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FEEDBACK LINEARIZATION WITH FUZZY COMPENSATION FOR ELECTRO-HYDRAULIC ACTUATED SYSTEMS

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Abstract. *Electro-hydraulic servo-systems are widely employed in industrial applications such as robotic manipulators, active suspensions, precision machine tools and aerospace systems. They provide many advantages over electric motors, including high force to weight ratio, fast response time and compact size. However, precise control of electro-hydraulic systems, due to their inherent nonlinear characteristics, cannot be easily obtained with conventional linear controllers. Most flow control valves can also exhibit some hard nonlinearities such as dead-zone due to valve spool overlap. This work describes the development of a nonlinear controller based on the feedback linearization method and including a fuzzy compensation scheme for an electro-hydraulic actuated system with unknown dead-band. Numerical results are presented in order to demonstrate the control system performance.*

Keywords: *Dead-zone, Electro-hydraulic systems, Feedback Linearization, Fuzzy logic, Nonlinear control.*

1. INTRODUCTION

Electro-hydraulic actuators play an essential role in several branches of industrial activity and are frequently the most suitable choice for systems that require large forces at high speeds. Their application scope ranges from robotic manipulators to aerospace systems. Another great advantage of hydraulic systems is the ability to keep up the load capacity, which in the case of electric actuators is limited due to excessive heat generation.

However, the dynamic behavior of electro-hydraulic systems is highly nonlinear, which in fact makes the design of controllers for such systems a challenge for the conventional and well established linear control methodologies. The increasing number of works dealing with control approaches based on modern techniques shows the great interest of the engineering community, both in academia and industry, in this particular field. The most common approaches are the adaptive (Guan and Pan, 2008b,a; Yanada and Furuta, 2007; Yao *et al.*, 2000) and variable structure (Bessa *et al.*, 2010c; Mihajlov *et al.*, 2002; Bonchis *et al.*, 2001; Liu and Handroos, 1999) methodologies, but nonlinear controllers based on quantitative feedback theory (Niksefat and Sepehri, 2000), optimal tuning PID (Liu and Daley, 2000), adaptive neural network (Knohl and Unbehauen, 2000) and adaptive fuzzy system (Bessa *et al.*, 2010a) were also presented.

In addition to the common nonlinearities that originate from the compressibility of the hydraulic fluid and valve flow-pressure properties, most electro-hydraulic systems are also subjected to hard nonlinearities such as dead-zone due to valve spool overlap. It is well-known that the presence of a dead-zone can lead to performance degradation of the controller and limit cycles or even instability in the closed-loop system. To overcome the negative effects of the dead-zone nonlinearity, many works (Tao and Kokotović, 1994; Kim *et al.*, 1994; Oh and Park, 1998; Šelmić and Lewis, 2000; Tsai and Chuang, 2004; Zhou *et al.*, 2006) use an inverse function even though this approach leads to a discontinuous control law and requires instantaneous switching, which in practice can not be accomplished with mechanical actuators. An alternative scheme, without using the dead-zone inverse, was originally proposed by Lewis *et al.* (1999) and also adopted by Wang *et al.* (2004). In both works, the dead-zone is treated as a combination of a linear and a saturation function. This approach was further extended by Ibrir *et al.* (2007) and by Zhang and Ge (2007), in order to accommodate non-symmetric and unknown dead-zones, respectively.

In this work, a nonlinear controller is developed for an electro-hydraulic system subject to a dead-zone input. The adopted approach is based on the feedback linearization method, but enhanced by a fuzzy inference system in order to compensate for the undesired dead-zone effects and other uncertainties. Numerical simulations are carried out in order to demonstrate the improved performance of the proposed control scheme.

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2. FEEDBACK LINEARIZATION

Due to its simplicity, feedback linearization scheme has been largely applied in industrial control systems, specially in the field of industrial robotics. The main idea behind this control method is the development of control law that allows the transformation of the original dynamical system into an equivalent but simpler one (Slotine and Li, 1991).

