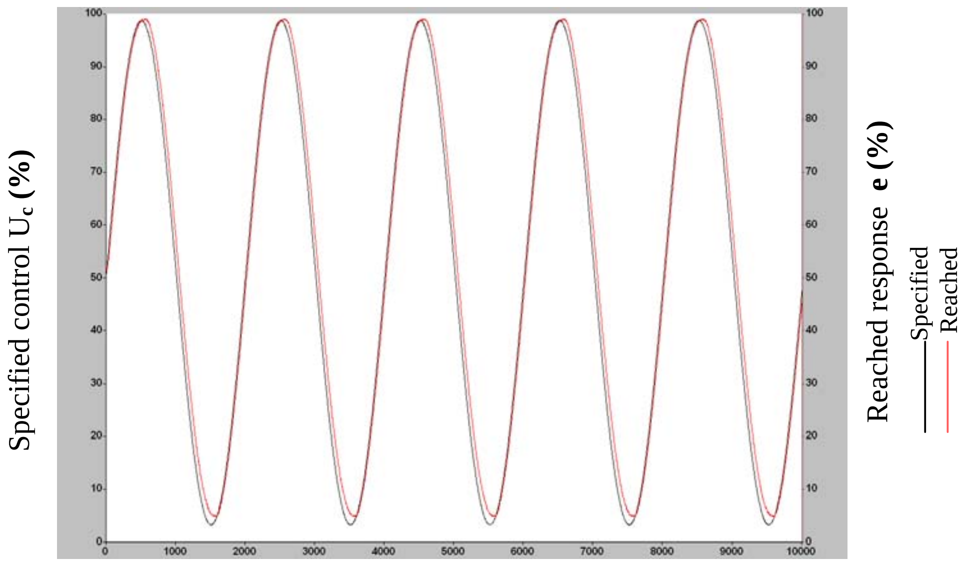


Fig. 9. Response of positioning system to a range of step-type signals

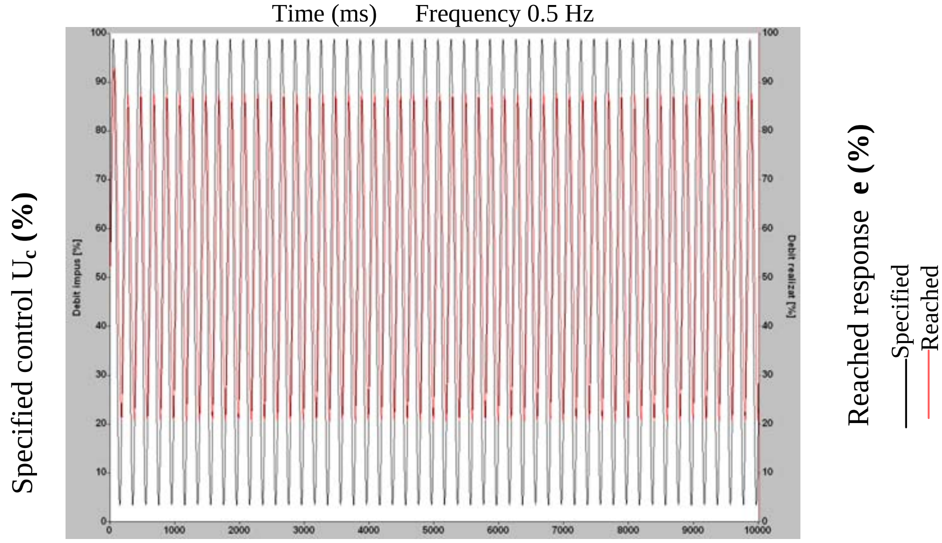
5.2.The response to the sinusoidal signal

For finding the response of the system in this case were applied control signals of various ranges of frequency and amplitude.

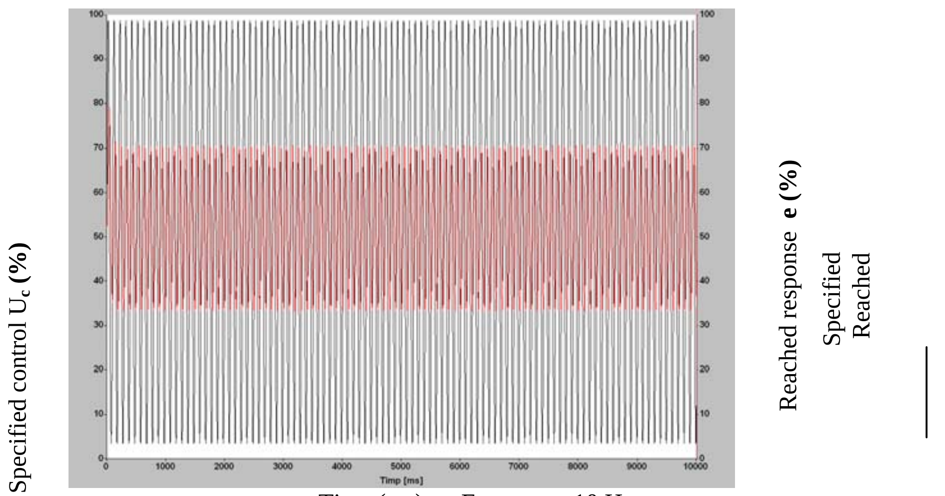
In Fig. 10 is shown the response to the sinusoidal signal with an amplitude of 100% and frequency of 0.5; 6; 10 Hz.



