Influence of the Controller Settings on the Behaviour of the Hydraulic Servo Drive

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Abstract. The article presents the influence of the position loop gain setting on the position error values of the hydraulic servo drive. Results of the experimental tests have also been compared with the analytically determined gain values. For the tests, servo cylinder has been loaded with the active force of 2500 N, pointed in the direction opposite to the direction of piston rod extension. The result of the experimental tests was the discovery that the accuracy and stability of the servo drive positioning depends not only on the value of K_v factor, but is also influenced by the direction of the active force loading the servo cylinder. In the tested hydraulic servo drive (hydraulic axis controller), the Compax3F controller by Parker Hannifin was used.

Keywords: hydraulic servo drives, controller settings, hydraulic proportional valves.

1 Introduction

Electrohydraulic servo drives have been in use for many years and are characterized by many beneficial features, such as high stiffness and the possibility to take high values of the velocity enhancement factor [1, 2, 4]. In spite of these advantages, the users are not particularly enthusiastic about hydraulic servo drives. One of the reasons mighty be their slightly different, when compared to typical electric servo drives. In a typical electrohydraulic servo drive, analog control signal is sent either to the modular regulator (mostly PID) or directly to the servo amplifier card with embedded PID regulator. The first solution is used in case of the systems featuring proportional control valves, where the output of the modular regulator is the control signal for the proportional amplifier. The limitations of the presented systems are as follows: no scalability and difficulties in developing/modifying machine drive systems, no possibility of digital control and hindered parameterization of servo drives [5].

The solution to these problems is modern, integrated drive systems that use both electric and hydraulic drives in a similar way. As an example, we could mention IndraMotion systems by Bosch Rexroth or Compax3 by Parker Hannifin [6].

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2 Impact of the Dynamics of Servo Valves on the Adjustment of the Electrohydraulic Servo Drives

Fig. 1 presents the flow chart for the electric servo drive. Due to the specificity of the hydraulic drive systems, it is usually assumed that the movable masses actuated by the cylinders or hydraulic motors are large. It may result in low natural frequencies of the cylinder – actuated inert mass assembly (hydraulic motor – actuated element with the mass moment of inertia). It is commonly assumed that for the hydraulic drives, natural frequencies of movable masses reach a few Hz. If this condition is fulfilled, the dynamics of the electrohydraulic servo drives is not limited by the velocity of the remaining elements of the system.

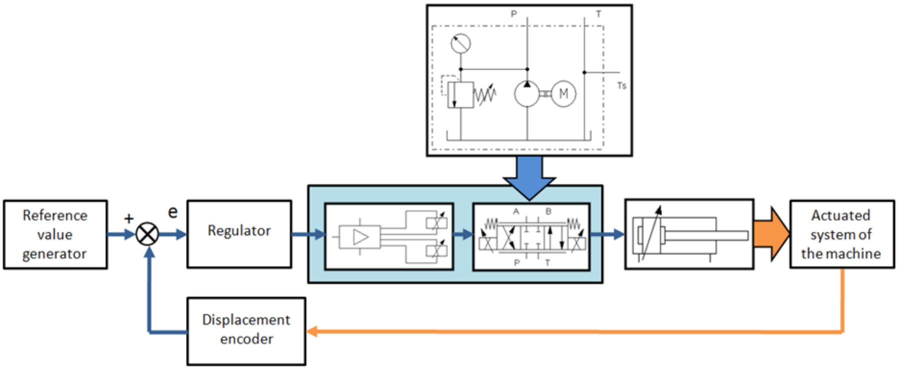

Fig. 1. Flow chart of the electrohydraulic servo drive

This paper presents the procedure of analytical selection of the adjustments for the proportional regulators [3]. First, it is necessary to determine the minimum natural pulsation of the cylinder ω_0, i.e. actuated mass system (hydraulic motor – actuated rotational mass):

$$\omega_0 = \sqrt{\frac{C_{cyl}}{m_r}} \tag{1}$$

where: C_{cyl} – hydraulic stiffness of the cylinder, m_r – movable mass reduced on the piston rod of the cylinder.

