Research on the properties of a hydrostatic transmission with different controllers

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Výskum vlastností hydrostatickej transmisie regulovanej rôznymi regulátormi

In this paper the possibility of a uses control system with a signal processor DSP to control hydrostatic transmission was described. A hydrostatic transmission with pump variable efficiency and engine radial with constant working absorptivity was chosen. The control of the efficiency of pump was realized by using the electrohydraulic control system. This hydraulic system consists of a servo-cylinder and electrohydraulic servovalve. Such an object is one of the most important parts of working machines. Because the object is nonlinear and not time invariant its control is very difficult. In the last few years using the signal processor DSP for control has become very popular. In this paper the use of cascade controllers in DSP was described. The cascade controllers realize control of the position of the servo-cylinder and the control of the rotational speed of the hydraulic engine To chose the controller's parameters the simulation model adopted in Matlab/Simulink was used. The object used parameters from simulation tests. Many different tests were conducted on a laboratory hydrostatic transmission.

Key words: *hydrostatic transmission, controller, module DSP.*

Introduction

The hydrostatic transmission, a basic hydraulic system, has been well known and widely practical for a long time. Nevertheless, new solutions and possibilities were sought to improve the techno-exploitational properties of the range control work parameters of the hydraulic engine transmission. The stabilization of either rotational speed or torque on the motor shaft is a frequent hydraulic task. In some uses, the hydrostatic transmission is only part of the machine. For example: the drive. In this case the function of the transmission is to drive the coal longwall shearer loader. This makes it possible to suitably control the torque acting on the working heads, because they are a function of the loaders speed. (Pluta,1996).

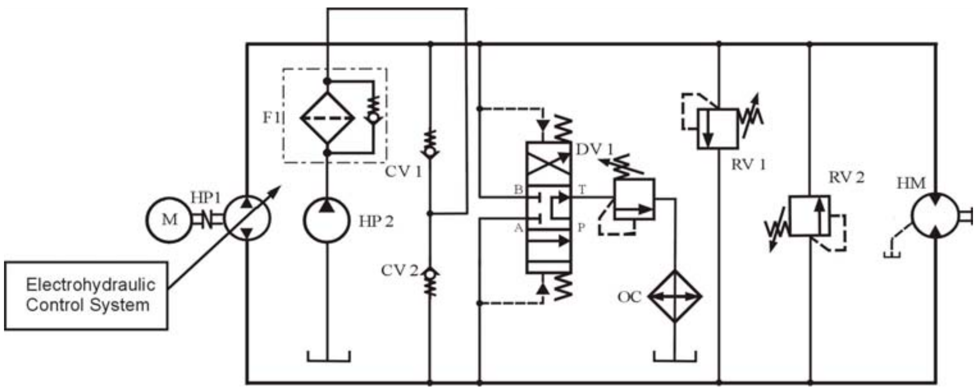

Fig.1. Hydrostatic transmission scheme: HP1. HP2 – hydraulic pumps, CV1, CV2 – check valves, RV1, RV2 – pressure relief valves, DV – directional control valve, HM - hydraulic motor, OC – oil cooler, F1-filter, M-electrical motor.

The hydrostatic transmission is difficult to control because its properties as a hydrostatic transmission make it sensitive to disturbances. The transmission requires the possibility of a wide range of speed, but that speed must be set precisely. Since all possibilities of hydraulic transmission construction have been exhausted, selection of a suitable control system will be of the most consequence.

Control by using the hydraulic drive is not a time invariant process, and consequently, it is affected by inconsistencies in the parameters of the hydraulic liquid. (for example: kinematic viscosity, compressibility, thickness). The efficiency of pump control is realized by using the electrohydraulic servo - system, which has a non-linear characteristic. The rotational speed of hydraulic engine of transmission the two-state controllers PID controllers are use as the popular circuit or modified under resistance on disturbance.

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In this article use of the DSP module (SPAC20 from Mitsubishi Electric Corporation), which is used to control the rotational speed of the hydraulic motor shaft through use of a PID controller, will be described.

Selection of the controller's parameters

A hydrostatic transmission is a very complex object. Having in mind the huge knowledge accumulated on this subject and the constructed model's target application, a certain strategy was adopted. This strategy aimed at writing the model in the simplest possible form, while at the same time, preserving the significant features of the real object.

