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ELECTROHYDRAULIC SYSTEM DESIGN AND CONTROL

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ABSTRACT

This paper deals with the position control of a hydraulic actuator regulated by a proportional directional control valve. The control algorithms based on the proportional-derivative controller have been applied to a specially designed two-channel electro-hydraulic servo system, which has the capacity to experimentally simulate common circumstances of a typical fluid power circuit. The first system includes the positioning hydraulic cylinder controlled using the proportional valve, a displacement measuring system and a control device in a closed-loop system. In order to add further realism in the control system a second double acting hydraulic cylinder is used to generate the disturbing load. The effectiveness of the proposed control algorithms is demonstrated for both moving directions as well as for both unloaded and loaded system. The results of the experimental testing have shown that for some industrial applications a relatively good motion control performances can be achieved using a proportional valve as a control component with a simple control algorithm, but for the case of changing operating conditions, some advanced control approaches should be used.
Keywords: electro-hydraulic system, position control, proportional valve

1. INTRODUCTION

Hydraulically actuated systems are used in a wide range of industrial applications and mobile systems because of their high power capabilities, high stiffness, self-cooling, high durability and ability to produce large forces at high speed. Hydraulic controlled or servo systems are usually composed of an actuator (cylinder or motor) regulated using the traditional two-stage flapper/nozzle servo valve and a load. The high-bandwidth servo valves give better dynamic characteristics of the controlled systems; however they are more expensive and more sensitive to fluid contamination then the proportional directional control valve. Continuous development and technological improvements of proportional valves reduce the difference between the overall characteristics of proportional and servo valves so that proportional valves can be a suitable solution in many operations, which have traditionally been dominated by servo valves. Electrically actuated proportional or servo valves offer the opportunity of precise hydraulic system regulation using computer control technique.

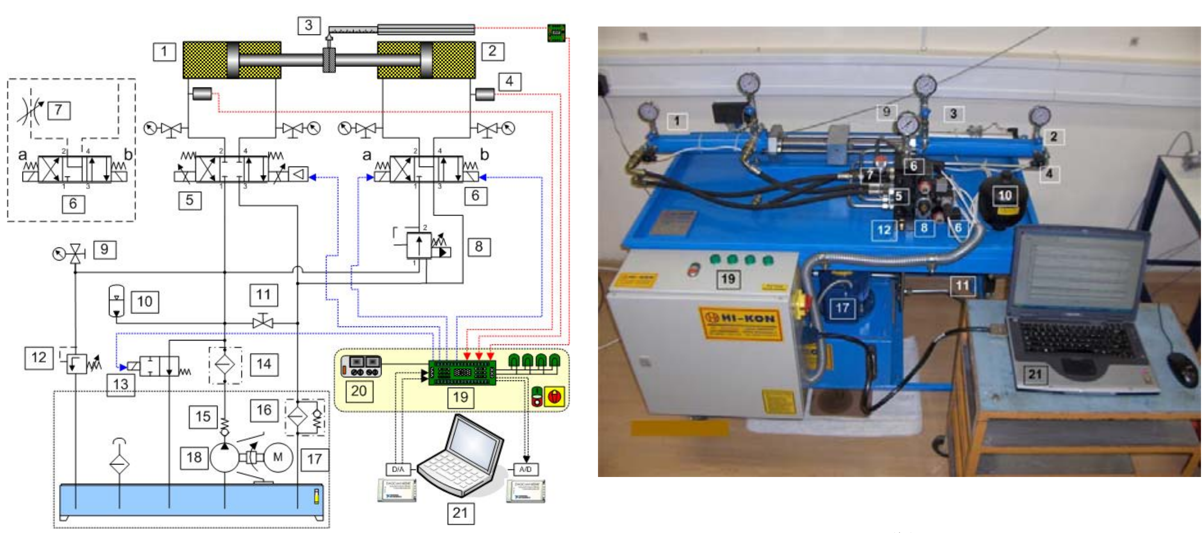
The nonlinear effects in hydraulic systems caused by the phenomena associated with the compressibility of the hydraulic fluid, the complex flow properties of the valve, friction effects, load variations and disturbances make them relatively difficult to control for variable set point applications. Thus, the conventional approach to the control of hydraulic systems mainly based on the linearization of the system around an operating point (typically for a null valve spool position, a nominal loading and actuator mid-stroke position) may not guarantee satisfactory control performance for high precise motion when the operating condition of the system changes. Due to the existing limitations of classical controllers, the idea of using the control strategies that have the ability to cope with changing system parameters, such as the adaptive control algorithms, the variable structure control methods, or modern control techniques based on fuzzy logic has been developed [1-2].