Time (ms) Frequency 0.5 Hz



Time (ms) Frequency 5 Hz



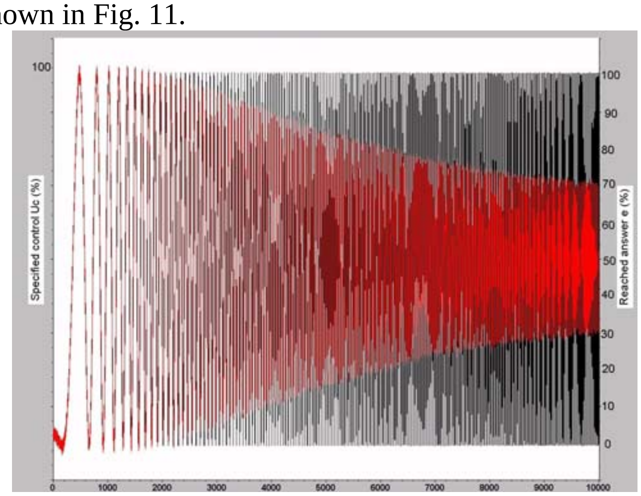
Time (ms) Frequency 10 Hz

Fig. 10. Response of positioning system to sinusoidal signals

It is found that the amplitude of the response decreases when the frequency of the signal increases and the delay between prompt and response increases when the frequency of the control signal increases.

5.3.The frequency response

For finding the response in frequency of the positioning system it was applied a sinusoidal signal with max. amplitude and increasing frequency starting from 0.1 Hz up to 20 Hz for 10 sec. The evolution in time of the position of the sliding ring of the mechanism represents the response in frequency of the positioning system and it is shown in Fig. 11.



Time (ms)

Fig. 11. Response in frequency of the positioning system

6. BODE diagram

The dynamic performances of the positioning system are given by:
- the characteristic of attenuation of amplitude of the response depending on the frequency of the control signal;
- the delay/phase difference of the response depends on the frequency of the control signal.

These two characteristics were found based on the tests from point 7. They form the Bode diagram shown in Fig. 12. The measurements for amplitude and phase difference were performed for 8 frequencies 0.1; 0.2; 0.5; 1; 2; 5; 10; 20 Hz.

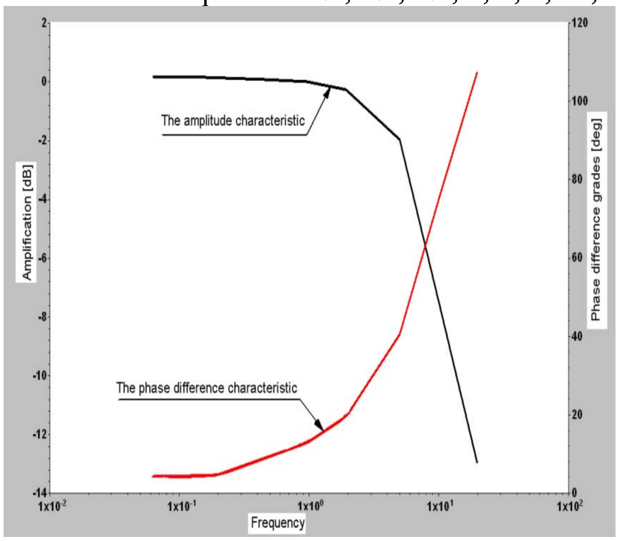


Fig. 12. BODE Diagram

From the Bode diagram it results that the attenuation of amplitude of 3 dB (which means a decrease with 30% of the response amplitude) takes place at a frequency of 5.5 Hz of the control signal. At this frequency the phase difference is 44^{o}.

7. Conclusions

The experimental model designed and realized by the author at the institute INOE 2000 IHP (The R & D Institute of Hydraulics and Pneumatics) was in accordance with the functional requirements of a positioning mechanism used for regulating the geometrical volume of the pumps with radial pistons.

The test stand designed and realized by the author was used for performing on it the experimental studies in static and dynamic conditions of the system.

The dynamic performances of the model:

- the dead time of the system $T_m = 0$;
- the time of delay $T_i = 95$ ms;
- the time constant of the system $K_t = 115$ ms;
- stabilization time $T_s = 160$ ms;
- overregulation:
  - attenuation at f = 5.5 Hz .... A = - 3 dB (70%)
  - phase difference at f = 18 Hz .... $\varphi = 90^0$.

The experimental model may be considered satisfactory as a positioning mechanism used for regulating the geometrical volume of a pump used in industrial operations.

R E F E R E N C E S

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