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DESIGN AND TUNING OF THE LYAPUNOV BASED NONLINEAR POSITION CONTROL OF ELECTROHYDRAULIC SERVO SYSTEMS

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Abstract. *A complete study of the development of a nonlinear backstepping controller for an electrohydraulic servo system is shown in this paper. The authors present an optimized nonlinear mathematical model used as fundamental for computer simulation. A proposed nonlinear controller is suitable for research of behavior of the complete system in control. Special attention is paid to the selection of tuning parameters. Using the experience of earlier studies of the state-space controller where the additional feedback signals such as velocity and acceleration signal increase the frequency and damping factor of the system, the results were proved by computer simulation. The results show that by appropriate selection of tuning parameters the system can achieve the best reference signal tracking performance with a small tracking error. The proposed approach seems to be adequate not only for step reference signals but also for ramp and sinusoidal reference signals. However, the parameters of the backstepping controller can be optimized manually to achieve the best results required.*

Key Words: *Nonlinear Control, Nonlinear Modeling, Lyapunov Methods, Electrohydraulic Servo System, Computer Simulation*

1. INTRODUCTION

Electrohydraulic servo systems (EHS-systems) are used in several fields of industry and represent an important part of automatization. The systems like robots, computer numerical control machines (CNC), press brakes, and computer-controlled testing machines use the advantage of hydraulic power systems. Additionally to this, the positioning systems have some specific use spatially in testing machines, where the dynamic and accuracy are very important. Smooth response characteristics with the

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capacity of generating high force and positioning accuracy are important for use in the area of testing machines ideal for material or testing of constructions.

This capability in positioning exerts a significant impact on modern equipment for position control applications [1,2]. Different authors present research proof of the usage of a different approach to improving the positioning of Electro-Hydraulic Servo (EHS) systems [3,4]. The development of a suitable controller, which could cover most demands is very significant. It is well known that the dynamics of EHS systems are highly nonlinear [5-8].

The system model is normally stiff with fast dynamics for the hydraulics and relatively slow dynamics for the mechanical parts. Models usually contain strong nonlinear elements such as the flow in orifices, friction, valve overlap, and input saturation. The oil characteristics depend on parameters such as temperature and air content [9-11]. Therefore, the control strategy must be based on a nonlinear type of controller such as the nonlinear integrator backstepping approach. Emphasis is essentially on the effect of the tuning parameters and on how it influences the dynamic behavior of the system. The backstepping control ensures the global asymptotic stability, the tuning parameters of the controller, nonetheless do greatly affect desired dynamic behavior [4].

A commonly used approach in the design of feedback systems, use the method of feedback linearization to eliminate nonlinearities. Feedback linearization employs changes of coordinates to transform a given nonlinear system into an equivalent linear one. A major advantage of the feedback linearization approach is related to the cancelations of the system's nonlinear dynamics that are introduced in the design process [9-11]. Some nonlinearities have positive effects on system stability; therefore, their cancellation can achieve the opposite effect, which can lead to unstable operation of the system. As a solution to this problem the integrator backstepping approach is an appropriate solution. The fundamental concept of the backstepping method is introduced by Krstic et al. in their book [12].

The approach focusing on the stabilization problem in stochastic nonlinear systems is developed in the extension of this book. The backstepping control method is also presented in [13-15] where this technique is explained in detail for regulating and tracking problems.

Numerous applications in the industry were used to obtain successful control of electric machines, wind turbines, based power production, robotic production systems, and flight trajectory control.

A backstepping controller is one of the recursive designed controllers. It is designed by step back toward the control input starting with the scalar equation which is separated from it by the largest number of integrators. Because of this recursive structure, the design process of this controller starts at the known stable system and back out new controllers that progressively stabilize each other subsystem. The process terminates when the final external control is reached [9-11].

Without uncertainties, backstepping can be used to force a nonlinear system to behave like a linear system in a new set of coordinates. Backstepping can avoid cancellations of useful nonlinearities and pursue the objectives of stabilization and tracking reference input. For tracking problem backstepping always use the error between the actual and desired input to start the design process.

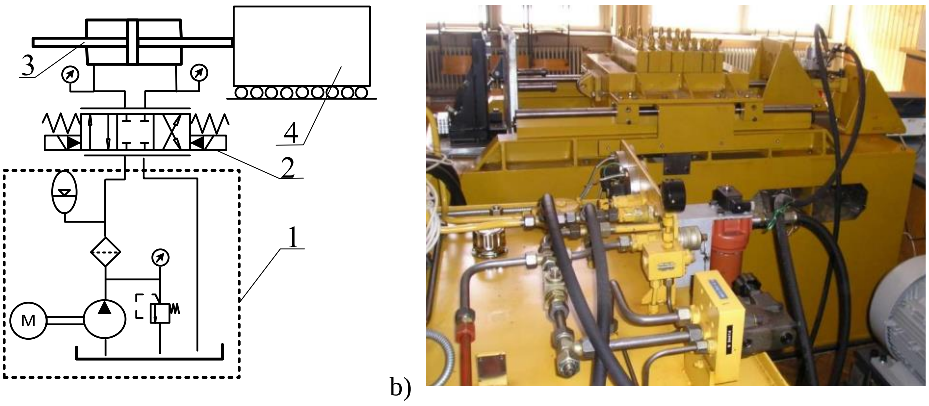
This method of control also provides a solution for optimal control problems [16], estimating parameters and adaptive control design as well as the development of robust nonlinear controllers. The observer-based backstepping technique has been designed for force control of the electrohydraulic actuator system [17] and an active suspension control system [18]. To obtain the best value for its tuning parameters the particle swarm optimization tool was proposed [19].

