Impulse Identification and Discrete P/PD Control of Electro-Hydraulic Servodrive

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Abstract:
This article presents the systematic design methodology of a discrete Proportional and Proportional-Derivative (PD) controller for the electro-hydraulic servodrive position control. The controller is based on the identified linear model of the system, with P/PD parameters adjusted with the help of different methods given in the literature. There are compared experimental results of the proposed control system with different controller parameters.

Keywords: *PD controller, Identification for control, Analytic design, Identification and control methods.*

1. Introduction

Nowadays electro-hydraulic servodrive systems are very important components of machines and technological lines because of their high power to weight ratio, high stiffness, and high payload capability. Their unrivaled high energy efficiency, exceeding the efficiency of devices using other media, ease of control, the possibility of obtaining large gear ratios and low inertia make them perfect elements of high precision mechanical systems [1]. They can perform fast and precise mold feed in injection molding machine [2], moving and stacking products on the production line or drilling and tightening screws with a specific torque [3]. In aircraft technology, the operation of landing gear, flaps, flight control surfaces, and brakes is largely accomplished with hydraulic power systems [5]. In automotive industry electro-hydraulic actuators are used in active suspension systems [6]. Recently, they become popular due to the development of walking robotics, where it is necessary to use the low-power input signal and its conversion to high power output [7].

However, the control of electro-hydraulic systems can be a difficult task since their dynamics is highly nonlinear [6]. Therefore, the research is conducted on the position control or the force control for electro-hydraulic actuators using more advanced control techniques i.e. feedback linearization [8], adaptive control [9] or sliding mode control [10].

Despite all the progress in the advanced control, the Proportional-Integral-Derivative (PID) algorithm remains the most popular. Its gains are often chosen based on experience and through some simple selection methods such as Ziegler-Nichols [11] or Cohen-Coon [12]. However, regardless of the type of controlled process, there is usually requirement of the exact response of the system to set-point changes and disturbances. Without a proper methodology of controller parameters tuning the quality of the control system may be unacceptable. Therefore many researchers in academia and industry develop tuning rules for different processes, with different objectives. The survey presented in [13] gives the total of 1134 separate rules for PI and PID controllers, and one can expect that until now this number has increased. Recently there are developed methodologies to choose proper tuning rules and improve the performance of the control system [14], [15].

This paper presents the systematic design methodology of a discrete P/PD controller for the electro-hydraulic servodrive position control, based on the identified linear model of the system, and controller parameters tuned using different methods.

The article is organized as follows. Section 2 describes the laboratory test stand as an electro-hydraulic system. Section 3 describes the step identification of the electro-hydraulic servodrive and presents an obtained model of the control object in the form of the transfer function. Section 4 gives a brief introduction to the discrete P/PD controller design procedure and methods of tuning its parameters. Section 5 describes experiments carried out on the laboratory test stand and there is also given a discussion on the performance of the control system. In the last Section the conclusions from the design and experiments are presented, and a proposal for a further investigation is given.

2. Laboratory Test Stand

The operation of the PD controller was tested on a hydraulic test stand, whose structure is shown in the Fig. 1.

The laboratory test stand system consists of a few main parts: hydraulic pump, pressure relief valve, manometer, filter, servo valve, piston, linear position encoder, and PC computer with MATLAB/Simulink[1] software and control-card dSpace DS1104[2]. The lab-

[1] https://www.mathworks.com/

[2] https:/www.dspace.com/

oratory test stand consists of electro-hydraulic servo system shown in the Fig. 2 and hydraulic power station shown in the Fig. 3.

