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POSITION CONTROL OF AN ELECTRO-HYDRAULIC SERVO SYSTEM USING SLIDING MODE CONTROL ENHANCED BY FUZZY PI CONTROLLER

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Abstract. *Variable structure control with sliding mode, also called sliding mode control, is a particularly suitable as possible control solution in electro-hydraulic systems. In this paper, we apply the technique of the sliding mode control enhanced by fuzzy PI controller to a typical electro-hydraulic system, whose mathematical model accounts for the main nonlinearities including internal friction. The position control problem in the presence of unmodeled dynamics, parametric uncertainties and external disturbances was investigated. Fuzzy controller is added in the feedforward branch of the closed loop to improve the performance of the variable structure controller (with fixed boundary layer) with regard to the position precision and disturbance rejection. The performance improvement is demonstrated through simulation results.*

1. INTRODUCTION

Electro-hydraulic systems play an indispensable role in industrial applications where large inertia and torque loads have to be handled, providing a high degree of both accuracy and performance [12]. The dynamics of these systems are highly nonlinear and their models inevitable contain parametric uncertainties and unmodeled dynamics. In recent articles [2], [11], [18], it is demonstrated that the application of nonlinear robust control techniques is a necessity for successful operation of electro-hydraulic systems.

Among these techniques, variable structure control with a sliding mode (also called sliding mode control) have attracted considerable attention because it provides a systematic approach to the problem of maintaining stability and consistent performance in the face of modelling imprecision and disturbances [5], [17]. However, the discontinuous term in traditional sliding mode control (SMC) causes an effect called chattering which is highly

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undesirable. A well known and simple solution for chattering reduction is boundary layer approach, in which a continuos control action is used in the vicinity of the switching surface. By introducing boundary layer, chattering can be reduced, but tracking performance and robustness are compromised. There are two different approaches proposed in the literature that deal with the problem of tracking error reduction in the presence of boundary layer. One approach is to use the variable boundary layer [8], [17]. The other approach is to use integral action inside the fixed boundary layer [6]. On the other hand, in [16] fuzzy PD controller is added in the feedforward branch, complementary to sliding mode controller, in order to improve the robustness property and performance of the SMC. In the last case, the designs for SMC and fuzzy block are mutually independent.

In this paper, we investigate the position tracking problem of an electro-hydraulic servo system applying a traditional SMC approach [17] in the first step of the controller design. To improve tracking accuracy in the presence of boundary layer, fuzzy inference is employed. In [1], [13], we considered fuzzy tuning of sliding mode controller parameters – the switching gain and the thickness of the boundary layer. Here, a fuzzy PI controller is added in the feedforward branch of the closed-loop, in parallel with the SMC with fixed boundary layer, following the idea proposed in [16].

2. Problem Formulation

The electro-hydraulic servo system which is the object of this study is composed of a double-ended hydraulic cylinder driven by a direct drive servo-proportional valve (Fig. 1). The configuration, a part of simulation model and physical parameters of this system were taken from [11]. The objective is to construct a sliding mode controller with fixed boundary layer to obtain precise position control of a nonlinear electro-hydraulic servo system.

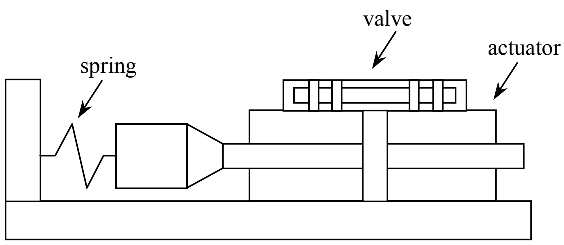


Fig. 1. A schematic diagram of the electro-hydraulic system [11]

In order to represent servo valve dynamics through a wider frequency range, a second order transfer function is used as approximation of the valve dynamics. The relation between the servo valve spool position x_v and the input voltage u can be written as

$$\frac{x_v}{u} = \frac{k_v}{s^2/\omega_v^{\ 2} + 2\xi_v s/\omega_v + 1}, \tag{1}$$

where k_v is the valve gain, ξ_v is the damping ratio of the servo valve and ω_v is the natural frequency of the servo valve.

The differential equations governing the dynamics of the actuator are given in [12].