It is proved that the technique improves the transient stability and damping presented in the system. The performance of the backstepping controller depends on its gains or controller parameters. With the use of a simplified mathematical model, the value of control parameters K_1, K_2, and K_3 are tuned manually to prevent the control signal from chattering.

2. Mathematical Model of the Electro-Hydraulic Servo System

A typical electrohydraulic positioning servo system consists of a hydraulic power supply (Fig. 1a, pos. 1) e.g. constant pressure hydraulic pump with a relief valve and accumulator (Fig. 1a, pos. 1), a flow control Servo Valve (SV) (Fig. 1a, pos. 2), a hydraulic cylinder (Fig. 1a, pos. 3), moving mass (Fig. 1a, pos. 4), a position sensor and an electronic control unit. Fig. 1 represents electrohydraulic servo drive (a-schematic representation, b-photo of the testing rig with added mass in moving trolley).

Because designing positioning servo systems are complex spatially for additional boundary conditions, computer simulation of the behavior of such systems is necessary. The analysis of the system dynamics is realized through mathematical modeling and computer simulation of dynamic behavior [20]. The mathematical model is based on physical laws that express the dynamic behavior of the EHS system through differential equations. However, this task is troublesome due to the multidisciplinary nature of the electrohydraulic system that requires electric, magnetic, mechanical, and hydraulic knowledge. Nonlinearities of systems used in EHS drives consist of nonlinearities, which contribute to the dynamical response of the complete system.



a) b)

Fig. 1 a) EHS positioning system b) Electrohydraulic servo system with load

The detailed description of different components used in the EHS system, the authors described in the articles [9-11]; therefore, for giving some explanation only the important facts will be presented.

2.1 Servo valve dynamics

A detailed description of the dynamic response of Servo Valve (SV) can be presented with the use of basic equations based on physics laws (detail description given in [9, 10, 21].

The dynamic behavior of SV involves a large number of parameters. To obtain an optimal solution, the inspection of step responses and frequency diagrams suggests an approximation of a second-order model of the form [22].

$$\frac{1}{\omega_v^2}\ddot{x}_v + \frac{2\xi_v}{\omega_v}\dot{x}_v + x_v + f_{HS}\cdot \mathrm{sgn}\left(\dot{x}_v\right) = k_v \cdot u \tag{1}$$

where: ω_v – valve natural frequency, ξ_v - damping coefficient, x_v–valve spool displacement, f_{HS} – valve hysteresis and response sensitivity, k_v – valve flow gain, u – valve input signal.

Most producers of SV give information on the most needed parameters needed to obtain constants in Eq. (1). Additionally, the limitation of the highest spool speed is recommended for the simulation process.

2.2 Hydraulic actuator dynamics

The hydraulic actuator (cylinder) represents the next important part of the EHS system as the converter of hydraulic power into mechanical power. For calculation, the behavior of pressure in both chambers is needed. Eqs. (2) and (3) give the chamber pressure dynamics by [21], as follows:

$$\dot{p}_A = \frac{\beta_{Aeff}}{V_A}\left[Q_A - A_p\dot{x}_p + Q_{Li} - Q_{LeA}\right] \tag{2}$$

$$\dot{p}_B = \frac{\beta_{Beff}}{V_B}\left[Q_B - A_p\dot{x}_p + Q_{Li} - Q_{LeB}\right] \tag{3}$$

In Eqs. (2) and (3) the effective bulk module is also variable, which depends on the pressure. In literature, there is also given the dependence on temperature, which can be also included in the calculation. The empirical equation for calculation of the effective bulk modulus presented by Lee [21] for hydraulic cylinders is expressed as:

$$\beta_{Aeff} = a_1 \cdot \beta_{\max} \cdot \log\left(a_2\,\frac{p_A}{p_{\max}} + a_3\right) \tag{4}$$

$$\beta_{Beff} = a_1 \cdot \beta_{\max} \cdot \log\left(a_2\,\frac{p_B}{p_{\max}} + a_3\right) \tag{5}$$

with parameters a_1=0.5, a_2=90, a_3=3, β_{max}=1800 MPa, p_{max}=28 MPa.

The cylinder chambers volumes are given by

$$V_A = V_{A0} + \left[A_p \cdot \left(x_0 + x_p \right) \right] \tag{6}$$

$$V_B = V_{B0} + \left[\alpha \cdot A_p \cdot \left(x_0 + x_p \right) \right] \tag{7}$$

where: x_0 - initial piston position, x_p - actual piston position, V_{A0},V_{B0} - the initial chamber volumes. These volumes consist of an efficient part (the volume required to fill only the chambers) and an inefficient part (volume of the pipelines between the valve and actuator) for side *A* and *B*, respectively. A_A= A_p is a piston surface area and the A_B=αA_p rode side area where α =A_B/A_A is a ratio of cylinder bore area and the annulus effective area at the rod side [9, 21].

Furthermore, in Eqs. (2) and (3), Q_{Li} and Q_{Le} denote the internal leakage flow and the external leakage flow, respectively. Leakage from one cylinder chamber to another, known as internal leakage flow, by an assumption of laminar flow can be calculated:

$$Q_{Li} = k_L \left(p_B - p_A \right) \tag{8}$$

where: k_L - the internal leakage flow coefficient. The external leakage can be neglected.