This paper deals with a design and position control of an electro-hydraulic servo system, which consists of two hydraulic cylinders: a main cylinder controlled using a proportional directional control valve and a load cylinder to simulate variations in load.

2. EXPERIMENTAL TEST RIG LAYOUT

To prove experimentally the validity of developed control algorithms, a test device of an electro-hydraulic position servo system is constructed. A schematic diagram of the test rig is shown in Fig. 1a), while Fig. 1b) shows a photo of the actual experimental test configuration.

It consists of two hydraulic cylinders: a main cylinder (1) shown in the left part of the figure, which represents a main servo system and a load cylinder (2) shown in the right part of the figure, which represents a disturbance simulation system. Two cylinders are coupled by a steel joint. Both the main and the load actuator are double acting 300–mm stroke cylinder with a 50–mm bore and 36–mm diameter rod. The motion control of the main cylinder is accomplished using a *Bosch-Rexroth*® three-way proportional directional control valve (5), model 4WRA-E-6-07 with integrated electronics and ±10 V analogue input signal. The electronic driver regulates a proper electrical current supplied to the valve's solenoid according to the calculated control signal from the control device. The solenoid converts the electrical current into a mechanical force acting the spool of the valve against a return spring and the spool displacement allows fluid flow from the pressure port to one of the actuator chambers. The disturbance is generated by using a directional control valve (6) and a load cylinder which is able to generate the reaction force in respect of the main cylinder motion direction. This force is equivalent to the product of the piston area and the controlled pressure which is generated by using a pressure control valve (8). The hydraulic power is provided by a hydraulic gear pump (18), model KV-1P from *ViVoil*, with a maximum rate of 3.7 l/min and maximum nominal pressure of 25 MPa. The oil pump is driven by a single-phase electrical motor (17), 1.1 kW at 1380 rpm. The piston position of the main cylinder along its stroke is measured by using a displacement encoder (3), *Festo*, type MLO-POT-300-LWG, with a resolution of 0.01 mm, which is attached to the actuator. The measured signal from the encoder is used for the realization of a control algorithm for the main servo system. Two pressure transducers (4), *Siemens*, type 7MF1564, 0-10V, are added to measure cylinder pressures. The experimental test rig can also be used to demonstrate the working principle of conventional hydraulic system. In that case the proportional valve should be replaced with another directional control valve and a throttle valve (7) using flexible hydraulic pipes. Data acquisition of the system is handled by a *National Instruments* DAQCard-6024E (for PCMCIA), which offer both 12-bit analogue input and analogue output, and control of the system is accomplished by the use of the computer card in conjunction with the Matlab/Simulink/Real-TimeWorkshop® platform. This control solution allows the continuous monitoring of the process, data acquisition and the real time control.



(a) (b)

1– Main cylinder, 2–Load cylinder, 3–Linear encoder, 4–Pressure sensor, 5–Proportional control valve, 6–Directional control valve, 7–Throttle valve, 8–Pressure control valve, 9–Pressure gauge, 10–Hydraulic accumulator, 11–Shut-off valve, 12–System pressure relief valve, 13–Upload valve, 14–Pressure filter, 15–Check valve, 16–Return flow filter, 17–Electric motor, 18–Gear pump, 19–Electronic interface, 20–DC power supply unit, 21– Control computer with data acquisition/control card

Figure 1. a) Schematic diagram of the control system, b) Photo of the experimental equipment

3. MODEL OF THE CONTROL SYSTEM

The dynamic model of an electro-hydraulic system is nonlinear and can be derived by analyzing the proportional valve dynamics (the relation between the control voltage and the valve spool displacement), then by applying the flow continuity through the valve orifice, by analyzing the pressure behaviour in the cylinder chambers, and finally by applying the force balance equation for the cylinder. Such nonlinear model can be used in simulation studies of the dynamic behaviour of the system [3], but for the purpose of the classical feedback controller synthesis, a linearized control-oriented model is needed, which can still represent the real system behaviour, Fig. 2.