Defining the load pressure P_L as P_L=P_1-P_2 and the load flow Q_L as Q_L=(Q_1+Q_2)/2, the relationship between the load pressure P_L and the load flow Q_L for an ideal critical servovalve with a matched and symmetric orifice can be expressed as follows:

$$Q_L = C_d w x_v \sqrt{\frac{P_S - \operatorname{sgn}(x_v) P_L}{\rho}}\,, \tag{2}$$

where C_d represents discharge coefficient, w is the spool valve area gradient, P_s is the supply pressure and ρ is the fluid density.

When the continuity equation is applied to the fluid flow, the following expression can be derived:

$$Q_L = A\dot{x} + C_{tp} P_L + \frac{V_t}{4\beta_e}\dot{P}_L\,, \tag{3}$$

where A represents the actuator ram area, x is the actuator piston position, C_{tp} is the total leakage coefficient, V_t is the total actuator volume and β_e is the effective bulk modulus of oil.

The piston force equation is given by

$$P_L A = m\ddot{x} + kx + F_f\,, \tag{4}$$

where m represents mass of the piston and load, k is the spring constant and F_f is the friction force.

The mathematical model given by the equations (1)-(4) represents a base for the simulation model. Nonlinearities in the simulation model, besides mentioned above, include friction, servovalve rate limit, stroke limit and input current saturation.

Since the sliding mode control is robust to unmodeled dynamics a 3rd order simpler model is obtained by combining (1)-(4) and neglecting the valve dynamics (x_v=$k_v u$):

$$\begin{aligned} \dot{x}_1 &= x_2(t) \\ \dot{x}_2 &= x_3(t) \\ \dot{x}_3 &= f(\mathbf{x},t) + b(\mathbf{x})u(t) + d(t) = -\sum_{i=1}^{3} a_i(t)x_i(t) + b(\mathbf{x})u(t) + d(t) \end{aligned} \tag{5}$$

where

$$\begin{aligned} &[x_1\ x_2\ x_3]^T = [x\ \dot{x}\ \ddot{x}]; \\ &a_1 = \frac{4C_{tp}\beta_e}{V_t m}k; \quad a_2 = \frac{k}{m} + \frac{4A^2\beta_e}{V_t m}; \quad a_3 = \frac{4\beta_e C_{tp}}{V_t m}; \\ &b = \frac{4\beta_e A}{V_t m} k_v C_d w \sqrt{\frac{P_s - \operatorname{sgn}(x_v)P_L}{\rho}}; \\ &d(t) = -\frac{4\beta_e C_{tp}}{V_t} F_e - \frac{1}{m}\dot{F}_e. \end{aligned}$$

Friction in the hydraulic cylinder is taken into account as an external disturbance and the dynamics of the servo valve is neglected.

3. INTERNAL FRICTION MODELLING

Due to the tight sealing, hydraulic cylinders feature strong dry friction effect. The behavior of this friction force is rather complex [10], [14]. Friction is usually modeled as a discontinuous static mapping between the velocity and the friction force that depends on the velocity's sign. It is often restricted to the Coulomb and viscous friction components. However, there are several important properties observed in systems with friction which cannot be explained by static models only. This is basically due to the fact that the friction does not have an instantaneous response to a change of velocity, i.e. it possesses an internal dynamics. Examples of these complex properties are: (1) stick-slip motion, characterized by large friction at rest and on low velocities, and small friction during rapid motion; (2) pre-sliding displacement, which shows that the friction behaves like a spring when the applied force is less than the static friction break-away force; (3) friction lag, which means that there is a hysteresis that characterizes the relationship between the friction and velocity.

All these static and dynamic characteristics of the friction are captured by the analytic model of friction dynamics proposed in [19], which is called LuGre model and defined by

$$F_f = \sigma_0 z + \sigma_1 \frac{dz}{dt} + k_v \dot{x}\,, \tag{6}$$

$$\frac{dz}{dt} = \dot{x} - \frac{|\dot{x}|}{g(\dot{x})} z\,, \qquad g(\dot{x}) = \frac{1}{\sigma_0}(F_c + (F_s - F_c)e^{-(\dot{x}/v_s)^2})\,, \tag{7}$$

where $\dot{x}$ (m/s) is the velocity, and $F(N)$ is the friction force described by linear combination of z, dz/dt and viscous friction. Equation (7) represents the dynamics of the friction internal state z and this state is not measurable. The function $g(\dot{x})$ describes part of the "steady-state" characteristics of the model for the constant velocity motions: v_s is the Stribeck velocity, F_s is the static friction, F_c is Coloumb friction, k_v is viscous friction. Thus, the complete friction model is characterized by four static parameters and two dynamic parameters, stiffness coefficient and damping coefficient.