Accordingly, it has been assumed that: a hydrostatic transmission is a system with lumped constants. The static and dynamic features of the transmission do not depend upon the direction of the hydraulic engine's rotation. Thus, a mathematical model was developed for only one rotation direction. It is assumed that the transmission is in a thermally balanced state, and that the module of volume elasticity is constant. The angular velocity of the main pump shaft is constant and pressure drop in the hydraulic cables is negligible. Leaks in the pump and in the engine can be summed, and neither the pump's efficiency, nor the absorptivity of the hydraulic engine depends upon their shaft's rotation angle. The safety valve is closed at all times.

The calculation scheme presented in Figure 2 was adopted for developing mathematical model of a hydrostatic transmission.

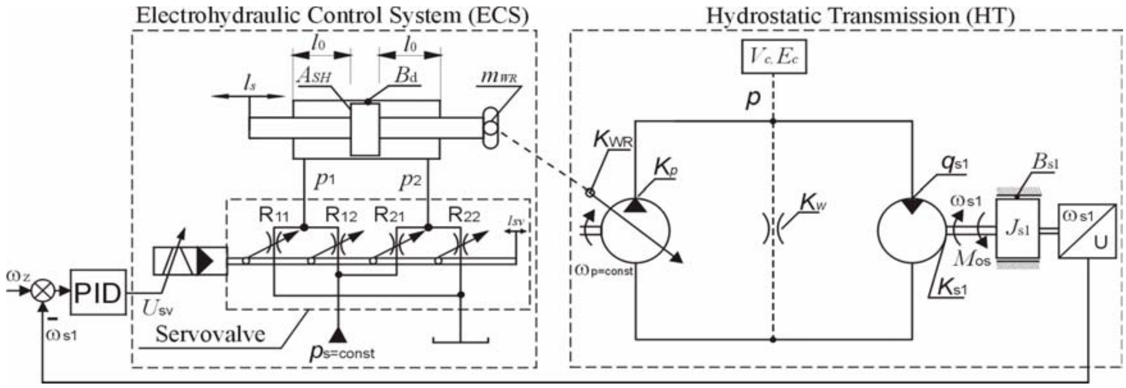

Fig.2. Hydrostatic transmission calculation scheme with electrohydraulic control system.

The mathematical model of the transmission presented in Figure 2 is described by the following equations:

$$\dot{p} = \frac{E_c}{V_c}\left(K_p l_s - K_w p - q_{s1}\omega_{s1}\right) \tag{1}$$

$$\dot{\omega}_{s1} = \frac{1}{J_{s1}}\left(K_{s1} p - B_{s1}\omega_{s1} - M_{os}\right) \tag{2}$$

$$\ddot{l}_s = \frac{1}{m_{WR}}\left(A_{SH}(p_1 - p_2) - B_d \dot{l}_s - K_{WR} p\right) \tag{3}$$

$$\dot{p}_1 = \frac{E_c}{A_{SH}(l_0 + l_s)}\left(RSV(l_{sv}, p_s - p_1) - A_{SH}\dot{l}_s - RSV(-l_{sv}, p_1)\right) \tag{4}$$

$$\dot{p}_2 = \frac{E_c}{A_{SH}(l_0 - l_s)}\left(RSV(-l_{sv}, p_s - p_2) + A_{SH}\dot{l}_s - RSV(l_{sv}, p_2)\right) \tag{5}$$

$$RSV(l_{sv}, p_{sv}) = K_{sv} SAT(l_{sv}) \operatorname{sgn}(p_{sv})\sqrt{|p_{sv}|} \tag{6}$$

$$SAT(l_{sv}) = \max\{\min\{l_{sv\max}, l_{sv0} + l_{sv}\}, 0\} \tag{7}$$

The hydraulic resistances for the servovalves are described by the following formulae:

$$\begin{aligned} R_{11}(l_{sv}) &= K_{sv}\max\{\min\{l_{sv\max}, l_{sv0} + l_{sv}\}, 0\} \\ R_{21}(l_{sv}) &= K_{sv}\max\{\min\{l_{sv\max}, l_{sv0} - l_{sv}\}, 0\} \\ R_{22}(l_{sv}) &= K_{sv}\max\{\min\{l_{sv\max}, l_{sv0} + l_{sv}\}, 0\} \\ R_{12}(l_{sv}) &= K_{sv}\max\{\min\{l_{sv\max}, l_{sv0} - l_{sv}\}, 0\} \end{aligned} \tag{8}$$