The dynamic subsystem of the piston and the moving mass is described by:

$$m \cdot \ddot{x}_p + F_f\left(\dot{x}_p\right) = \left(p_A - \alpha \cdot p_B \right) \cdot A_p - F_{ext} \tag{9}$$

where: m_t - the total mass, F_f - friction force, F_{ext} - external load force.

Total mass *m* consists of the piston mass and the mass of hydraulic fluid in the cylinder chambers and the pipelines and the mass of the load. However, the mass of fluid can usually be neglected with the piston mass.

In Eq. (9) an important part is given to friction force. Different authors give various solutions for representing the influence of friction. As a compromise for calculation presented equation is used:

$$F_f\left(\dot{x}_p\right) = F_v\left(\dot{x}_p\right) + F_c\left(\dot{x}_p\right) + F_s\left(\dot{x}_p\right) = \sigma_{vf} \cdot \dot{x}_p + F_{co} \cdot \mathrm{sgn}\left(\dot{x}_p\right) + \mathrm{sgn}\left(\dot{x}_p\right) \cdot F_{so} \cdot e^{-\frac{\left|\dot{x}_p\right|}{c_s}} \tag{10}$$

with: σ_{vf} - the parameter for viscous friction, F_{co} - static friction force, F_{so} – Stribeck friction force, c_s – Stribeck velocity. This friction force depends on the velocity sign and is restricted to viscous friction and Columb friction with the Stribeck effect. Moreover, the friction force must be considered to obtain acceptable tracking accuracy.

The natural frequency of components and the whole system is important data in the dynamical analysis. The natural frequency for the overall system equals the natural frequency of the hydro-mechanical part; therefore, the electro-hydraulic control system's natural frequency will be:

$$\omega_H = \sqrt{\frac{C_H}{m}} \tag{11}$$

To calculate the natural frequency of the hydraulic cylinder, the hydraulic cylinder stiffness must be found in advance. The total stiffness of the differential cylinder is defined by the following equation [21]:

$$C_H = \frac{\beta_A \cdot A_A^2}{V_A} + \frac{\beta_B \cdot A_B^2}{V_B} \tag{12}$$

In the case of a synchronous cylinder (double rod of equal area), A_A is equal A_B. For performance calculation, only the minimum stiffness will be considered since it has the worst effect on the system dynamics.

2.3 Verification of mathematical model

For simulation of electrohydraulic servo system (EHS), the Matlab-Simulink software is used. The parameters used in the simulation are given in Tab. 1. External force F_{ext} can be used in the case where the system has additionally load opposite moving direction (in our case is no additional external force present).

Table 1 Main parameters of the EHS system

Parameter	Value	Parameter	Value
A_p	6.4×10^{-4} m^2	m	200 kg
σ_{vf}	70 (N s)/m	V_t	131.85×10^{-6} m^3
F_{co}	19.62 N	β	1.5×10^9 Pa
F_{ext}	0 N	p_s	21 MPa
k_L	3×10^{-13} m^5/Ns	ρ	850 kg/m^3
C_d	0.63	k_V	5.53×10^{-7} m^2/V

The validation of the dynamical model of the EHS system has been proved by earlier experimental testing, where the ramp reference was used (Fig. 2). The error difference between experimental and simulated responses is shown in Fig. 3, where the difference between simulated and experimental is also shown. The mathematical model perfectly emulates the real system dynamics; therefore, the results of computer simulation of the system by using a nonlinear controller are relevant.

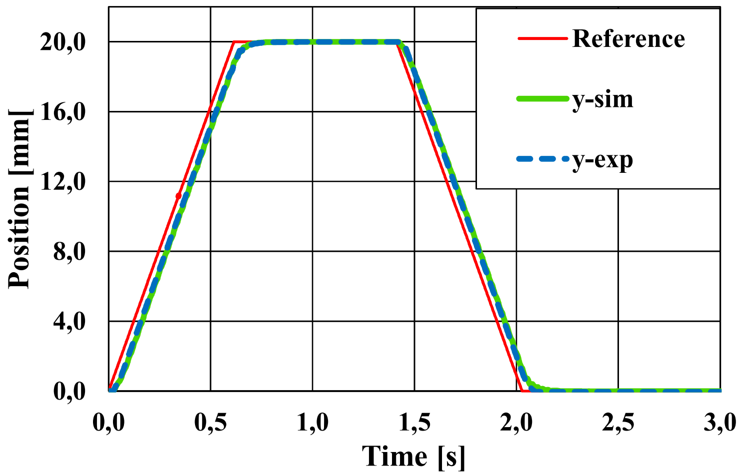


Fig. 2 Comparison of nonlinear mathematical model and the experimental response of the EHS system with ramp response