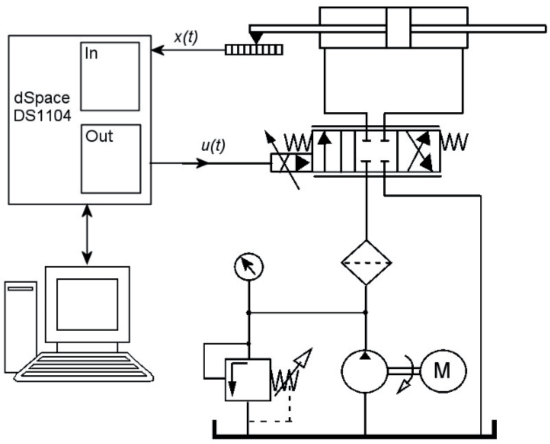

Fig. 1. Schematic diagram of the electro-hydraulic system with PD controller

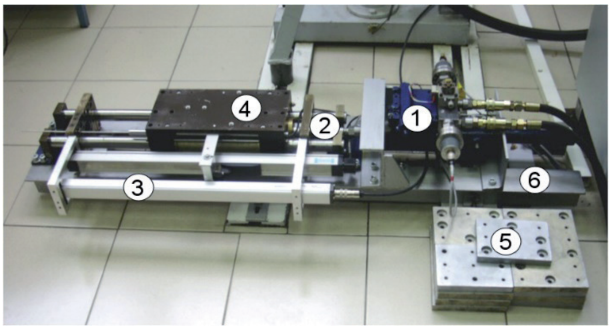

Fig. 2. View of the electro-hydraulic servo system: 1) servo-valve, 2) piston, 3) position encoder, 4) load platform, 5) mass, 6) support

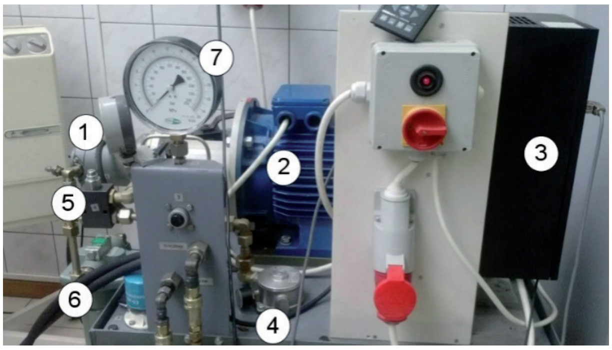

Fig. 3. View of the hydraulic power station: 1) pump Hydral PT02, 2) electric motor, 3) variable frequency drive, 4) oil tank, 5) pressure relief valve, 6) filter, 7) manometer

In the system there is used a double-acting actuator. In order to stabilize the movement of the actuator, the platform is positioned on slideways. The position of the piston rod is changed by the servo-valve, controlled by the voltage signal in the range [–10 V, 10 V]. The position of the actuator's piston rod is obtained by means of a magnetostrictive transducer.

The data transfer between the position transmitter, the regulator and the servo-valve is carried out using the 1104 dSPACE controller card with 16-bit analog-to-digital converters (ADCs).

The controller algorithm and the data acquisition are done using the PC computer and Matlab/Simulink software.

A feedback in the system is obtained by means of a position transmitter whose signal is compared with a set-point signal. This way, the information about the current position error is received and it goes to the controller. On this basis, the controller algorithm generates a control signal that changes the position of the servo valve, which in turn affects the position and velocity of the piston rod of the actuator. Changing the position of the piston rod causes a change in the position signal from the transducer.

The direction and the speed of the actuator was controlled by the two-stage electro-hydraulic servo valve Dowty 4553. The first stage of the valve uses a flapper nozzle and the torque motor. The input current from the DS1104 control card controls the torque motors and this same the flapper position. The flapper position controls the pressure in both chambers of the second stage of the valve – spool valve. The change of input current changes the flapper position and the pressure in chambers on both sides of the spoon in the valve, which cause the servo to move in one direction or the other.

The main advantage of this system is that a low power electrical signal can be used to accurately position an actuator, and the speed of the actuator is almost proportional to the electrical input control signal.

3. Procedure of the Controller Parameters Tuning

The methodology used in the presented research divides the controller tuning process into three main steps [15]:

1) Process identification.
2) Calculation of the controller parameters.
3) Verification of the control system performance in time and frequency domains.