Consider a class of n^{th}-order nonlinear systems:

$$x^{(n)} = f(\mathbf{x}, t) + b(\mathbf{x}, t)u \tag{1}$$

where u is the control input, the scalar variable x is the output of interest, $x^{(n)}$ is the n-th time derivative of x, $\mathbf{x} = [x, \dot{x}, \ldots, x^{(n-1)}]$ is the system state vector and $f, b : \mathbb{R}^n \to \mathbb{R}$ are both nonlinear functions.

Let us now define an appropriate control law that ensures the tracking of a desired trajectory $\mathbf{x}_d = [x_d, \dot{x}_d, \ldots, x_d^{(n-1)}]$, *i.e.* the controller should assure that $\tilde{\mathbf{x}} \to 0$ as $t \to \infty$, where $\tilde{\mathbf{x}} = \mathbf{x} - \mathbf{x}_d = [\tilde{x}, \dot{\tilde{x}}, \ldots, \tilde{x}^{(n-1)}]$ is the related tracking error. On this basis, assuming that the state vector $\mathbf{x}$ is available to be measured and the functions f and g are well known, with $|b(\mathbf{x})| \geq 0$, the following control law:

$$u = b^{-1}(-f + x_d^{(n)} - k_0\tilde{x} - k_1\dot{\tilde{x}} - \cdots - k_{n-1}\tilde{x}^{(n-1)}) \tag{2}$$

guarantees that $\mathbf{x} \to \mathbf{x}_d$ as $t \to \infty$, if the coefficients k_i $(i = 0, 2, \ldots, n-1)$ make the polynomial $p^n + k_{n-1}p^{n-1} + \cdots + k_0$ a Hurwitz polynomial.

The convergence of the closed-loop system could be easily established by substituting the control law, Eq. (2), in the nonlinear system, Eq. (1). The resulting dynamical system could be rewritten by means of the tracking error:

$$\tilde{x}^{(n)} + k_{n-1}\tilde{x}^{(n-1)} + \ldots + k_1\dot{\tilde{x}} + k_0\tilde{x} = 0 \tag{3}$$

where the related characteristic polynomial is Hurwitz, which ensure the exponential convergence of the tracking error to zero.

Although feedback linearization represents a very simple approach, an important drawback is the requirement of a perfectly known dynamical system, in order to ensure the exponential convergence.

3. ELECTRO-HYDRAULIC SYSTEM MODEL

In order to design the adaptive fuzzy controller, a mathematical model that represents the hydraulic system dynamics is needed. Dynamic models for such systems are well documented in the literature (Merritt, 1967; Walters, 1967).

The electro-hydraulic system considered in this work consists of a four-way proportional valve, a hydraulic cylinder and variable load force. The variable load force is represented by a mass–spring–damper system. The schematic diagram of the system under study is presented in Fig. 1.

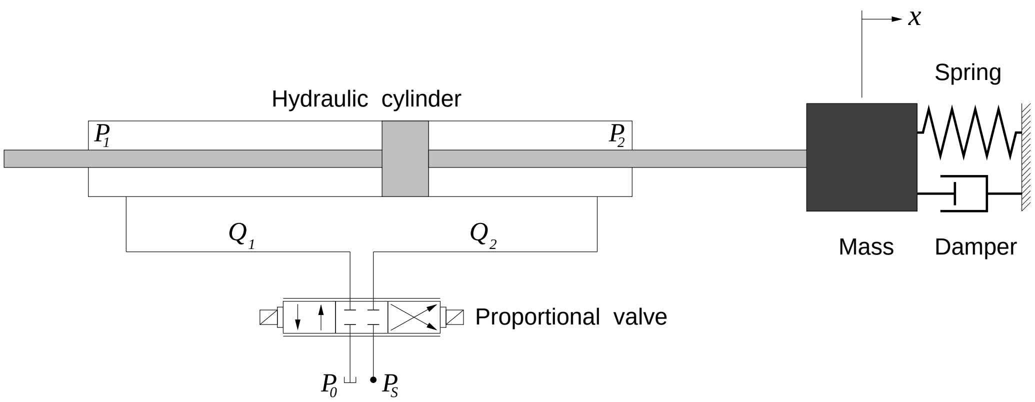

Figure 1. Schematic diagram of the electro-hydraulic servo-system.