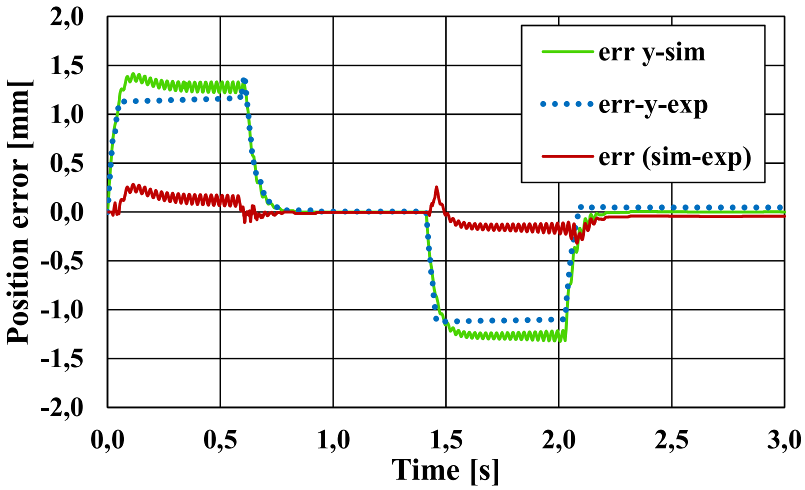


Fig. 3 The error of simulated and experimental response to reference

3. Controller Design

The backstepping controller design in the case of nonlinear dynamical systems is a successful control strategy. Nonlinear systems can be divided into subsystems, whose numbers depend on dynamical model order. Because of the recursive structure, the designer can start the design process at the known-stable system and design," back out", new controllers that progressively stabilize each outer subsystem. The process is finished when the final external control is reached [9, 11].

The mathematical description of a nonlinear system in a new set of coordinates allows us to use the backstepping approach to force it to behave like a linear one [12]. The ability to avoid the cancelation of useful nonlinearities enables a backstepping approach to pursue the objectives of stabilization and tracking rather than that of the linearization method. Therefore, the use of a backstepping controller is possible for solving tracking and regulation problems. For tracking problems, backstepping always uses the error between the actual and desired input to start the design process.

3.1 Model simplification

Due to the complexity of the EHS system, the simplification of the model is preferable. Because of the construction of the system the dynamics of components must be taken into consideration. If we compare the dynamical properties of SV and other parts of the system, we can see that the dynamic of the servo valve with control electronics is much higher than the dynamic of the cylinder and load.

For a hydraulic system with A_p=6.4×10^{-4} m^2, V_t=131.85×10^{-6} m^3 and m=200 kg, the minimum natural frequency is:

$$\omega_{H\min} = \sqrt{\frac{2\beta \cdot A_p^2}{m \cdot V_t}} = 215.866\frac{rad}{s} \rightarrow 34.35Hz \tag{13}$$

Natural frequency can be read out, from the manufacturer's datasheet for SV MOOG 769 [20] we can read out f_v=325Hz.

Because of a very high natural frequency of the servo valve in comparison to hydraulic cylinder ($\omega_v >> \omega_{Hmin}$), taking into account also a dynamic effect of the manifold which decreases the hydraulic natural frequency, the dynamic of SV can be neglected. The servo valve can be described only by the static relationship between the spool position and the valve current input [9, 22]. Moreover, the combined assembly of the servo valve and the electronic amplifier can be described by the equation:

$$x_v = k_v \cdot u \tag{14}$$

where: k_v (m/V) - combined SV and electronic amplifier gain. Simplified system equations take the following form:

$$\begin{aligned}
\dot{x}_p &= x_p \\
\dot{v}_p &= \frac{A_p}{m}\cdot p_L - \frac{1}{m}\left[\sigma_{vf}\cdot\dot{x}_p + F_{co}\cdot\operatorname{sgn}\left(\dot{x}_p\right) + \operatorname{sgn}\left(\dot{x}_p\right)\cdot F_{so}.e^{-\frac{\dot{x}_p}{c_s}}\right] - \frac{F_{ext}}{m} \\
\dot{p}_L &= \frac{4\beta}{V_t}\cdot C_d\cdot k_v\sqrt{\frac{p_s - \operatorname{sgn}(u)\cdot p_L}{\rho}}\cdot u - \frac{4\beta}{V_t}\cdot A_p\cdot\dot{x}_p - \frac{4\beta}{V_t}\cdot k_L\cdot p_L
\end{aligned} \tag{15}$$

where: x_p - actual piston position, v_p - piston velocity, p_L - load pressure.

3.2 Nonlinear controller design

The system state equations can be represented in “strict feedback form”:

$$\begin{aligned} \dot{x}_1 &= x_2 \\ \dot{x}_2 &= a_1 x_3 - a_2 \cdot \text{sgn}(x_2) - a_4 \\ \dot{x}_3 &= a_5 \sqrt{p_s - \text{sgn}(u) \cdot x_3} \cdot u - a_6 x_2 - a_7 x_3 \end{aligned} \tag{16}$$

where: $x_1 = x_p$, $x_2 = v_p$, $x_3 = p_L$, with:

$$a_1 = \frac{A_p}{m},\ a_2 = \frac{\sigma_{vf}}{m},\ a_3 = \frac{F_{co}}{m},\ a_4 = \frac{F_{ext}}{m},\ a_5 = \frac{4\beta}{V_t \cdot \sqrt{\rho}} \cdot C_d \cdot k_v,\ a_6 = \frac{4\beta}{V_t} \cdot A_p,\ a_7 = \frac{4\beta}{V_t} \cdot k_L$$

The designing process of a backstepping controller is the usage of linear terms which corresponds to the Lipschitz inequality given by:

$$\left\| f(x) - f(y) \right\| \le L \left\| x - y \right\| \tag{17}$$

To satisfy the condition in Eq. (17) the substitute of the “*sgn*” function with the continuously differentiable function “*tanh = th*” (hyperbolic tangent) is recommended.

$$\dot{x}_1 = x_2 \tag{18a}$$

$$\dot{x}_2 = a_1 x_3 - a_2 x_2 - a_3 \cdot th(\lambda x_2) - a_4 \tag{18b}$$

$$\dot{x}_3 = a_5 \sqrt{p_s - \text{sgn}(u) \cdot x_3} \cdot u - a_6 x_2 - a_7 x_3 \tag{18c}$$

where λ – free coefficient (selected value of is λ=2).