In this paper, the LuGre model is used for friction modelling. Based on the friction model in [11], the parameters for LuGre model were identified. Velocity-friction diagram is shown in Fig. 2.

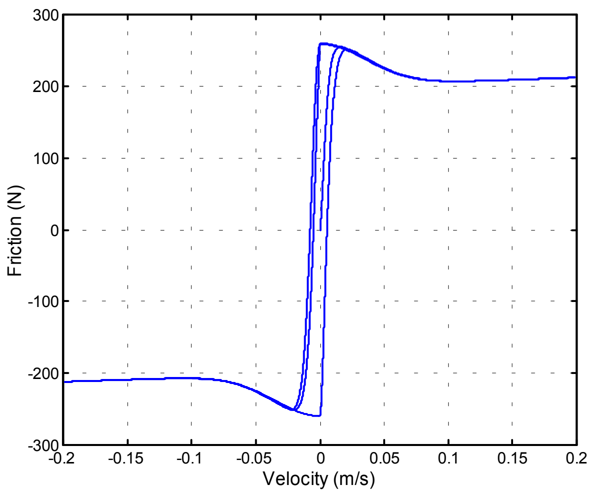


Fig. 2. Friction-velocity description.

From the control point of view the simplest yet most effective approach to counteract the friction phenomenon appears to be the use of sliding mode control; in fact, one of the main characteristic of this control technique is its robustness against bounded uncertainties and disturbances: friction is regarded as a bounded disturbance of unpredictable sign and therefore counteracted by choosing a suitable control amplitude.

4. SLIDING MODE CONTROLLER DESIGN

The sliding mode control method proposed in [17] is used in the first step of the controller design. Consider a nonlinear system described in control canonical form (5). The following assumptions are made:

$$\begin{aligned} & b(\mathbf{x},t) = \hat{b}(\mathbf{x}) + \Delta b(\mathbf{x},t), \\ & | f(\mathbf{x},t) - \hat{f}(\mathbf{x},t) | \leq F(\mathbf{x},t), \\ & d(t) < D. \end{aligned} \tag{7}$$

The nonlinear dynamics $f(\mathbf{x},t)$ are not known exactly, but are estimated as $\hat{f}(\mathbf{x},t)$ with errors bounded by a known function $F(\mathbf{x},t)$. The term $d(t)$ are disturbances, $\hat{f}(\mathbf{x})$, $\hat{b}(\mathbf{x})$ correspond to nominal parameters of the system, and $\Delta b(\mathbf{x},t)$ are parameter uncertainties. The function $\hat{f}(\mathbf{x},t)$ is the estimated system nonlinearity at nominal values for system parameters. The control gain is also unknown exactly. It is bounded as

$$0 < b_{\min} \leq b \leq b_{\max} . \tag{8}$$

The control gain b and its bounds can be time varying or state dependent. Since the control input is multiplied by the control gain in the dynamics, the geometric mean of the lower and upper bounds of the gain is taken as the estimate of b:

$$\hat{b} = (b_{\min} b_{\max})^{1/2} . \tag{9}$$

Denote by $x_{1d}(t),\ x_{2d}(t) = \dot{x}_{1d}(t),\ x_{3d} = \ddot{x}_{1d}(t)$ desired system states, and assume that they are bounded. The control objective is to design a chattering attenuated sliding mode controller that provides robust performance in the presence of uncertainties, given the bounds $\Delta b(x,t)$ and $F(x,t)$. To this end, the following switching function is defined:

$$S(\mathbf{e},t) = (d/dt + \lambda)^2 e_1 = \lambda^2 e_1 + 2\lambda e_2 + e_3 , \tag{10}$$

where

$$\begin{aligned} & \mathbf{e} = [e_1\ e_2\ e_3]^T \\ & e_1(t) = x_1(t) - x_{1d}(t) \\ & e_2(t) = x_2(t) - x_{2d}(t) = \dot{x}_1(t) - \dot{x}_{1d}(t) \\ & e_3(t) = x_3(t) - x_{3d}(t) = \ddot{x}_1(t) - \ddot{x}_{1d}(t) \end{aligned} \tag{11}$$

are the tracking errors and λ is strictly positive constant to be specified according to the desired dynamics of the closed-loop system.