Thorough procedure divided into different sub-stages is presented in Fig. 4.

4. Impulse Response Identification

4.1. Theory

The simplified model of the electro-hydraulic servodrive is usually presented in the form of a serial coupling of the proportional gain, the oscillating component and the integrating element [16], with parameters approximation based on the physical properties of the system.

Another method is the ARMAX model identification based on experimental data [17]. This method is popular because of its high reproducibility of the electro-hydraulic servodrive model and the ability to describe it in the state-space form.

During the research, there was used a simple, practical method for the calculation of the 2nd order inertial model for astatic systems based on the system pulse response [18].

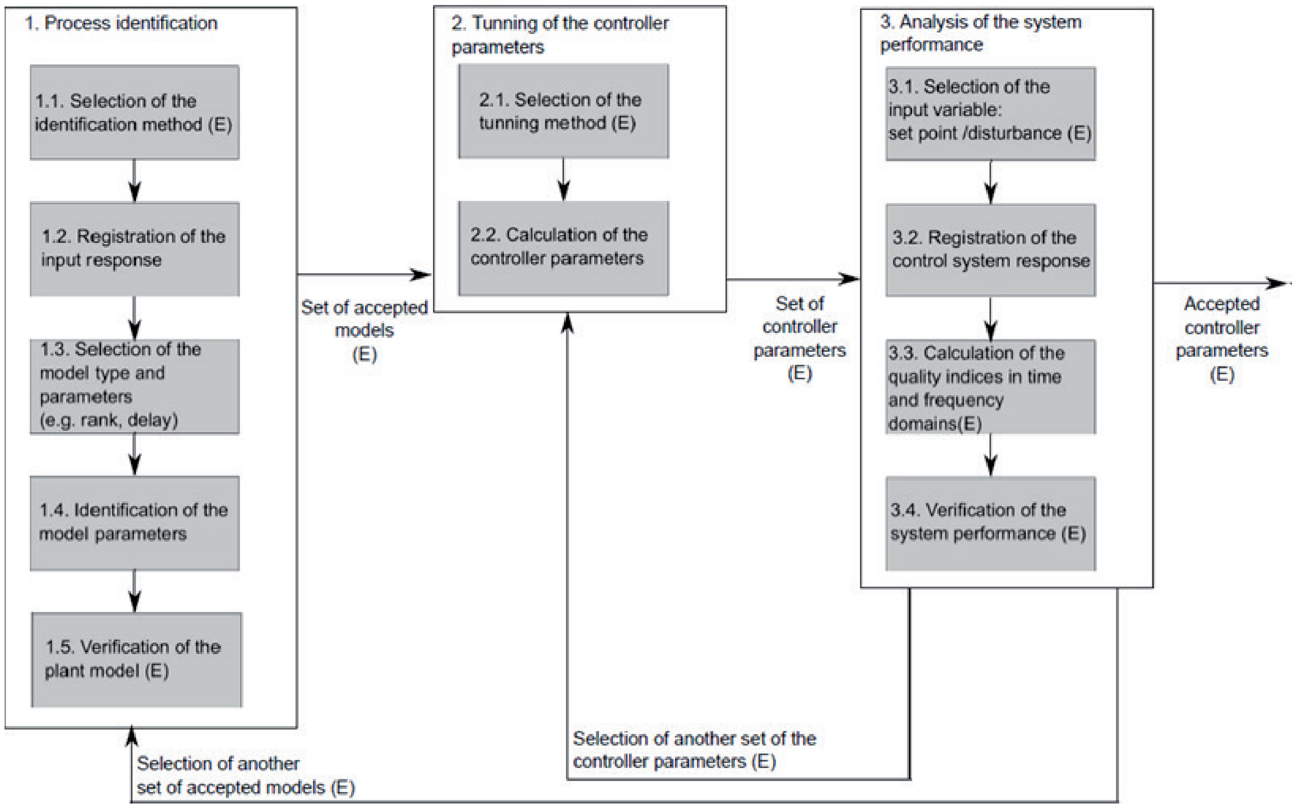

Fig. 4. Procedure of PD controller tuning process [15]