The control task is to stabilize the plant and track the given reference signal asymptotically. The detailed derivations of finding backstepping control input are covered in the three steps [9, 11].

Step 1

The first step is to find the first error variable and its derivate as shown in Eq. (19) and (20).

$$z_1 = x_1 - r \tag{19}$$

$$\dot{z}_1 = \dot{x}_1 - \dot{r} \tag{20}$$

where: *r(t)* - the reference input , x_1 – position variable.

Defining virtual control α_1 follows from in consideration of Eq. (18a). The difference between actual and virtual control can be expressed with:

$$z_2 = x_2 - \alpha_1 \qquad \rightarrow \qquad x_2 = z_2 + \alpha_1 \tag{21}$$

Now we can define a candidate control of Lyapunov functional for this equation:

$$V_1 = \frac{1}{2} \cdot z_1^2 \tag{22a}$$

$$V_1 = \frac{\rho}{2} \cdot z_1^2 \tag{22b}$$

$$V_1 = \frac{1}{2\rho} \cdot z_1^2 \tag{22c}$$

Authors Rozali [3] and Kadisi [4] proposed an additional constant ρ for fine-tuning needs shown in Eqs. (22b) and (22c). This enables more flexibility in the tuning process, but also cause additional estimation for values in the tuning process. With the conventional method (without added fine-tuned parameter) the results can be achieved successfully in comparison to the classical linear control. The derivative of *V* yields:

$$\dot{V}_1 = (x_1 - r) \cdot z_2 + (x_1 - r) \cdot (\alpha_1 - \dot{r}) \tag{23}$$

Now we can select a virtual control for the first-order system:

$$\alpha_1 = \dot{r} - K_1 z_1 = -K_1 x_1 + K_1 r + \dot{r} \tag{24}$$

where $K_1>0$. Hence, $\dot{V}$ can be rewritten as:

$$\dot{V}_1 = -K_1 z_1^2 + z_1 z_2 \tag{25}$$

Step 2

We use a similar method to achieve Define a second virtual control as given in Eqs. (26), (27), (28), (29) and finally (30).

$$z_3 = x_3 - \alpha_2 \qquad \rightarrow \qquad x_3 = z_3 + \alpha_2 \tag{26}$$

$$\dot{z}_2 = \dot{x}_2 - \dot{\alpha}_1 \tag{27}$$

$$V_2 = V_1 + \frac{1}{2} \cdot z_2^2 \tag{28}$$

$$\dot{V}_2 = \dot{V}_1 + z_2 \cdot \dot{z}_2^2 \tag{29}$$

$$\alpha_2 = \frac{1}{a_1}\left(a_2 x_2 + a_3 \cdot th(\lambda \cdot x_2) + a_4 + \dot{\alpha}_1 - z_1 - K_2 z_2\right) \tag{30}$$

where $K_2>0$, therefore:

$$\dot{V}_2 = -K_1 \cdot z_1^2 - K_2 \cdot z_2^2 + a_1 \cdot z_2 \cdot z_3 \tag{31}$$

Step 3

This step is the final step to become the final control for the system:

$$\dot{z}_3 = \dot{x}_3 - \dot{\alpha}_2 \tag{32}$$

$$V_3 = V_2 + \frac{1}{2} \cdot z_3 \tag{33}$$

$$\dot{V}_3 = \dot{V}_2 + z_3 \cdot \dot{z}_3 \tag{34}$$

$$u = \frac{1}{a_5} - \frac{1}{\sqrt{p_s - x_3}}\left[-a_1 z_2 + a_6 x_2 + a_7 x_3 + \dot{\boldsymbol{\alpha}}_2 - K_3 z_3\right] \tag{35}$$

where $K_3>0$. Then we get:

$$\dot{V}_3 = -K_1 \cdot z_1^2 - K_2 \cdot z_2^2 - K_3 \cdot z_3^2 \tag{36}$$

where K_1, K_2, $K_3>0$. Note that Eq. (28) is the Lyapunov function of the system defined by Eqs. (18a) to (18c) and that the control law is given by Eqs. (24), (30), and (35) render its derivative negative semidefinite [9, 11]

4. Selection of the Design Parameters

Before we start with the selection of the parameters of the backstepping controller the desired dynamic behavior of the system should be discussed first. Different authors proposed different approaches to selecting the proper controller and parameters for it [23, 24]. References of servo-hydraulic drives are mostly: step, ramp, and sinusoidal signals. By use of conventional controllers such as PID controllers, the control parameters should be selected according to the dynamic characteristics of the hydraulic drive. In the case of positioning the system contains an integral dynamic behavior. The tuning method of the PID controller is based on the fact that I part of the controller should be avoided due to stability reasons. Also, the D part of the controller should be avoided due to signal noise and the low damping factor of the hydraulic system. In some special cases, it is also possible to use a lag-lead controller. Generally, only the P part of the PID controller is appropriate for position control.