The equivalent control u_{eq} is determined by the necessary condition for existence of a sliding mode $\dot{S}=0$ with nominal values, i.e. for the system without uncertainties and disturbances. This results in

$$u_{eq}=\hat{b}^{-1}(-\lambda^2 e_2-2\lambda e_3-\hat{f}(x,t)+\ddot{x}_{1d})\,. \tag{12}$$

The control law must satisfy a sliding condition regardless the estimation errors. Thus, a discontinuous term is added to (12) and total control law is as follows

$$u=u_{eq}+u_{sw}\,, \tag{13}$$

where

$$u_{sw}=-\hat{b}^{-1}K\,\mathrm{sgn}(S)\,. \tag{14}$$

In essence, to achieve a zero tracking error all system trajectories must be forced to converge to S in finite time and to remain on S afterwards. By choosing the Lyapunov function V=0.5S^2, a reaching condition is obtained as:

$$\dot{V}=0.5\frac{d}{dt}S^2(\mathbf{e},t)\le-\eta|S(\mathbf{e},t)|\,, \tag{15}$$

where η is strictly positive design parameter. This leads to

$$S(\mathbf{e},t)\dot{S}(\mathbf{e},t)\le-\eta|S(\mathbf{e},t)|\,. \tag{16}$$

Discontinuous gain K in (14) is determined such that the above condition is met. The final condition for K is:

$$K\ge\beta(F+\eta)+(\beta-1)\hat{b}|U|+D\,, \tag{17}$$

where

$$|d|\le D,\ |u_{eq}|\le U,\ |f-\hat{f}|\le F\,, \tag{18}$$

and gain margin β is introduced as

$$\begin{aligned}&\beta=\sqrt{b_{\max}/b_{\min}}\ge 1,\\ &b^{-1}\hat{b}\le\beta.\end{aligned} \tag{19}$$

Assuming that the coefficients of state equations (5) may fluctuate by 50% around their nominal values, the bounds of the uncertainties and the estimation error were determined:

$$\beta=\sqrt{b_{\max}/b_{\min}}=\sqrt{(\hat{b}\cdot 1.5)/(\hat{b}\cdot 0.5)}=\sqrt{3};\quad F=|f-\hat{f}|=0.5|\hat{f}|.$$

In conclusion, if the switching gain K satisfies (17), the reaching condition will hold. This means that in a finite time, the tracking will be achieved, within specified error bounds F and β.

Discontinuity in the switching control (14) usually results in chattering which may excite undesirable high-frequency or unmodelled dynamics. The simple and wide-used method to alleviate chattering is to replace a sign function in discontinuous term in the control law by a saturating continuos approximation that introduces the concept of boundary layer:

$$u_{sw} = -\hat{b}^{-1} K \tanh(S/\phi)\,, \tag{20}$$

where $\phi>0$ is the thickness of the boundary layer and its value is to be adjusted to achieve an optimal balance between the position error and the level of control chattering.

5. FUZZY CONTROLLER DESIGN

To reduce the tracking error in the case of sliding mode controller with fixed boundary layer given in the previous section, the introduction of fuzzy controller is considered, following the idea [16]. However, unlike this solution, a fuzzy PI controller of incremental type is used. The motivation for choosing this type of fuzzy controller is to alleviate the problems in calculation the switching gain, and to improve operating efficiency in the presence of additional external disturbance that is not taken into account in the design for SMC.