Notation: K_{sv} – coefficient of the reinforcement servovalve ($2.2048*10^{-5}$), m_{WR} – mass reduced on the cylinder piston rod (0.5 kg), l_{sv} – servovalve's spool position, l_{sv0} – overlap position ($20*10^{-6}$ m), p_{sv} – pressure drop in servovalve's spool, E_c – fluid bulk modulus ($1.385*10^{9}$ N/m^2), V_c – fluid volume ($2*10^{-3}$ m^3), B_{s1}, B_d – resistance coefficient of viscous friction (50 Ns/m), K_w – volumetric leakage coefficient of the hydrostatic system, K_p – pump delivery coefficient, K_{s1} –motor's torque coefficient, q_{s1} – hydrostatic motor displacement, J_{s1} – total moment of inertia reduced on the shaft of motor (0.02 kgm^2).

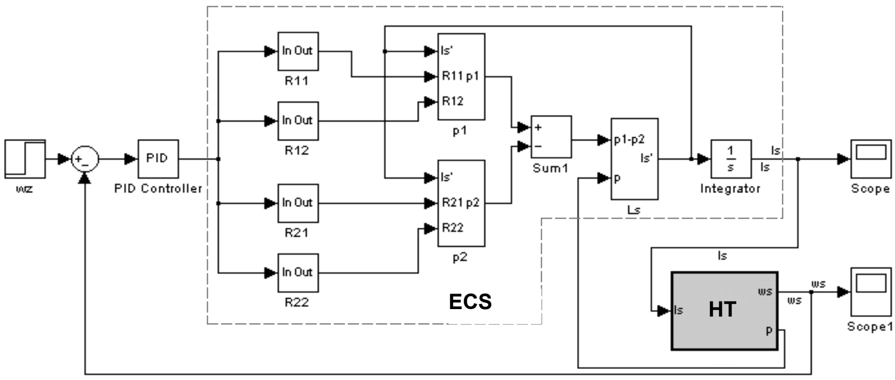

Fig.3. Matlab/Simulink scheme of the hydrostatic transmission model.

The above model was adopted in the Matlab/Simulink program (Fig. 3) with the aim of running simulation tests. The results of these tests enabled the controller's parameters to be determined by use of the NCD program. The PI controller was chosen because it cancelled offset. In tests, the controller's parameter was chosen to be: k = 55%, T_i = 0.9 s. In simulation tests, the parameters of the laboratory object were used.

Laboratory tests

Verification of the model and controller's parameters was done through laboratory tests. The hydrostatic transmission used for the experimental investigation was a hydraulic system that consisted of a power transmission system, a load system and a control system (Fig. 4 and 5).

The laboratory hydrostatical transmission used a closed loop of hydraulic liquid. This transmission is built using a multipiston axial pump with variable efficiency HP, an engine radial multipistons with constant working

Fig.4. Hydraulic schematic of the stand.

absorptivity HM, and a suitably well chosen set of valves. Control of the efficiency of the pump, and the rotational speed of the motor shaft was realized by using an electrohydraulic control system. This hydraulic system consists of a servo-cylinder HC joint with piston rod deflection of the pump rotor, and an electrohydraulic servovalve SV built-up directly on cylinder (Fig. 6). The displacement of the piston rod of the cylinder is measured using a position sensor that detects the angular deflection of the pump rotor. The position of the pump rotor depends on the efficiency and direction of the droving of the working liquid. Thus it also depends on the rotational speed, as well as the sense of rotation. The hydraulic engine load is realized by the pump that is connected with this engine mechanically. The pressure proportional valve, along with load pump, allows the possibility of electrohydraulic control of the hydraulic engine load. (Pluta, 1995).

Fig.5. Laboratory stand of the hydrostatic transmission.

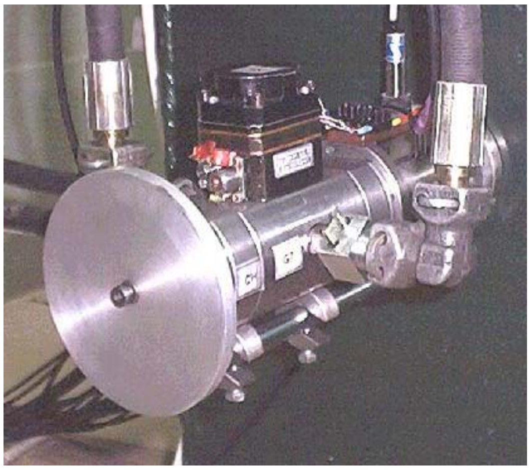

Fig.6. Electrohydraulic servo control of the pump.