The transfer function of the identified model is described as follows

$$G(s)=\frac{1}{T_I s}\frac{1}{Ts+1}e^{-T_{02}s} \tag{1}$$

Model (1) is the First Order Lag Plus Integral Plus Time Delay (FOLIPD) model, from the class of non-self-regulating process models.

The identification procedure should be performed in the open loop, by providing a pulse input signal

$$u(t)=u_0\left[1(t)-1(t-T_u)\right] \tag{2}$$

where: T_u – the known pulse period.

Then, the object's response described as $p(t)$ should be analyzed as it is presented in Fig. 5.

It can be seen, that $p(t)$ for $t > T_u$ can be described as follows

$$p(t)=u_0\frac{1}{T_I}\left[T_u-Te^{-\frac{t}{T}}\left(e^{\frac{T_u}{T}}-1\right)\right]\xrightarrow[t\to\infty]{}u_0\frac{T_u}{T_I} \tag{3}$$

Therefore

$$T_I=\lim_{t\to\infty}\left(u_0\frac{T_u}{p(t)}\right)=u_0\frac{T_u}{p_\infty} \tag{4}$$

where p_∞ is the impulse response in the steady state.

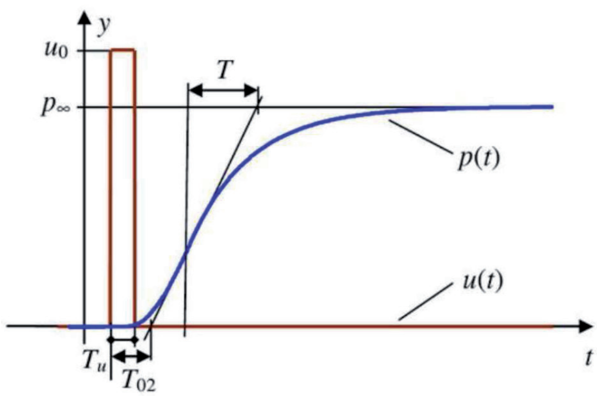

Fig. 5. Impulse response of an astatic object [18]

Determination of missing parameters of the searched model (T and T_{02}) requires analysis of the obtained impulse response $p(t)$ (see Fig. 5). It was shown in [18] that they can be calculated using the tangent the same way as for inertial model, in the following steps

Determination of the FOLIPD parameters.

1) Find an inflection point, i.e. point where $\ddot{p}(t_0)=0$.
2) Calculate slope coefficient of the tangent line at the inflection point as $a=\dot{p}(t_0)$.
3) Calculate the bias of the tangent as

$$b = p(t_0 - at_0) \tag{5}$$

4) Calculate T defined as the time difference between the moment when the tangent reaches asymptote p_∞, and time of the inflection point (t_0).

5) Calculate T_{02} defined as determined as the time difference between the moment when the tangent has value 0 and the moment of impulse excitation.

Then, the values of T and T_{02}, together with the previously calculated value T_I can be finally substituted into the transfer function (1).

4.2. Experiment

The identification was carried out on a laboratory stand, giving to the servo valve the impulse signal with an amplitude u_0 = 5 V and a pulse period T_u = 0.1 s. The obtained response allowed to determine the position of the actuator piston rod in a steady state p_∞ = 4.3986 mm and the local point of inflection at the time t_s = 0.601 s. On this basis, a tangent was designated at the point t_0 described by the formula

$$y(t)=0.0395t-20.9918 \tag{6}$$

On this basis, the intersection points of the tangent line with 0 and p_∞ have been determined using relations from Section 4.1, the model of the control object is obtained as the transfer function

$$G(s)=\frac{1}{0.0144s(0.0422s+1)}e^{-0.0318s} \tag{7}$$

5. Controller Design

To implement the PD controller on a microprocessor system, it should be determined in a discrete time form. Such implementation was crucial in the electro-hydraulic servodrive control system with dSPACE controller-card used during research.