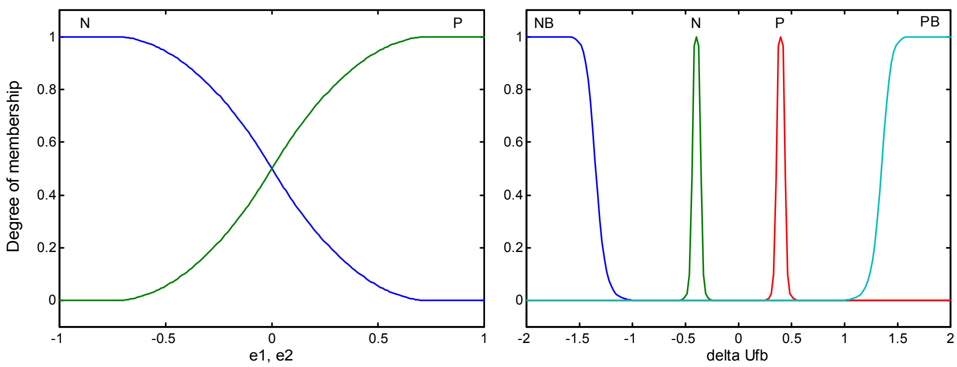
The fuzzy controller acts complementary to the SMC controller and total control low is

$$u = u_{eq} + u_{sw} + u_{fb}\,, \tag{21}$$

where u_{eq} and u_{sw} are given by (12) and (20), respectively. The term u_{fb} in (21) is the output of the fuzzy PI controller whose design procedure is explained in the following.

The inputs to the fuzzy controller are the position tracking error e_1 and the velocity tracking error e_2. The output of the fuzzy controller is the control signal increment. Fuzzy rules are obtained based on the following reasoning: if the position tracking error is Negative and if the velocity tracking error is Negative then the control signal is to be increased for Positive Big value.

Number of fuzzy values of fuzzy controller inputs, shape and location of membership functions, and inference mechanism (max-min method, centre of gravity as defuzzification method) are the same or similar as in [16]. Fuzzy values of controller inputs and controller output are shown in Fig. 3. The type of fuzzy controller and the rule base are essentially different, according to the role of fuzzy controller in this problem.



a) b)

Fig. 3. Fuzzy values of fuzzy controller inputs (a) and output (b).

The set of fuzzy rules is as follows:

1. If *error* is N and *change in error* is N then *output* is PB
2. If *error* is N and *change in error* is P then *output* is P
3. If *error* is P and *change in error* is N then *output* is N
4. If *error* is P and *change in error* is P then *output* is NB

The input-output mapping which is the result of inference by the fuzzy rules is shown in Fig. 4.

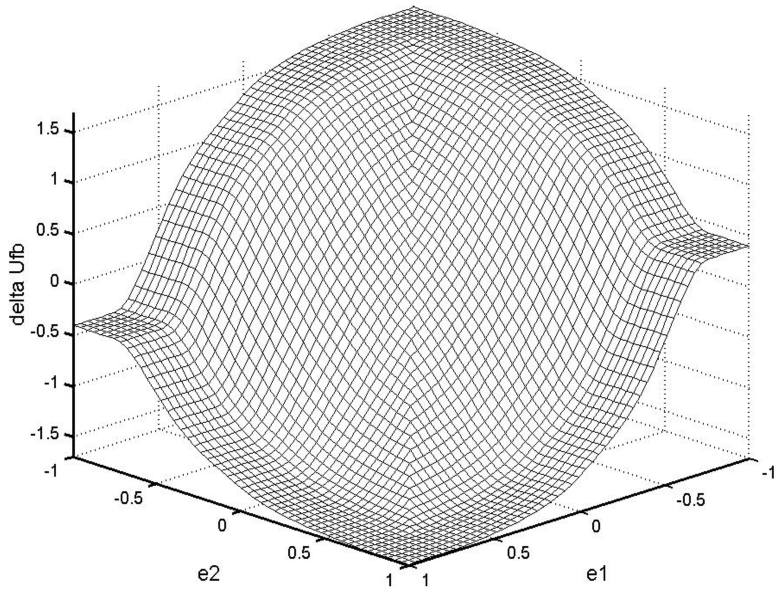


Fig. 4. Input-output mapping of the fuzzy PI controller

6. SIMULATION RESULTS

For high performance, it is necessary to incorporate as much prior information on the desired trajectory into the controller design as possible. In the particular case, the sliding mode controller makes use of the third derivative of the desired trajectory. In order to provide a feasible reference motion trajectory that the actuator can track and to avoid overshoots, the maximum values of desired velocity and acceleration were determined on the ground of mathematical and simulation model of the electro-hydraulic servo system, taking into account the maximum value of the control voltage u_{max}=10V. Then, a continuously differentiable velocity profile was determined. The reference trajectory, obtained in such a way, is shown in Fig. 5.