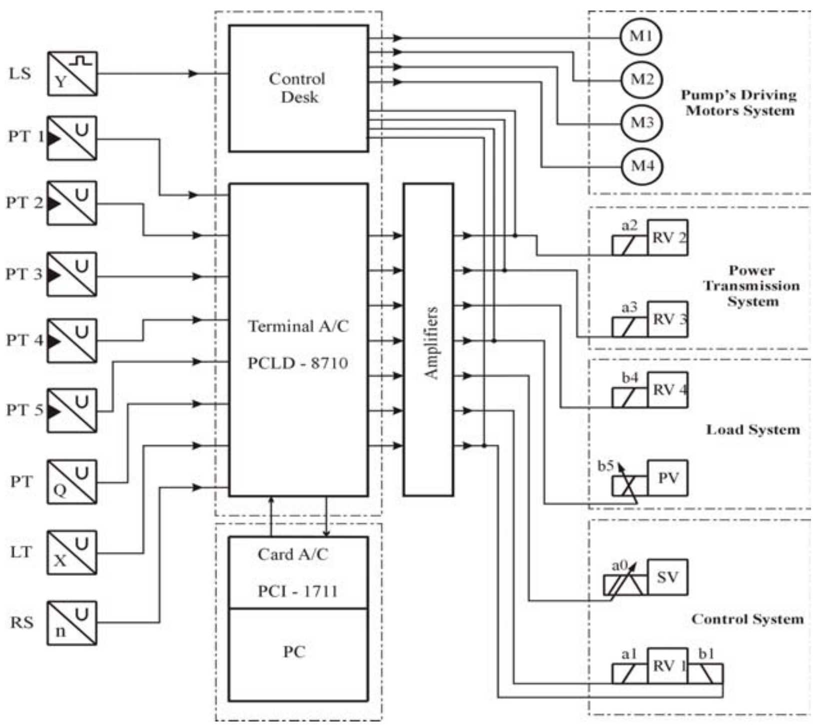

Fig.7. Control and measurement system.

The most important element is pump HP5, which by use of circulation valve CS3 presses hydraulic fluid to three other elements: pressure relief valve PV4, proportional pressure relief valve PV and directional control valve DV4. The proportional pressure relief valve PV enables control of pump pressure continuously, and the pressure relief valve RV4 limits the maximum value of the pressure. For the fast loading and unloading of hydraulic pump HP5, the directional control valve DV4 is used. By using this element we can realise different kinds of load, such as: step function, sinusoidal, stochastic etc.

The measuring and control system that was built for research is shown in Figure 7. The control part consists of: torque motor a0 servovalve's SV, and propor–tional electromagnet b5 propor–tional pressure valve's PV. The measurement part is composed of sensors, transducers, amplifiers, control board, measurement card PCI-711 with terminal strip PCLD 8710 Advantech and elements of thecomputer system. The software for the control system of the hydrostatic transmission was built using Matlab/Simulink. The measurement signals are converted and given in the analog input of the measurement card. A PC computer is used for registration of the data.

The laboratory investigation included choosing some static and dynamics characteristic for the control system.

The research started by preparing the object's static characteristic. For example, Figure 8 shows a dependence between pressure drop on the servovalve and the control signal (voltage) for delivery pressure 2.5 MPa. This characteristic was of a hysteresis form. It means that the object is nonlinear. One visible characteristic is a fold that comes from the servovalve's overlap.

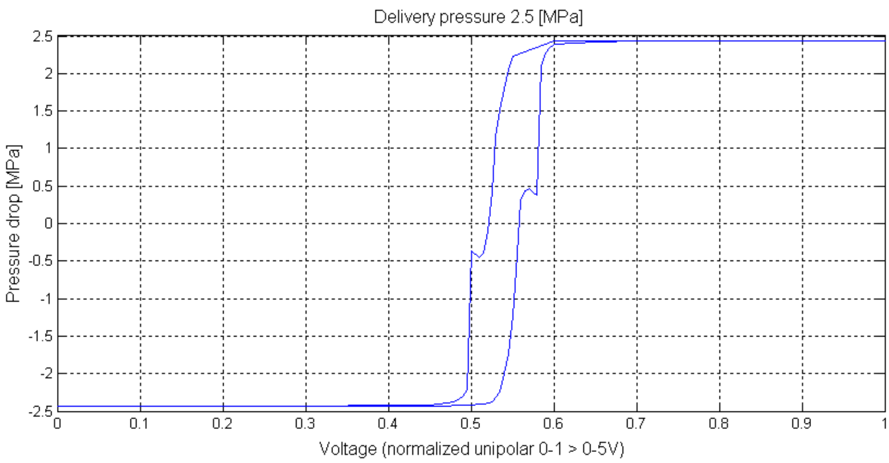

Fig.8. Dependence between pressure drop and voltage.