In order to do this, we must set the basic parameters of the hydraulic system fragment from the servo valve (proportional valve) to the receiver and inert mass reduced on the piston rod of the cylinder (mass moment of inertia reduced on the hydraulic motor shaft), as shown in Fig. 2.

Due to the fact that for the cylinders we can observe similar throttling on the power supply and outflow, for small oscillations of the piston we sum the hydraulic stiffness of both chambers of the cylinder.

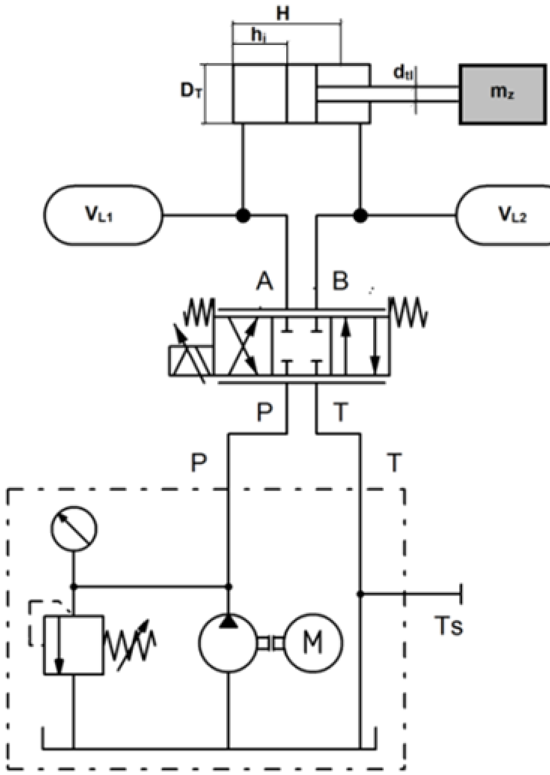

Fig. 2. Hydraulic system diagram used to determine natural pulsation of the actuated element

Hydraulic stiffness C_{cyl} of the liquid locked in the chambers of the receiver is the result of its compressibility:

$$C_{cyl} = C_1 + C_2 = \frac{A_T^2 \cdot B}{A_T \cdot h_i + V_{L1}} + \frac{A_{tl}^2 \cdot B}{A_{tl} \cdot (H - h_i) + V_{L2}} \tag{2}$$

where: H – piston stroke [m], h_i – piston position with minimum stiffness [m], A_T – piston area on the piston side [m^2], A_{tl} – piston area on the piston rod side [m^2], V_{L1} – volume of wires on the piston side [m^3], V_{L2} – volume of wires on the piston rod side [m^3], B – bulk modulus of hydraulic liquid [Pa].

The Equation 2 proves that the hydraulic stiffness depends on the position of the piston. In order to adjust the servo drive we should take the minimum value of stiffness, equivalent of the lowest value of the natural pulsation of the drive system. It will present for the piston position h_k described by the Equation 3.

Having determined the minimum value of natural pulsation of hydraulic drive ω_0, it should be compared with the pulsation of the servo valve ω_V. Pulsation of the servo valve is determined (Equation 3) taking account of the producer's cut-off frequency of a valve f_{res} and a small amplitude control signal.

$$h_k = \frac{\frac{A_{Tl} \cdot H}{\sqrt{A_{Tl}^3}} + \frac{V_{L1}}{\sqrt{A_{Tl}^3}} + \frac{V_{L2}}{\sqrt{A_T^3}}}{\frac{1}{\sqrt{A_{Tl}}} + \frac{1}{\sqrt{A_T}}} \tag{3}$$

Usually, amplitudes of values 5–10 % of the maximum amplitude control signal for a given servo drive are taken:

$$\omega_V = 2\pi \cdot f_{res} \tag{4}$$

Producers provide f_{res} values for the servo valves and proportional control valves (directly or as frequency characteristics of valves). In case of regular proportional valves, not intended to be used in electrohydraulic servo drives, producers provide only step response T_{res}. Here, pulsation of the valve may indicatively be defined as:

$$\omega_V = \frac{\pi}{T_{res}} \tag{5}$$

If we compare natural pulsation of the drive ω_0 with the pulsation of the servo valve ω_V we distinguish [3] two types of electrohydraulic servo drives: $\omega_0 > 3\omega_V$ or $\omega_0 < 3\ \omega_V$.