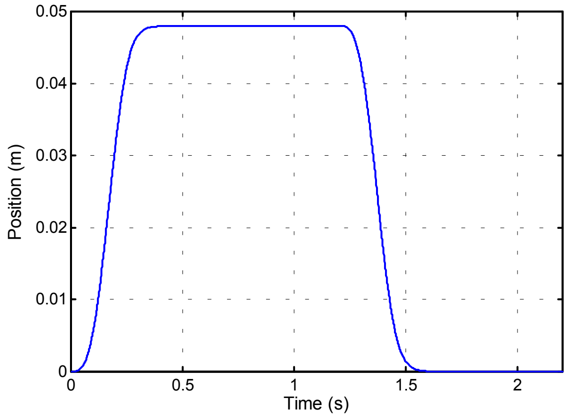


Fig. 5. Reference trajectory

Sliding mode controller with a fixed boundary layer (control law defined by (13), (12) and (20)) was tested for different values of velocity and acceleration. In the simulation, the following values for the controller parameters were used: K=3500, λ=90, φ=3.5. Simulation results showed satisfactory performance throughout the range of the cylinder stroke. Note that the constant value for the switching gain K was used, assuming that the condition (17) was satisfied. Physical parameters for the electro-hydraulic servo system are given in Table 1.

Fig. 6 shows the tracking error for SMC+FB (SMC plus fuzzy controller, control law defined by (21), solid line) and for SMC (sliding mode controller, dashed line). The scaling factors for fuzzy input variables and fuzzy output variable are selected as Ne_1=1667, Ne_2=50, $N_{\Delta u}$=0.008. By adding fuzzy controller, steady state error was reduced by factor bigger than 4. Note that in the case of SMC, the cylinder follows the reference without overshoot, no matter of direction, which is not the case of SMC+FB.

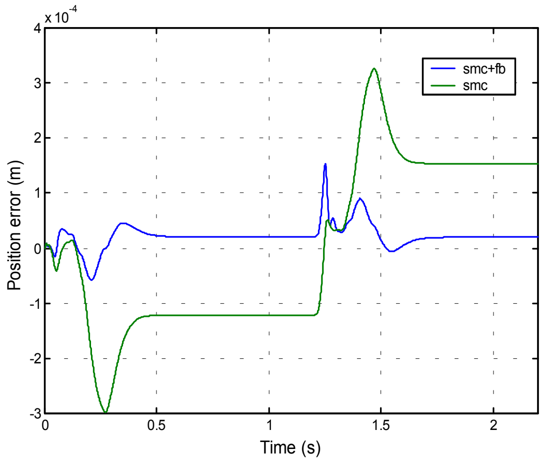


Fig. 6. Tracking error (nominal parameters, without additional disturbance)

In Fig. 7 the values of control signals are depicted. Fig. 8 shows the chosen sliding function dynamics with and without the fuzzy control. In both cases, the system state remains inside the boundary layer, starting from **e**=[0 0 0]^T.

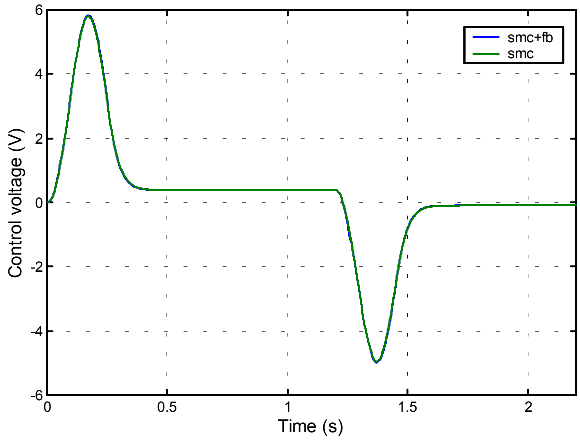


Fig. 7. Control signal (nominal parameters, without additional disturbance)

To examine the performance of SMC+FB under conditions when SMC does not follow the reference trajectory, an additional disturbance of step type was involved, starting from t=0.7 s. This additional disturbance was chosen as a force F_d=1 kN. Fig. 9 shows the system response in the presence of this disturbance. It is to emphasize that the purpose of this figure is not to compare SMC with and without the fuzzy controller, than to illustrate that the magnitude of disturbance is such that SMC cannot fulfill the given task.