The next part describes the investigation of control system dynamics. Controller type PI was chosen to stabilise the physical quantity. The controller was built by using a signal coprocessor DSP SPAC 20 with a program that was written in IRD Bloc firmware. First, the controller for servo-cylinder position was investigated (Fig. 9). The position of servo-cylinder ranges from –5 to +5 V, which corresponds to from –27.5 to +27.5 mm. For control, the PI controller was chosen and given the parameters: k = 55%, T_i = 0.9 s, which come from the simulation tests. Tests were done for three values: 10, 20 and 25 mm of HL-46 oil at 30°C temperature (Fig. 10). The value of the step impulse was given after 1 s.

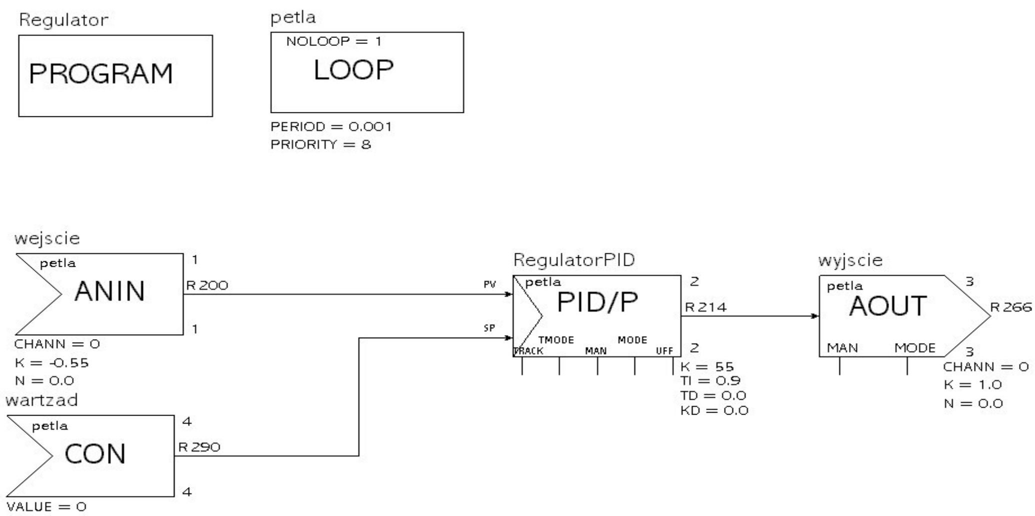

Fig.9. Program which realises the PI controller for the servo-cylinder position,

Tab.1. Performance indices obtained from tests.

Performance indices	Set value 10 [mm]	Set value20 [mm]	Set value 25 [mm]
Settlement time 2% [s]	2.205	2.28	2.245
Over -regulation [%]	12.66	6.347	4.468
Offset [mm]	0	-0.028	-0.018
Integral of absolute error: IAE Integral of squared error: ISE	0.4585 0.2458	0.6955 0.4309	0.8328 0.534

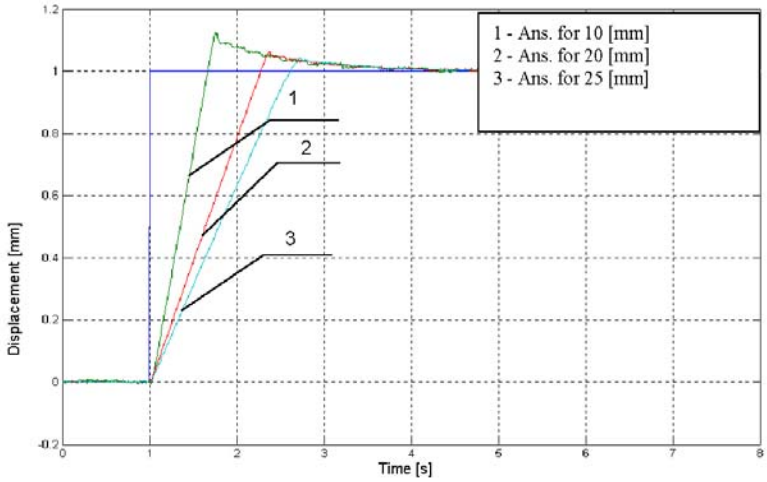

Fig.10. Comparison of step responses for three different values of the piston position.