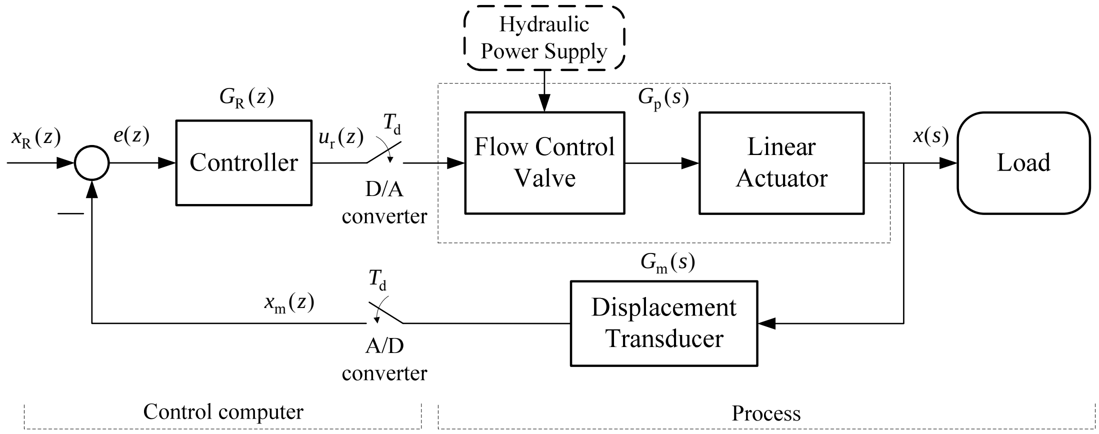


Figure 2. Principal block scheme of hydraulic position control system

The transfer function $G(s)$ that relates the position of the actuator x and the valve control signal u can be expressed as follows:

$$G(s) = \frac{x(s)}{u(s)} = \frac{k_v \omega_v^2 K_q / A_p}{s(s^2 + 2\zeta_v \omega_v + \omega_v^2)\left[\frac{V_t M_t}{4 B A_p^2} s^2 + \left(\frac{K_{ce} M_t}{A_p^2} + \frac{b V_t}{4 B A_p^2}\right) s + 1\right]} \tag{1}$$

The parameters of the experimental test system for controller design are summarized in Table 1. For the case of hydraulic cylinder position control there exists an inherent integral action in the system structure, and the system performance can be improved by introducing a zero near the process time constant using a PD controller. In this structure only the measurement of the cylinder position is necessary for the implementation of the control algorithm. The negative effects of the derivative action are abrupt changes in the control signal. The root-locus of the control system using PD controller is shown in Fig. 3.

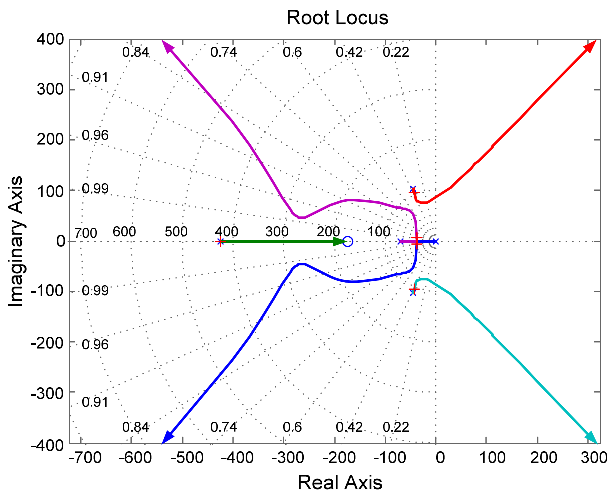


Figure 3. Root-locus of the control system using PD controller

Table 1. Values of the system parameters

System parameters	Value
Valve spool position gain	$k_v = 5.5 \cdot 10^{-6}$ m/V
Valve natural frequency	$\omega_v = 113$ rad/s
Valve damping ratio	$\zeta_v = 0.4$
Valve flow gain	$K_q = 1.433$ m^2/s
Piston annulus area	$A_p = 14.54$ m^2
Cylinder volume	$V_t = 0.654 \cdot 10^{-3}$ m^3
Load mass	$M_t = 580$ kg
Effective bulk modulus	$B = 1350 \cdot 10^6$ Pa
Total flow-pressure coef.	$K_{ce} = 6 \cdot 10^{-11}\ m^3 / Pa\ s$
Viscous damping coef.	$b = 455$ Ns/m
Measuring system gain	$K_m = 33.33$ V/m