In the first case, it is not required to take account of the servo drive dynamics to adjust the servo drive regulator. If the regulator is a P type regulator, we should make sure that when we select its amplifier, the value of the velocity enhancement factor K_v for the electrohydraulic servo drive complies with the following equation:

$$K_v = \frac{\omega_0}{3} \tag{6}$$

In the second case, it is necessary to take account of the servo valve dynamics. The value of the velocity enhancement factor K_v for the electrohydraulic servo drive is determined based on the equivalent pulsation ω_{eq}:

$$K_v = \frac{\omega_{eq}}{3} \tag{7}$$

Equivalent pulsation ω_{eq} of the hydraulic system with drive natural pulsation ω_0 and servo valve pulsation ω_V equals half of the harmonic mean of these two values:

$$\omega_{eq} = \frac{\omega_0 \cdot \omega_V}{\omega_0 + \omega_V} \tag{8}$$

It should be mentioned that equivalent pulsation ω_{eq} is smaller than the smallest constituent pulsation. This leads to the conclusion that if we did not take account of the valve dynamics (in the second case), the adopted value of the velocity enhancement factor for the servo drive would be too high. It could result in the instability of the servo drive.

Another conclusion is the fact that we should always make sure that the condition from the first case is met when we select servo valves. Then, the properties of a servo drive will only be limited by the pulsation of the drive (it can be improved by increasing the hydraulic stiffness).

Pairing the hydraulic drive with proportional valve of low dynamics will force the necessity of lowering the value of the velocity enhancement factor of such electrohydraulic servo drive, which in turn will result in much smaller accuracy of the servo drive positioning Δx. Servo drive positioning accuracy can be determined using the following equation [3]:

$$\Delta x = \frac{0.05 \cdot v_{\max}}{K_v} \tag{9}$$

where: v_{max} – drive speed with the servo valve fully opened.

3 Structure of the Lab Workstation

The essential element of the lab workstation is the advanced Compax3 FD2F12 I12 T11 M00 controller. The electronic part also features the position sensor type BTL7-E100-M0305-B-S32 embedded in the servo drive double-acting cylinder with unilateral piston rod CHMIXRPF24M-M1100 (piston diameter 40 mm, piston rod diameter 28 mm, piston stroke 300 mm) connected by the piston rods with the identical cylinder without positioning the piston. Both cylinders have been placed over a rigid foundation. Thanks to this, it is possible to simulate the impact of a variable load on the servo drive behaviour. Servo drive cylinder has been powered from the proportional valve with enhanced dynamics D1FPE01FC9NB00 by Parker Hannifin. Schematic diagram of the lab workstation hydraulic system is presented in Fig. 3.

The remaining elements of the hydraulic system are as follows: power supply with a fixed displacement pump with the possibility of setting the pump shaft rotational speed (power supply nominal parameters: pressure up to 10 MPa, flow rate up to 12 dm^3/min), reduction valve VM064A06VG15 to power both chambers of the cylinder loading the servo drive, necessary pressure inverters, flexible double braided hoses with dry break quick coupling and other essential hydraulic components.

For the described station, the following data essential in order to determine the natural pulsation of the mass actuated by the cylinder and the equivalent pulsation of the servo drive have been adopted: moving mass m = 10 kg, additional volumes $V_{L1} = V_{L2}$ = 86 cm^3 (cables between the proportional valve and cylinder), cut-off frequency f_{gr} = 350 Hz for the valve D1FPE01FC9NB00.

Having used the Equation 3 in order to determine the position of the piston for the minimum hydraulic stiffness (h_k = 287 mm), Equation 2 has been applied in order to determine the pulsation ω_0 of the drive (ω_0 = 1028 1/s). Pulsation of the proportional valve D1FPE01FC9NB00 has also been determined (ω_V = 2199 1/s). Having compared pulsation ω_0 and ω_V we know that we need to determine the value of the equivalent pulsation of the hydraulic system and then use it to estimate the permissible value of the velocity enhancement factor of the servo drive.