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The balance of forces on the piston leads to the following equation of motion:

$$F_g = A_1P_1 - A_2P_2 = M_t\ddot{x} + B_t\dot{x} + K_sx \tag{4}$$

where F_g is the force generated by the piston, P_1 and P_2 are the pressures at each side of cylinder chamber, A_1 and A_2 are the ram areas of the two chambers, M_t is the total mass of piston and load referred to piston, B_t is the viscous damping coefficient of piston and load, K_s is the load spring constant and x is the piston displacement.

Defining the pressure drop across the load as $P_l = P_1 - P_2$ and considering that for a symmetrical cylinder $A_p = A_1 = A_2$, Eq. (4) can be rewritten as

$$M_t\ddot{x} + B_t\dot{x} + K_sx = A_pP_l \tag{5}$$

Applying continuity equation to the fluid flow, the following equation is obtained:

$$Q_l = A_p\dot{x} + C_{tp} + \frac{V_t}{4\beta_e}\dot{P}_l \tag{6}$$

where $Q_l = (Q_1 + Q_2)/2$ is the load flow, C_{tp} the total leakage coefficient of piston, V_t the total volume under compression in both chambers and β_e the effective bulk modulus.

Considering that the return line pressure is usually much smaller than the other pressures involved ($P_0 \approx 0$) and assuming a closed center spool valve with matched and symmetrical orifices, the relationship between load pressure P_l and load flow Q_l can be described as follows

$$Q_l = C_dw\bar{x}_{sp}\sqrt{\frac{1}{\rho}\big(P_s - \mathrm{sgn}(\bar{x}_{sp})P_l\big)} \tag{7}$$

where C_d is the discharge coefficient, w the valve orifice area gradient, $\bar{x}_{sp}$ the effective spool displacement from neutral, ρ the hydraulic fluid density, P_s the supply pressure and $\mathrm{sgn}(\cdot)$ is defined by

$$\mathrm{sgn}(z) = \begin{cases} -1 & \text{if} \quad z < 0 \\ 0 & \text{if} \quad z = 0 \\ 1 & \text{if} \quad z > 0 \end{cases} \tag{8}$$

Assuming that the dynamics of the valve are fast enough to be neglected, the valve spool displacement can be considered as proportional to the control voltage (u). For closed center valves, or even in the case of the so-called critical valves, the spool presents some overlap. This overlap prevents from leakage losses but leads to a dead-zone nonlinearity within the control voltage, as shown in Fig. 2.

The dead-zone nonlinearity presented in Fig. 2 can be mathematically described by:

$$\bar{x}_{sp}(t) = \begin{cases} k_v\big(u(t) - \delta_l\big) & \text{if} \quad u(t) \le \delta_l \\ 0 & \text{if} \quad \delta_l < u(t) < \delta_r \\ k_v\big(u(t) - \delta_r\big) & \text{if} \quad u(t) \ge \delta_r \end{cases} \tag{9}$$

where k_v is the valve gain and the parameters δ_l and δ_r depends on the size of the overlap region.

For control purposes, as shown by Bessa *et al.* (2010a), Eq. (9) can be rewritten in a more appropriate form:

$$\bar{x}_{sp}(t) = k_v[u(t) - d] \tag{10}$$

where $d(u)$ can be obtained from Eq. (9) and Eq. (10):

$$d = \begin{cases} \delta_l & \text{if} \quad u(t) \le \delta_l \\ u(t) & \text{if} \quad \delta_l < u(t) < \delta_r \\ \delta_r & \text{if} \quad u(t) \ge \delta_r \end{cases} \tag{11}$$

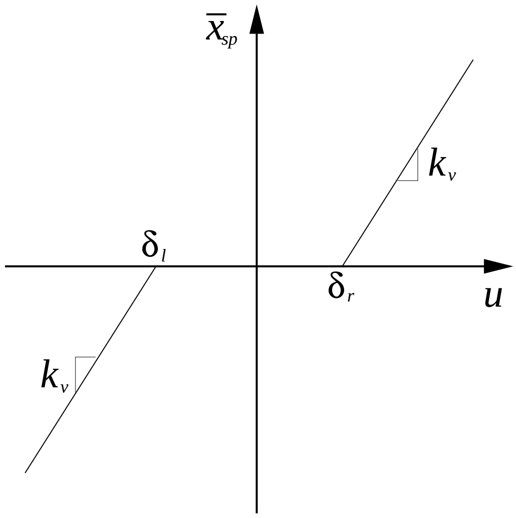


Figure 2. Dead-zone nonlinearity.