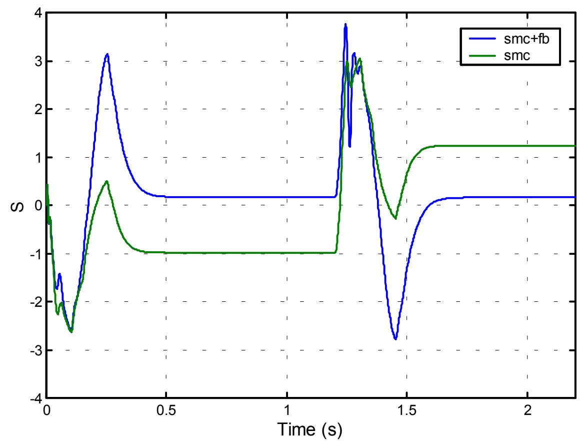


Fig. 8. Sliding surface (nominal parameters, without additional disturbance)

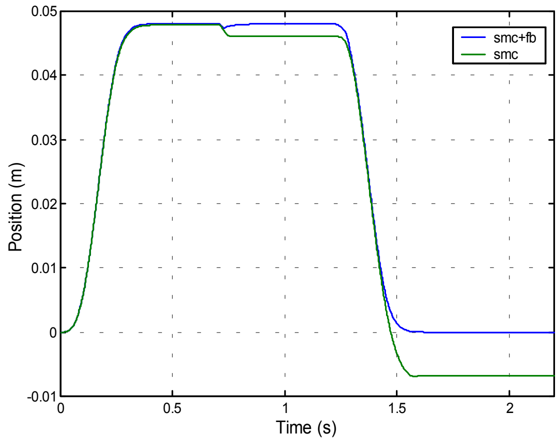


Fig. 9. System response in the presence of additional external disturbance

The tracking error of SMC+FB is shown in Fig. 10. It is clear that complete elimination of the additional disturbance was achieved in approximately 0.3 s. Furthermore, steady state error is the same as without additional disturbance.

From Fig. 11, the difference in control signal as a consequence of adding the fuzzy controller is clearly seen. The sliding function dynamics for this case is depicted in Fig. 12.