5.1. Discrete PD controller algorithm

In digital implementations, an incremental form is often used, i.e. the equation calculating not the absolute value of the control signal but its increase. This is due to the fact that it allows impactless switching of operating modes (manual work/ automatic work) and easier implementation of anti-windup algorithm [19].

The discrete time form of the PD controller can be described as

$$u(k)=P(k)+D(k) \tag{8}$$

with the **proportional term** described as

$$P(k)=k_p e(k) \tag{9}$$

the filtered **derivative term** described as

$$D(k)=\frac{T_d}{T_p k_d+T_d}D(k-1)+\frac{k_p T_d k_d}{T_p k_d+T_d}\left[e(k)-e(k-1)\right] \tag{10}$$

and the error signal calculated as

$$e(k)=y(k)-y_{SP}(k) \tag{11}$$

where $y_{SP}(k)$ – set point value, k– discrete time, $t = kT_p$, T_p – sampling time, k_p – coefficient of the proportional term, T_d – coefficient of the derivative term, k_d – dynamic gain.

Assuming that

$$u(k)=u(k-1)+\Delta u(k) \tag{12}$$

the following incremental PD controller algorithm can be stated

$$\begin{cases} \Delta u(k)=\Delta P(k)+\Delta D(k) \\ \Delta P(k)=P(k)-P(k-1)=k_p\left[e(k)-e(k-1)\right] \\ \Delta D(k)=D(k)-D(k-1)=\dfrac{T_d}{T_p k_d+T_d}\Delta D(k-1)+ \\ \dfrac{k_p T_d k_d}{T_p k_d+T_d}\left[e(k)-2e(k-1)+e(k-2)\right] \end{cases} \tag{13}$$

with the following initialization values

$$\begin{cases} \Delta D(k-1)=0 \\ e(k-1)=0 \\ e(k-2)=0 \end{cases} \tag{14}$$

5.2. Tuning rules

From the set of PD tuning methods for FOLIPD process described in [13], there were chosen 4 different methods, namely: Coon method (CM, [20], [21]), Haalman method (HM,[22]), Van der Grinten method (VG, [25]) and Viteckova method (V, [23], [24]).

The tuning rules for abovementioned methods are presented in Tab. 1[3].

Table 1. P and PD controller tuning rules for FOLIPD process

Method	k_P	T_d
Coon	$\frac{x_{1C}}{K_m(\tau_m+T_m)}$	0
Haalman	$\frac{0.6667}{K_m\tau_m}$	T_m
Van der Grinten	$\frac{1}{K_m\tau_m}$	$T_m + 0.5t_m$
Viteckova	$\frac{x_{1V}}{K_m\tau_m}$	T_m

Parameter x_{1C} for the Coon method is chosen on the basis of the value ratio $r = \tau_m/T_m$ according to Tab. 2.

[3] For FOLIPD model (1) the following conversion of the coefficients is required $K_m = 1/T_I$, $T_m = T$, $t_m = T_{02}$

Table 2. Values of the parameter x_{1C} depending on the ratio r values

r	x_{1C}	r	x_{1C}	r	x_{1C}
0.020	5.0	0.25	2.2	4.0	1.1
0.53	4.0	0.43	1.7		
0.110	3.0	1.0	1.3		

Parameter x_{1V} for the Viteckova method is chosen according to expected overshoot k as it is stated in Tab. 3.

The parameters of P and PD controllers for the identified model (7) and calculated using 4 different tuning methods are presented in Tab. 4[4].