Because the results were not good enough (difficulty with stability of the zero value), in the next stage cascade controllers were built. An internal loop (I step) realises control of the position servo-cylinder and the second external loop (II step) realises control of the rotational speed of the hydraulic engine. In both loops, the PI controller was used. It was built using the signal coprocessor DSP SPAC 20 and its program was written in IRD Bloc firmware, as before.

To control the rotational speed of the hydraulic engine, a PI controller was chosen with the parameters: $k_R = 2.5\%$, $T_i = 1.5$ s. The tests were done for three set values of rotational speed: 300, 600 and 900 rpm in HL-46 oil at 30°C temperature. The value of the step impulse was given after 1 s.

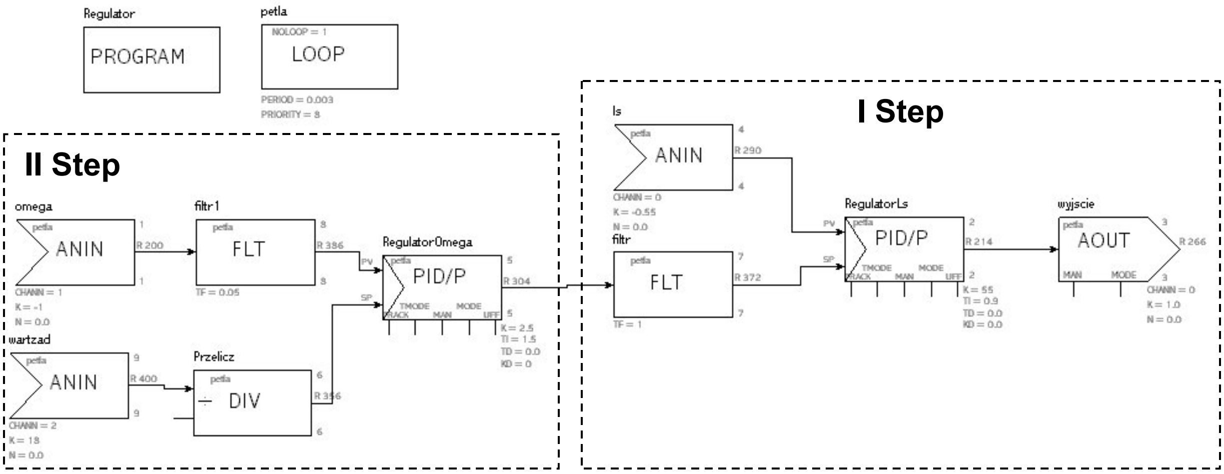

Fig.11. Program which realises the cascade controllers.

Tab.2. Performance indices obtained from tests.

Performance indices	**Set value 300 [rpm]**		
Settlement time 5% [s]	-	2.38	1.9050
Over -regulation [%]	42.82	30.127	7.0638
Integral of absolute error: IAE	0.8866	0.851	0.8889
Integral of squared error: ISE	0.2813	0.4328	0.5498

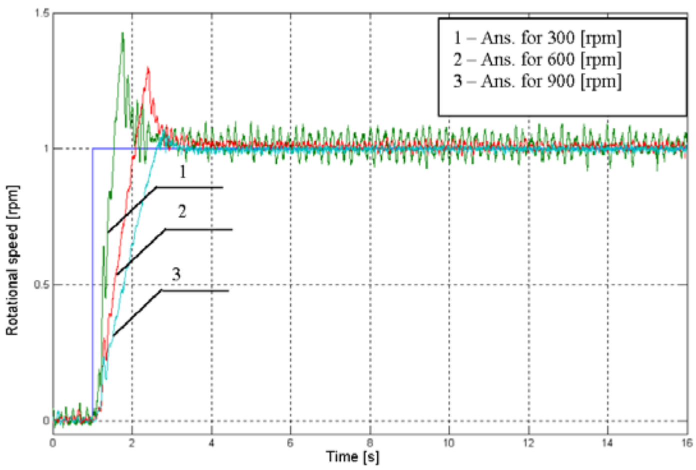

Fig.12. Comparison of step responses for three different values of rotational speed.