Combining equations (5), (6), (7), (10) and (11) leads to a third-order differential equation that represents the dynamic behavior of the electro-hydraulic system:

$$\dddot{x} = -\mathbf{a}^{\mathrm{T}}\mathbf{x} + bu - bd \tag{12}$$

where $\mathbf{x} = [x, \dot{x}, \ddot{x}]$ is the state vector with an associated coefficient vector $\mathbf{a} = [a_0, a_1, a_2]$ defined according to

$$a_0 = \frac{4\beta_e C_{tp} K_s}{V_t M_t} \tag{13}$$

$$a_1 = \frac{K_s}{M_t} + \frac{4\beta_e A_p^2}{V_t M_t} + \frac{4\beta_e C_{tp} B_t}{V_t M_t} \tag{14}$$

$$a_2 = \frac{B_t}{M_t} + \frac{4\beta_e C_{tp}}{V_t} \tag{15}$$

The control gain b should be defined as

$$b = \frac{4\beta_e A_p}{V_t M_t} C_d w k_v \sqrt{\frac{1}{\rho}\left[P_s - \mathrm{sgn}(u)\big(M_t\ddot{x} + B_t\dot{x} + K_s x\big)/A_p\right]} \tag{16}$$

Based on the dynamic model presented in Eq. (12), a nonlinear controller with a fuzzy compensation scheme will be developed in the next section.

4. FEEDBACK LINEARIZATION WITH A FUZZY COMPENSATION SCHEME

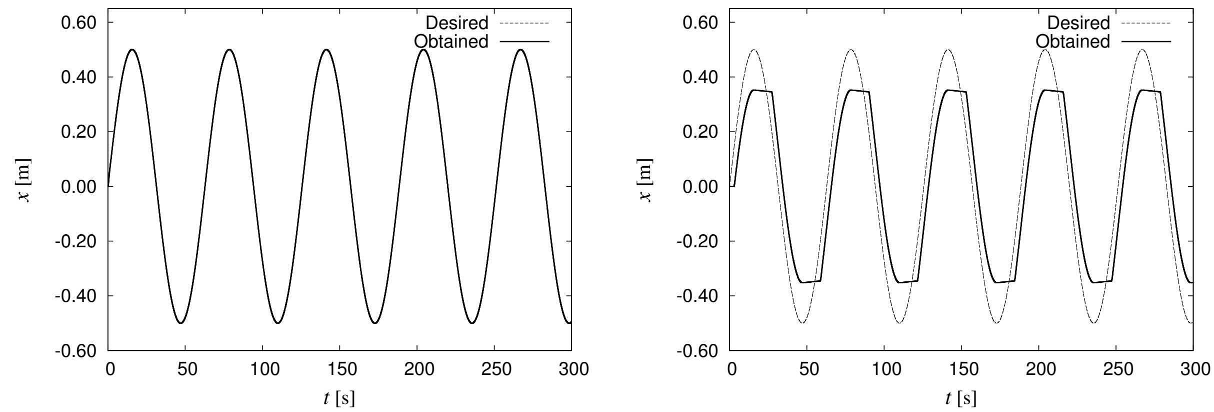
Consider the trajectory tracking problem and let $\tilde{\mathbf{x}} = \mathbf{x} - \mathbf{x}_d = [\tilde{x}, \dot{\tilde{x}}, \ddot{\tilde{x}}, \dddot{\tilde{x}}]$ be the tracking error associated to a desired trajectory $\mathbf{x}_d = [x_d, \dot{x}_d, \ddot{x}_d, \dddot{x}_d]$.