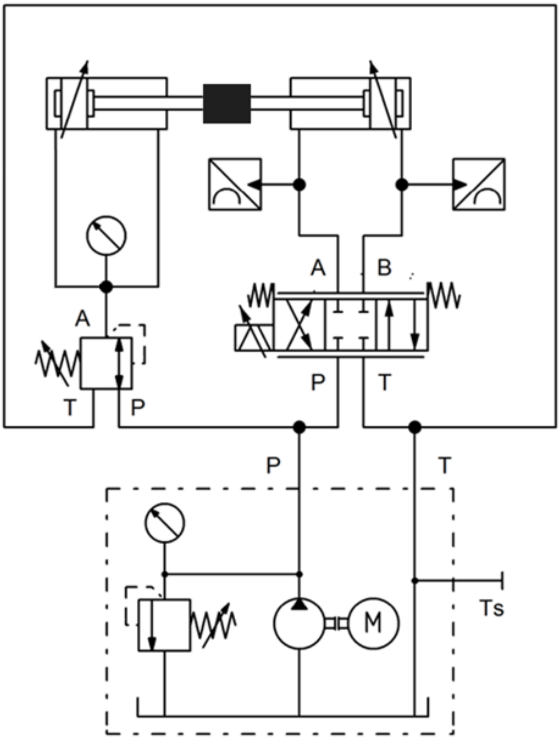

Fig. 3. Schematic diagram of the position for testing the hydraulic servo drive

After substituting the values into the Equation 8 we have received the value of the equivalent pulsation ω_{eq} = 700.7 1/s. This means that the optimum value of the velocity enhancement factor is K_v = 233.5 1/s.

Knowing the value of the suggested velocity enhancement factor of the servo drive, it is possible to determine the settings of the proportional controller (type P) using the following formula [3, 5]:

$$K_v = \frac{K_P \cdot V_q \cdot K_X}{A}$$
$$K_{P+} = \frac{K_v \cdot A_T}{V_q \cdot K_X} \tag{10}$$
$$K_{P-} = \frac{K_v \cdot A_{tl}}{V_q \cdot K_X}$$

where: K_P – proportional gain, K_{P+} – optimum proportional gain while extending the piston rod, K_{P-} – optimum proportional gain while retracting the piston rod, V_q – servo valve gain [cm^3/s/%], K_X – measuring transducer gain [%/cm], A – surface area of the piston [cm^2].

For the valve D1FPE01FC9NB00, V_q = 1.07 [cm^3/s/%] was taken with the assumed pressure drop on the control edge Δp = 1 MPa. Surface areas of the piston are relatively: A_T = 12.57 cm^2, A_{tt} = 6.16 cm^2. BTL7-E100-M0305-B-S32 position sensor amplifier gain is K_X = 3.33 [%/cm], the measuring range is 300 mm.

After substituting the values into the Equation 10 we have received the following values of the P controller settings while: extending the piston rod (K_{P+} = 824), retracting the piston rod (K_{P-} = 404).

4 Results of the Experimental Tests

The default settings of the Compax3F controller were modified prior to the tests. Feed-forward blocks were assigned zero gain for the speed and acceleration in the position regulation track. The integral action of the controller has also been switched off.

Compax3F controller is also capable of compensating the positive overlap of the servo valve. For this reason, the valve D1FPE01FC9NB00 has been previously tested at the separate measuring station in order to determine the values of the valve overlap corrections on the tracks P → A and P → B. The test revealed that the received values (17 % and 16 % relatively) differ significantly from the default values (while parameterizing the controller, the C3 ServoManager2 system has taken the default value of 23 %). Also modified were the default gain values correcting different gains of the cylinder with unilateral piston rod depending on the powered chamber. Additional gains were adopted, all equal to 1.

Variations for the selected values of the P-term parameter settings of the Compax3F controller relating to the K_v of the servo drive were presented in Fig. 4, Fig. 5 and Fig. 6.