Figures 13 ÷ 18 show the influence of a disturbance as a step function to the correct working of the object. In the first series of tests the rotational speed was changed: 300, 500, 700 and 900 rpm, but the disturbance was constant. The set value was given after 1 s. The disturbance was a rectangle signal. In the second series the rotational speed was constant but the disturbance was changed.

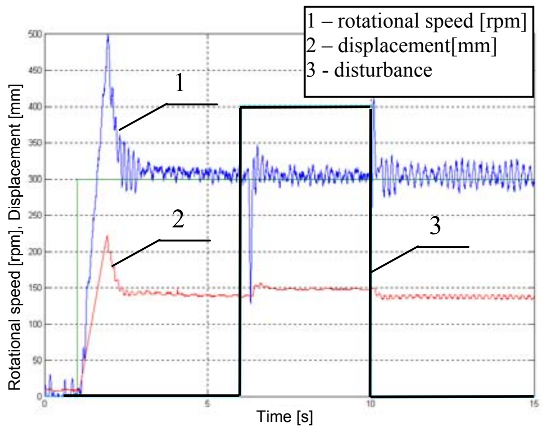

Fig.13. Rotational speed (300 rpm) and servo-cylinder displacement with disturbance.

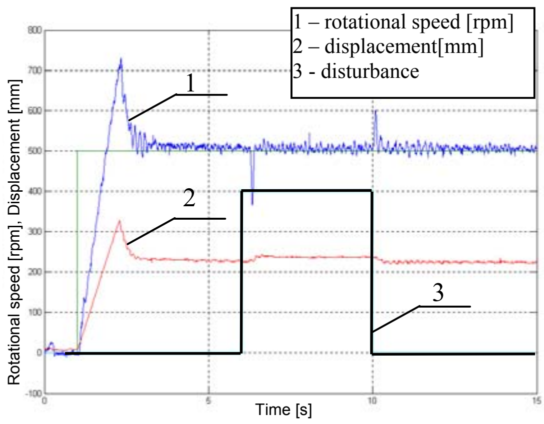

Fig.14. Rotational speed (500 rpm) and servo-cylinder displacement with disturbance.

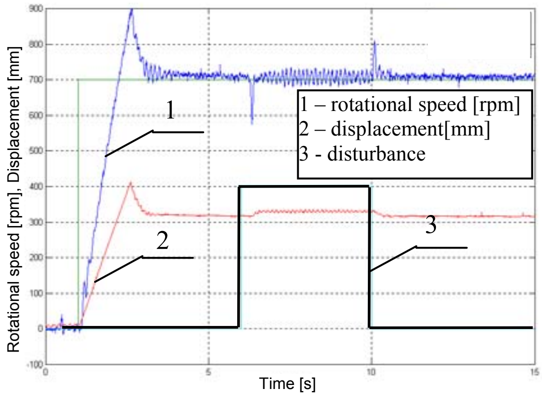

Fig.15. Rotational speed (700 rpm) and servo-cylinder displacement with disturbance.

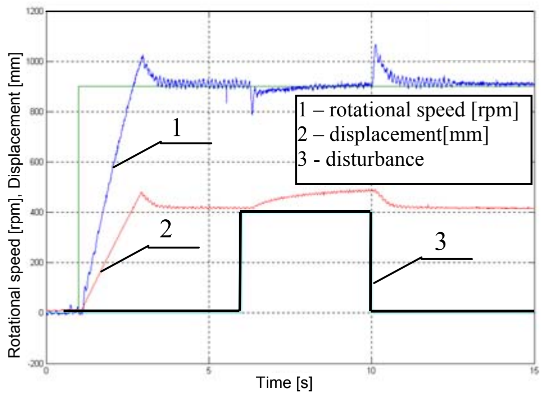

Fig.16. Rotational speed (900 rpm) and servo-cylinder displacement with disturbance.

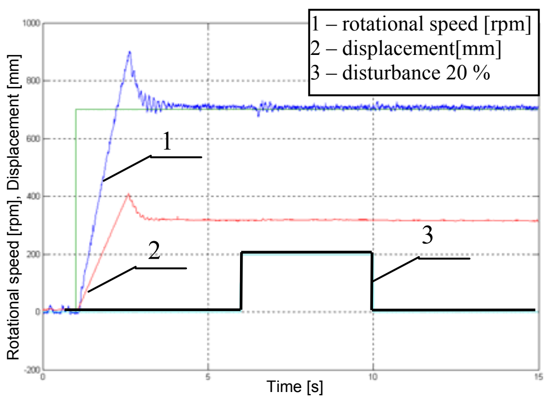

Fig.17. Rotational speed (700 rpm) and servo-cylinder displacement with disturbance.