In most industrial cases the selection of the gain factor of the P controller should follow the demand of the aperiodic step response without any overshoot. To achieve these, only small gain factors of the P controller can be applied.

Similarly, the same restrictions about the dynamic response of the EHS system are taken into account in the case of the Lyapunov-based backstepping controller.

To choose the design parameters of the above-developed Lyapunov-based position tracking controller we should inspect the closed-loop equations again. In our case, we have two virtual control equations $\boldsymbol{\alpha}_1$ (24), $\boldsymbol{\alpha}_2$ (30), with final control law (35) which contains three tuning parameters K_1, K_2, K_3.

K_1, K_2, K_3 were determined by an experimental computer simulation method. In the beginning, we choose a step reference signal with a magnitude of 1mm. This value corresponds to velocity error by the ramp reference signal. Starting point where a selection of equal values of all three parameters $K_1 = K_2 = K_3$.The result is shown in Fig.4.

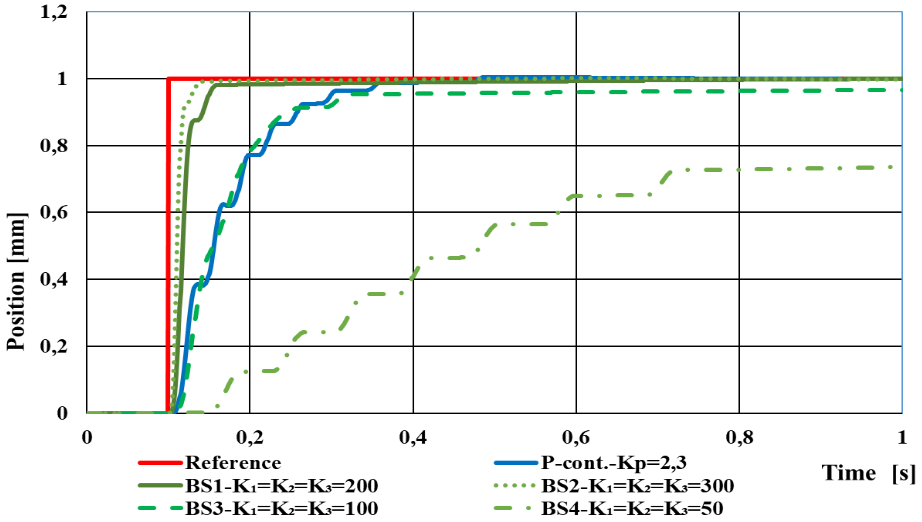


Fig. 4 Step response of EHS system by the backstepping controller

In Fig. 4 P-cont. means P controller with amplification K is equal to Kp. The notification BS-1, BS-2 and BS-3 represents backstepping controllers with different parameters K_1 , K_2 and K_3.

Aiming at achieving a fast response we try with a higher value of K_2, while K_1 and K_3 remain the same as in the previous experiment (Fig. 5). In the next step, we try with different values of K_3 (Fig. 6). Higher values of K_2 increase dynamic response (Fig. 5). Higher values of K_3 increase the damping factor (Fig. 6).