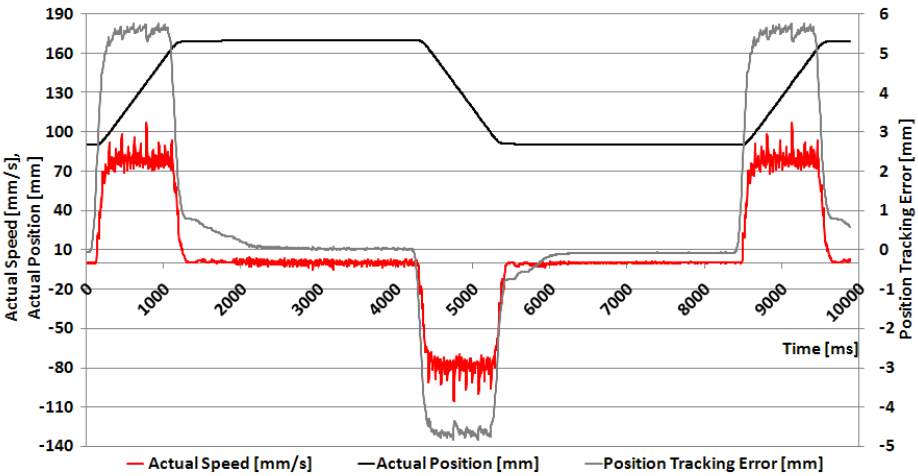

Fig. 4. Variations for the position loop gain setting P-term = 10

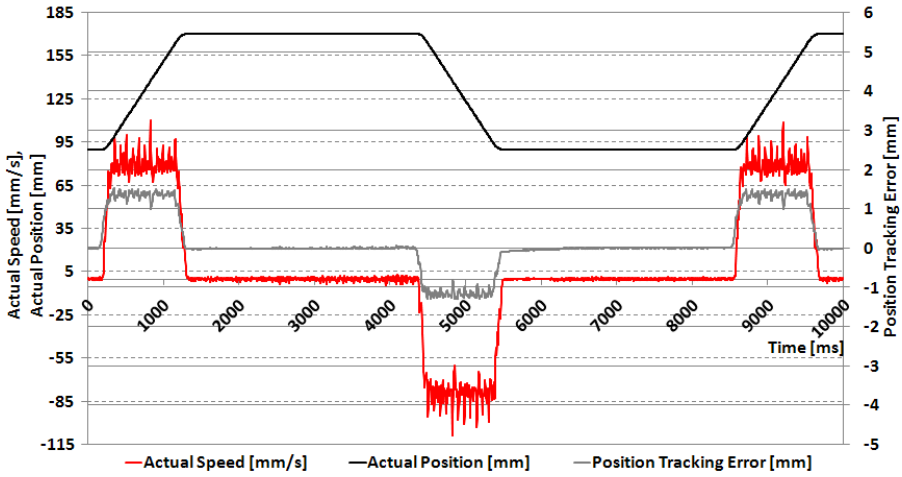

Fig. 5. Variations for the position loop gain setting P-term = 40

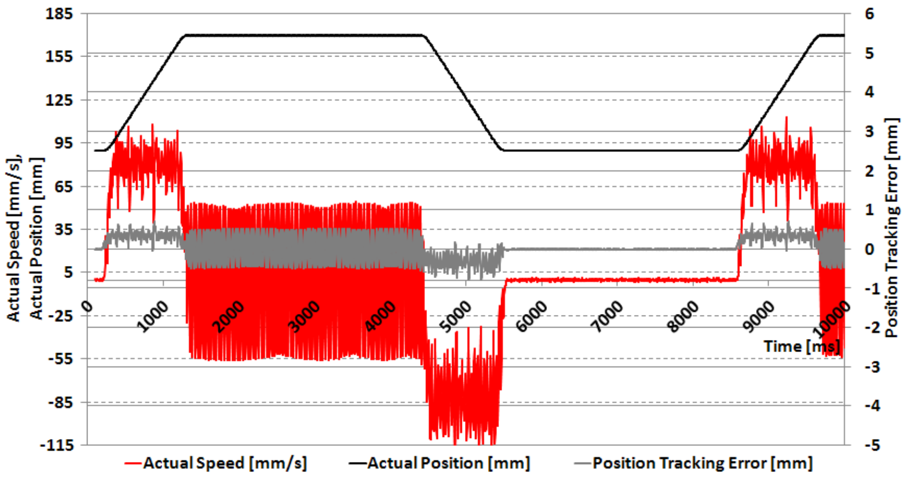

Fig. 6. Variations for the position loop gain setting P-term = 160

For the variations received as a result of the experimental tests, obtained for the subsequent adjustments of the position controller proportional gain in the Compax3F controller with the preset speed of 80 mm/s, the following values have been determined for the extension and retraction of the piston rod (average speed rate and average position offset, position loop gain, effective regulation error and shape factor).