4. EXPERIMENTAL RESULTS

The experimental results for the position control of the hydraulic cylinder using a PD controller are shown in Fig. 4, both for unloaded drive and for the case when the load was activated after 8 s. The controller parameter values were: K_P=40 V/m, T_D=0.005 s. The experimental results point out that the stable and well-damped response of the control system is obtained for both directions of actuator motion and for both unloaded and loaded system and the accuracy was within the margin of ± 0.5 mm. For the case of unloaded drive the control system has a relatively fast output, but for the case when the load is activated the controller with fixed gains is not able to obtain the same control performances. This indicates that in the case when the operating condition of the system varies some robust control algorithms should be used in order to redesign the controller gain parameters and to assure satisfactory control performance.

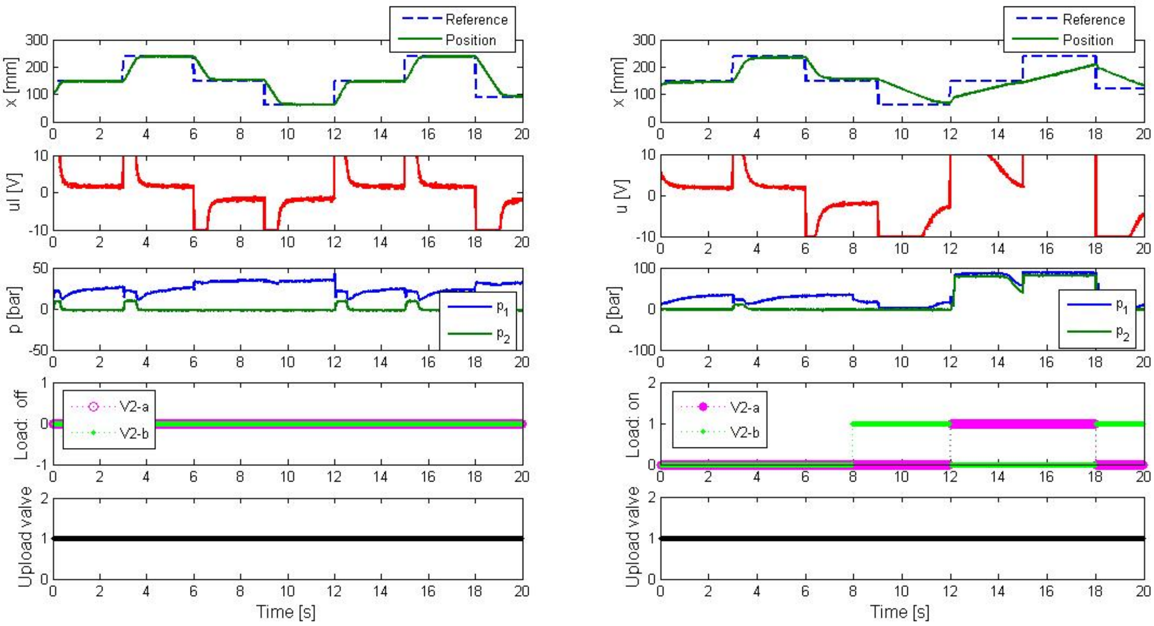


Figure 4. Experimental results for cylinder position control, left: without load, right: with load

5. CONCLUSION

The selection of a flow control valve as a control component depends on the required system performances and hardware cost. If an economical proportional valve can ensure satisfactory system response characteristics, this will reduce the costs of the overall system much more than in the case of using a more expensive high bandwidth servo valve. In this study a proportional directional control valve has been used for the feedback position control of a hydraulic cylindrical actuator. An experimental setup of hydraulic servo system has been made for the purpose of the experimental verification of control algorithms. The test system is equipped with a main and a load cylinder, hydraulic power supply, flow and pressure control valves and is controlled by a PC equipped with a data-acquisition card. The paper provides very initial analysis of simple PD controller application to position control of a hydraulic actuator. In order to get better control performances for the case of loaded drive, some advanced control approaches should be used.

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6. REFERENCES

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