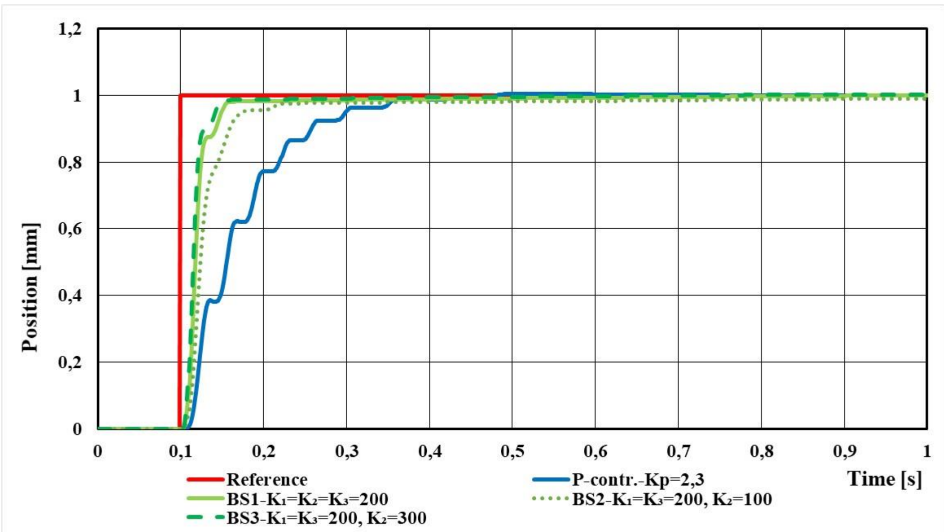


Fig. 5 Step response of EHS system by different values of K_2

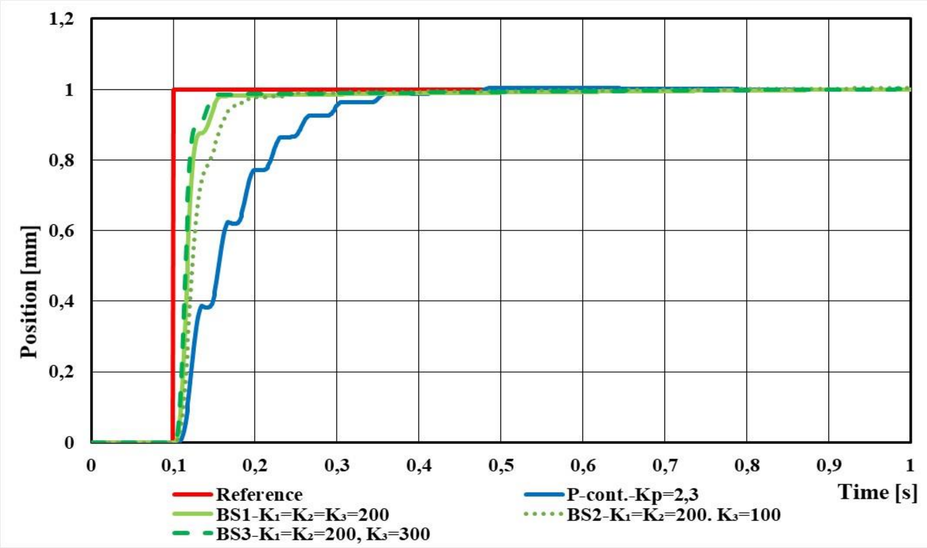


Fig. 6 Step response of EHS system by different values of K_3

Improvement of the previous result can be reached by increasing both K_2 and K_3, see Fig. 7.

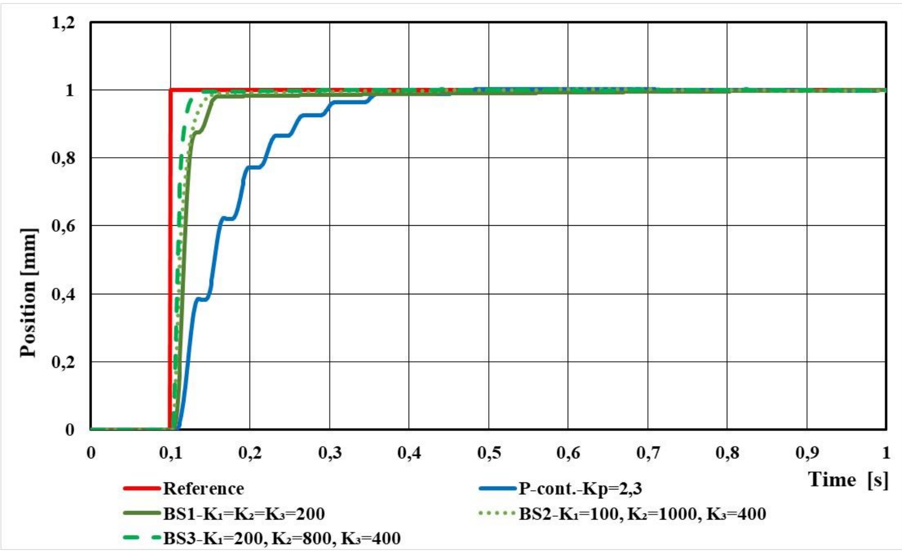


Fig. 7 Step response of EHS system with different values of K_2 and K_3

Finally, we realize the ramp reference response with previous values of K_2 and K_3. The results are shown in Figs. 8 and 9. The differences between time responses by the P–controller and Back-Stepping controller are brightened in the detail.