The selected results of the conducted tests have been presented in Tables 1 and 2. The results presented in Table 1 reveal that the analytically determined value of the optimum velocity enhancement for the tested servo drive is slightly inflated. In case of the results (row No. 9) we can observe unstable operation of the servo drive, even though the adjustment of the K_v factor did not exceed the analytically determined value.

Table 1. Results of the experimental tests regarding the piston movement

Average regulation error [mm]	Average speed [mm/s]	Gain K_v [1/s]	Direction of motion	Position loop gain P-term [%/unit]	Notes
5.538	79.875	14.424	extension	10	–
–4.608	–79.901	17.339	retraction	10	–
2.603	79.967	30.727	extension	20	–
–2.253	–79.695	35.376	retraction	20	–
1.345	79.674	59.239	extension	40	–
–1.153	–79.940	69.315	retraction	40	–
0.678	79.516	117.217	extension	80	–
–0.593	–79.553	134.064	retraction	80	–
0.356	79.300	223.009	extension	160	Loss of stability
–0.337	–79.776	236.502	retraction	160	–

Table 2. Analysis of the regulation errors in the preset servo drive position for the selected controller settings

Average regulation error [mm]	Effective regulation error [mm]	Direction of motion	Position loop gain P-term [%/unit]	Shape factor	Notes
0.020	0.025	extension	10	1.180	–
–0.077	0.077	retraction	10	1.001	–
–0.003	0.011	extension	20	4.187	–
–0.035	0.038	retraction	20	1.090	–
–0.001	0.011	extension	40	12.649	–
0.007	0.009	retraction	40	1.209	–
0.001	0.014	extension	80	13.359	–
0.005	0.007	retraction	80	1.355	–
0.013	0.358	extension	160	28.361	Loss of stability
0.006	0.008	retraction	160	1.368	–

Also the comparison of K_v factor in pairs: while extracting and retracting, can be seen as surprising. Gains of the electronic part of the servo drive were constant and independent of the direction of movement. The only difference was in the cylinder gain, which was the result of different surface areas of the piston while extending and retracting the piston rod. Theoretically, we should obtain twice as high values of the K_v factor while retracting the piston rod. Also the ratio of K_v while retracting to K_v while extending the piston rod should be constant. Tests confirmed neither of those two expectations. In order to explain these discrepancies, additional tests should be performed.

The overall results of examining the positioning accuracy of the servo drive were presented in Table 2. Due to the possibility of oscillation occurring in the set position, not only the average value of the regulation error, but also the effective regulation error was determined. In order to measure the oscillation we adopted the form factor, defined as the absolute value of the effective error to average regulation error ratio. The obtained results are to some extent consistent with the theoretical expectations. Initially, along with increasing the controller gain, both average and effective regulation error were decreasing. It has been observed that starting with the setting P-term = 40 for the position reached with the extended piston rod, the RMS value of control deviation increases, whereas the average position error decreases. The increasing tendency of the servo drive to oscillate in the steady state is evidenced by the significant increase of the shape factor value (from 4.188 for P-term = 20 to 12.649 for P-term = 40). For the setting P-term = 160, it may be assumed that there has been a loss of stability of the tested hydraulic servo drive. The positioning accuracy was by an order of magnitude worse (from 0.0114 mm for P-term= 40, to 0.3581 for P-term = 160). It is worth mentioning that the direction of the external load was contrary to the direction of the servo cylinder's movement.

5 Summary

The analysis of the obtained results of the experimental tests suggests that even for simple systems we may observe significant discrepancies between the theoretically predicted values of the servo drive controller's settings and the values of settings for which the actual servo drive works properly. Using the information received, we are able to state that theoretical settings are a good reference point to optimize the controller settings by means of the experiment carried out with the actual operating conditions of the machine.

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