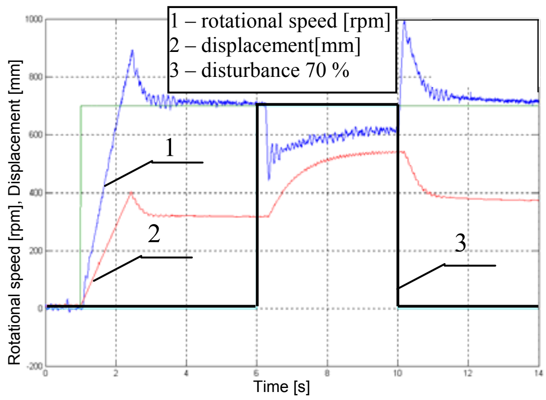

Fig.18. Rotational speed (700 rpm) and servo-cylinder displacement with disturbance.

Next, the influence of oil temperature (viscosity) was tested. Figures 19 ÷ 21 show the result of tests done at temperature 30°C, and then at 40°C, for hydraulic engine rotational speeds of 300, 600 and 900 rpm. The results we received show that a small change in oil temperature results in a visible change in setting time and the value of over-regulation.

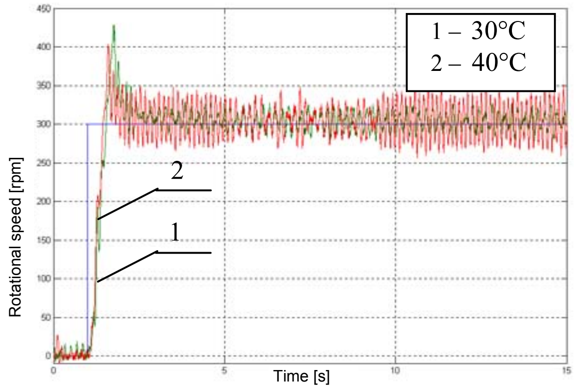

Fig.19. Rotational speed 300 rpm in 30°C and 40°C temperature oil.

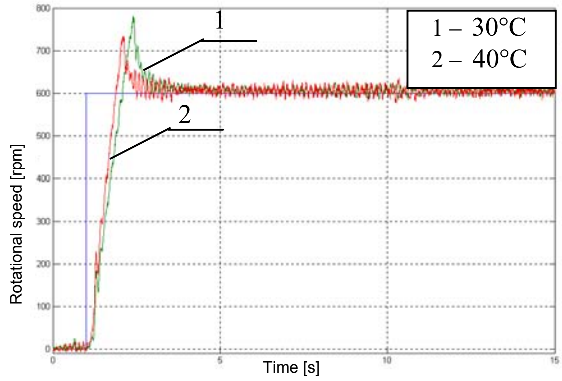

Fig.20. Rotational speed 600 rpm in30°C and 40°C temperature oil.

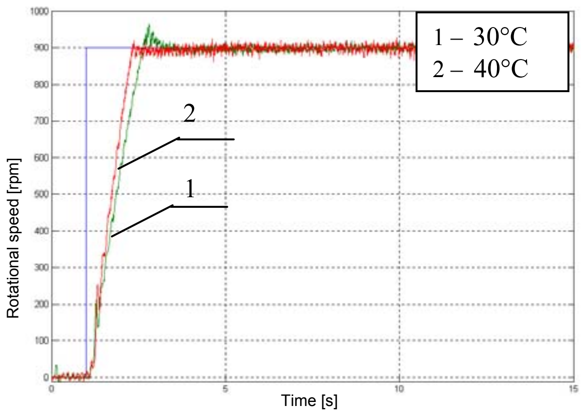

Fig.21. Rotational speed 900 [rpm] in 30°C and 40°C temperature oil.

Conclusions

The research shows that the object is nonlinear. The object's static characteristic has a hysteresis, as well as folds. The hydrostatic transmission is a very difficult to control object. More tests with the controller confirm those results. The controller is only good at one working point and the properties of the working depend on the oil temperature. Raising the oil temperature results in an improvement of performance indices. The tests show that the object is nonlinear and not time invariant (viscosity of the oil is a function of the temperature), therefore, as a final conclusion, a fuzzy controller will be considered in the next investigation.

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