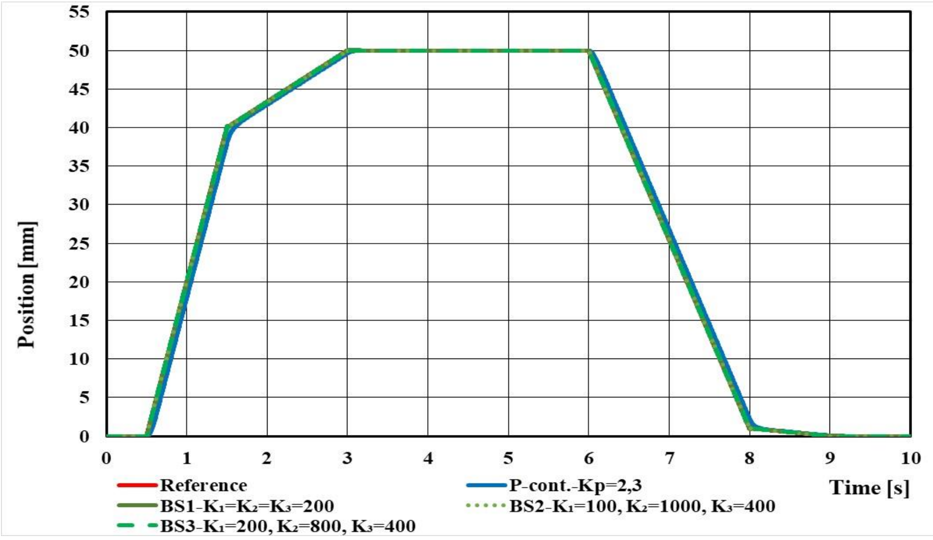


Fig. 8 Ramp response of EHS system with different values of K_2 and K_3

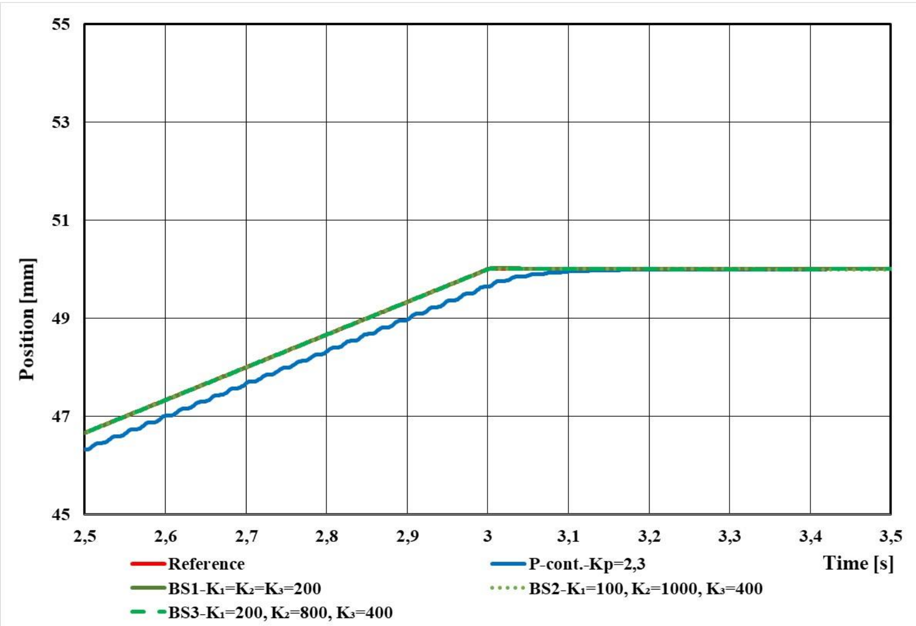


Fig. 9 Detail by approaching the final value of a ramp response of the EHS system with different values of K_2 and K_3

Fig. 10 shows the response of the system on a sinusoidal reference signal. In this case, we use the parameters as were used in previous cases. We can see the influence of nonlinear control in comparison to the P controller.

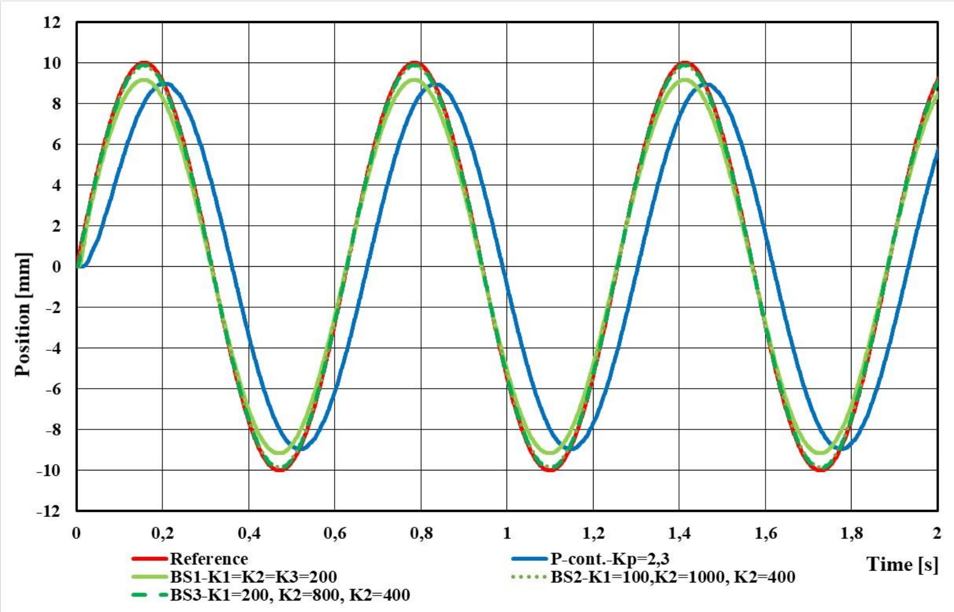


Fig. 10 Sinus reference response of EHS

As shown in Figs. 9 and 10, the behavior of the system controlled with the backstepping controller is significantly better than from linear P controller. We can see that the chosen parameters enabled better dynamical behavior of the system. The parameters with equal values of K ($K_1 = K_2 = K_3$) give a better dynamical response than the P controller.

However, the comparison of both shows significant improvement of system dynamic behavior in the case of backstepping controllers.

5. Conclusion

Electrohydraulic servo actuators will still be the right choice for many application domains, for many years. The actuators play a vital role in maneuvering industrial processes and manufacturing lines. Especially important is successfully closed-loop position, velocity, and force control of such drives. In the last years the clear goal of many researchers and engineers was where to find an improved control strategy. To acquire the highest performance of electro-hydraulic actuators a suitable controller has to be designed.

This paper deals with the development of a Lyapunov-based nonlinear controller based on the integrator backstepping approach. Described procedures are focused on the selection of tuning parameters of such a controller. The proposed approach shows that the selection of appropriate parameters for the backstepping controller can be satisfactorily realized by the computer simulation method. The results of computer simulation are represented and conclusions are explained. The results of computer simulation and the proposed method for select parameters can be used in different reference signals. The proposed method allows setting the best starting parameters for use in the real system. The fact, that computer simulation enables to predict critical operations conditions, allows developers to avoid damage to the system.

Finally, by using these experiences, the electro-hydraulic servo system's performances can be essentially improved and the best position reference tracking performance with small steady-state error can be achieved.

Our future research will include the real-time and practical realization of the above-developed nonlinear controller.

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