Table 3. Values of the parameter x_{1V} depending on the expected overshoot κ

x_{1V}	k	x_{1V}	k	x_{1V}	k	x_{1V}	k
0.368	0	0.641	15	0.801	30	0.957	45
0.514	5	0.696	20	0.853	35	1.008	50
0.581	10	0.748	25	0.906	40		

Table 4. Controller setting values according to different methods

Method	k_P	T_d
Coon	0.1997	–
Haalamn	0.2383	0.0101
Van der Grinten	0.3575	0.0207
Viteckova	0.1315	0.0056

6. Experiments

Effectiveness of different tuning methods and controllers has been confirmed experimentally. Within proposed experimental setup, there was possible to change input signal $u(t)$ from P/PD controller in automatic or manual mode. Therefore theoperator could perform identification procedure or examine thecontrol system by changing set-point value $y_{SP}(t)$.

6.1. Quality criteria of the control system

The quality of PD control system was analyzed in the time domain using the following criteria [14]

Steady state error of the pistons' linear position – $e_{x\,stat}$

Overshoot

$$\kappa = \left|\frac{e_2}{e_1}\right| \cdot 100\% \qquad (15)$$

where e_1, e_2 – the first 2 consecutive biggest errors with opposite signs, assuming the steady-state value of position after transient response as the zero level (baseline).

Transient response time t_r – which is the time between the beginning of input change (t_0) and the moment after which the error signal remains inside a boundary $d = 5\%e_{max}$.

Integral Time Absolute Error quality index

$$ITAE = \int_{t_0}^{t_r} t\left|e(t)\right| dt \qquad (16)$$

Integral Time Absolute Control quality index

$$ITAC = \int_{t_0}^{t_r} t\left|u(t)\right| dt \qquad (17)$$

6.2. Results

The step responses of control loops for different controller tuning methods of the electro-hydraulic system described in Section 2 are shown in Fig. 6-7. The control signal from the different controllers for the step response input are shown in Fig. 8-11. The quality parameters for different tuning methods are gathered in Tab. 5.

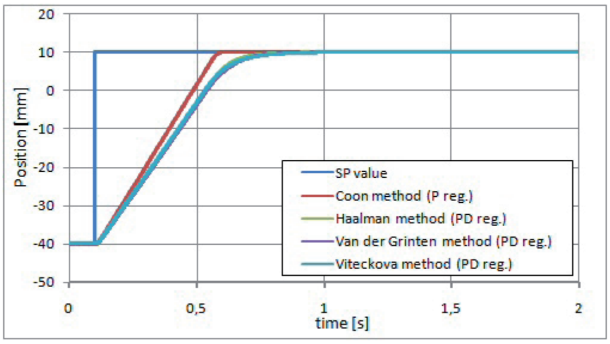

Fig. 6. Step response of electro-hydraulic servodrive with different controller settings

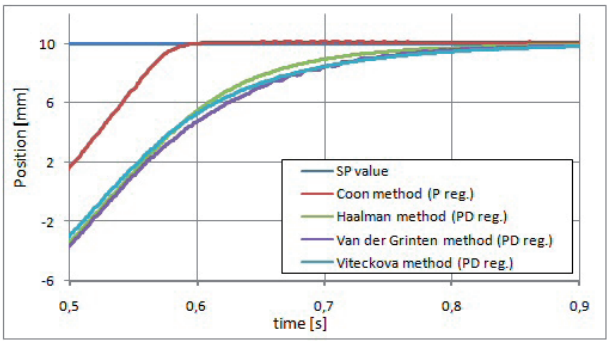

Fig. 7. Zoom of step response of electro-hydraulic servodrive with different controller settings

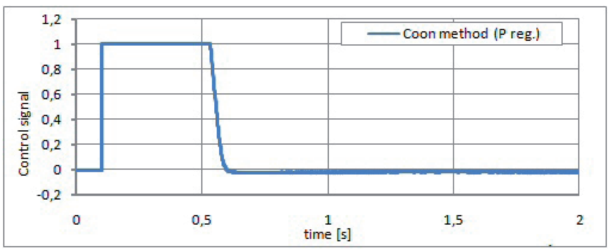

Fig. 8. Control signals for Coon method – P controller

[4] For all PD controllers dynamic gain was chosen as $k_d = 8$.