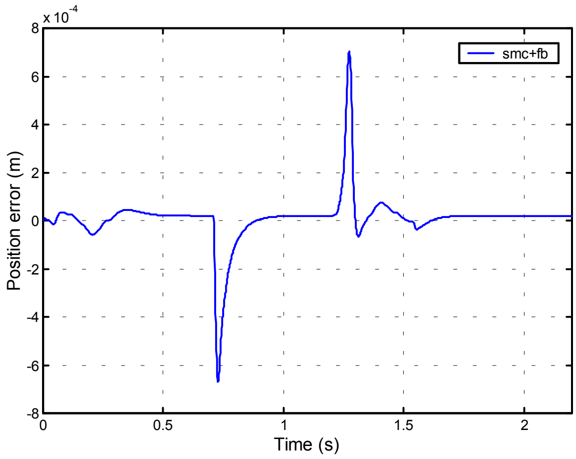


Fig. 10. Tracking error of SMC+FB in the presence of additional disturbance

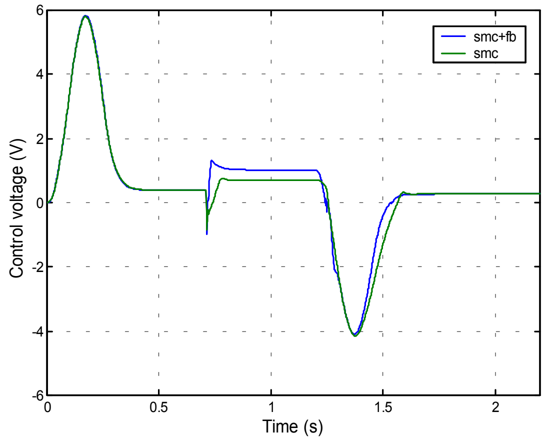


Fig. 11. Control signal of SMC+FB in the presence of additional disturbance

Type and magnitude of additional disturbance is explained as follows. First, the magnitude of disturbance was increased gradually in order to examine the system response when the system state is out of the boundary layer and SMC still fulfils the given tasks: it was noted that SMC with fuzzy controller force the state to go back into the boundary layer faster. In that sense, the role of SMC+FB is equivalent to the role of SMC with fuzzy tuning of the switching gain [1]. Second, magnitude of external disturbance was set on the value when SMC does not follow the reference trajectory.

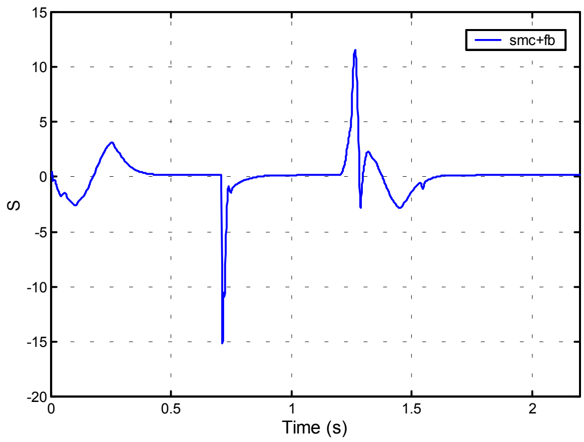


Fig. 12. Sliding surface (nominal parameters, with additional disturbance)

Table 1. Physical parameters for the electro-hydraulic system

Name	Symbol	Nominal Value	Unit
Supply pressure	P_s	1.034×10^{7}	Pa
Total actuator volume	V_t	6.535×10^{-5}	m^3
Effective bulk modulus	β_e	10^9	Pa
Actuator ram area	A	3.2673×10^{-4}	m^2
Total leakage coefficient	C_{tp}	2×10^{-12}	$m^3/(s\ Pa)$
Discharge coefficient	C_d	0.6	
Spool valve area gradient	w	0.022	m
Fluid mass density	ρ	840	kg/m^3
Mass of actuator and load	m	24	kg
Spring constant	k	16010	N/m
Static friction	F_s	260	N
Coulomb friction	F_c	200	N
Viscous friction	F_{visc}	60	N/(m/s)
Stiffness coefficient	σ_0	12×10^{5}	m/s
Damping coefficient	σ_{1e}	300	N s/m
Stribeck velocity	v_s	0.1	m/s

7. Conclusion

In the work reported here, we investigated the performance improvement of variable structure control system for position tracking of an electro-hydraulic servo system. The dynamic friction model is incorporated in the simulation model instead of the original one. Fuzzy PI controller of incremental type was added to act complementary to the SMC and to play the secondary role. In this way, the problems of variable switching gain and variable boundary layer were treated indirectly. Simulation experiments revealed that this combined controller performs much better than the SMC controller with regard to the position precision and disturbance rejection.

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UPRAVLJANJE POZICIJOM ELEKTROHIDRAULIČKOG SISTEMA PRIMENOM REGULATORA PROMENLJIVE STRUKTURE PROŠIRENOG FAZI PI REGULATOROM

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U ovom radu je primenjen regulator promenljive strukture proširen fazi PI regulatorom na tipični elektrohidraulički sistem čiji matematički model obuhvata glavne nelinearnosti, uključujući trenje u hidrauličkom cilindru. Razmatran je problem upravljanja pozicijom u prisustvu nemodelirane dinamike, parametarskih neodređenosti i spoljašnjih poremećaja. Uvođenjem fazi regulatora u direktnu granu zatvorenog kola poboljšane su performanse sistema upravljanja promenljive strukture sa fiksnim graničnim slojem u odnosu na tačnost praćenja zadate trajektorije i eliminaciju spoljašnjeg poremećaja. Poboljšanje performansi je ilustrovano rezultatima simulacije.