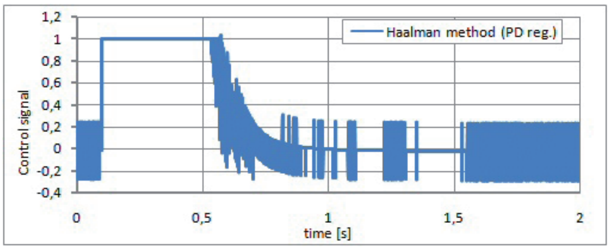

Fig. 9. Control signals for Haalman method – PD controller

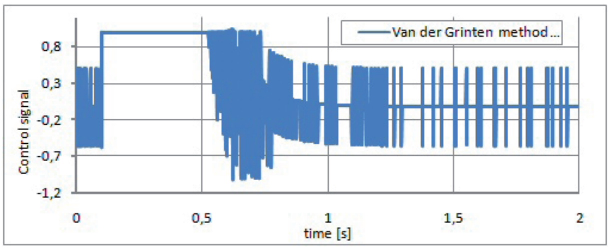

Fig. 10. Control signals for van der Grintenmethod – PD controller

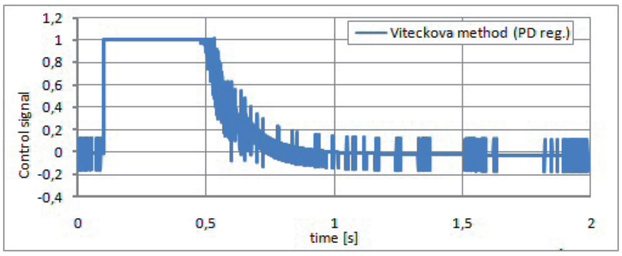

Fig. 11. Control signals for Viteckova method – PD controller

Table 5. Table of quality control indicators for different control systems

Method	$e_{x\,stat}$	k[%]	t_r	ITAE	ITAC
Coon	0.05	0.31	0.50	2012	113
Haalman	0.06	0	0.84	2664	149
Van der Grinten	0.07	0	0.89	2760	201
Viteckova	0.10	0	0.90	2694	133

7. Conclusions

This paper discuss the control system design methodology and quality analysis of the electro-hydraulic servodrive position control system with a discrete P/PD controller. It is based on the identified linear model of the system (First Order Lag Plus Integral Plus Time Delay – FOLIPD) model, and the performance is compared for controller parameters tuned using4 different methods. It should be emphasized that PID-type controllers are not recommended for servo control by hydraulic drives because the plant already have integral properties.

Experimental results show that in terms of the steady state error and the transient response time, the proportional (P) controller (with settings determined by the Coon method)managed best, but it was characterized with a small overshoot. In contrary, proportional-derivative (PD) controllers, although they reached the set point in a slightly longer time, did not have the overshoot. In addition, the advantage of PD controller is a larger band of the gain margin. Quality parameters of all evaluated PD controllers tuned with different methods (Haalman method, Van der Grinten method or Viteckova method) are similar to each other, despite the fact that every method gave different controller parameters. Unfortunately, the disadvantage of PD controllers is the visible noise in the control signal – in this terms, the Viteckova method is the best one, because of the smallest noise to signal ratio. On the other hand, in electro-hydraulic servodrive systems, a high frequency signal (called dither signal) is often added to control signal. It is used to reduce the hysteresis of the electromechanical transducer and keeps the servo valve spool in constant motion, thus reducing the static friction value.

In the future, it is planned to compare the frequency characteristics of the presented control system with different settings of P and PD controller, and perform more elaborated quality analysis of the control signal with filtering and the dither signal shaping.

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