Now, defining a control law according to Eq. (2):

$$u = b^{-1}(\mathbf{a}^{\mathrm{T}}\mathbf{x} + \dddot{x}_d - 3\lambda\ddot{\tilde{x}} - 3\lambda^2\dot{\tilde{x}} - \lambda^3\tilde{x}) + d \tag{17}$$

where λ is a strictly positive constant.

It must be emphasized that, since the dead-band parameters δ_l and δ_r are unknown, the dynamical model is not perfectly known and, in this way, the proposed control law is not sufficient to guarantee the exponential convergence of the tracking error to zero. Figure 3 shows the effect of the unknown dead-zone on the trajectory tracking obtained with Eq. (12) and Eq. (17) for $x_d = 0.5\sin(0.1t)$ m. The simulation studies were performed with a numerical implementation, in C, with sampling rates of 500 Hz for control system and 1 kHz for dynamic model. The adopted parameters for the electro-hydraulic systems were $P_s = 7$ MPa, $\rho = 850$ kg/m^3, $C_d = 0.6$, $w = 2.5 \times 10^{-2}$ m, $A_p = 3 \times 10^{-4}$ m^2, $C_{tp} = 2 \times 10^{-12}$ m^3/(s Pa), $\beta_e = 700$ MPa, $V_t = 6 \times 10^{-5}$ m^3, $M_t = 250$ kg, $B_t = 100$ Ns/m, $K_s = 75$ N/m, $\delta_l = -0.5$ V and $\delta_r = 0.5$ V. The parameter λ of the controller was defined as $\lambda = 8$.



(a) Without dead-zone input.

(b) With dead-zone input.

Figure 3. Tracking performance with feedback linearization and $x_d = 0.5\sin(0.1t)$ m.

As observed in Fig. 3(a) and Fig. 3(b), feedback linearization provides trajectory tracking only when the controlled system is perfectly known. In order to emphasize the inferior tracking performance caused by a dead-zone input, the phase portrait related to the tracking is shown in Fig. 4.

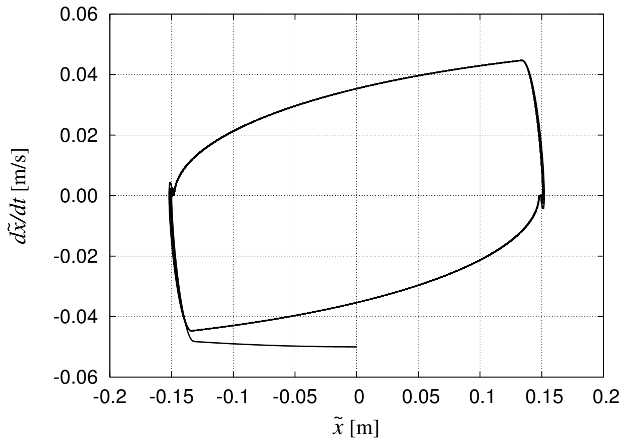


Figure 4. Phase portrait of the trajectory tracking with dead-zone input.

It can be clearly ascertained from Figs. 3 and 4 that the presence of an unknown dead-zone leads to an inferior tracking performance and to a limit cycle in the state space. On this basis, we propose the adoption of fuzzy inference system within the control law, in order to enhance the feedback linearization scheme. As demonstrated by Bessa and Barrêto (2010), fuzzy algorithms can be properly combined with nonlinear controllers in order to improve the trajectory tracking of uncertain nonlinear systems. It has also been shown that such strategies are suitable for a variety of applications ranging from remotely operated underwater vehicles (Bessa *et al.*, 2008, 2010b) to chaos control (Bessa *et al.*, 2009).

The adopted fuzzy inference system was the zero order TSK (Takagi–Sugeno–Kang), with the r^{th} rule stated in a linguistic manner as follows:

$$\textit{If } \tilde{x} \textit{ is } \tilde{X}_r \textit{ and } \dot{\tilde{x}} \textit{ is } \dot{\tilde{X}} \textit{ then } \hat{d}_r = \hat{D}_r \quad ; \quad r = 1, 2, \cdots, N$$

where $\tilde{X}$ and $\dot{\tilde{X}}$ are fuzzy sets, whose membership functions could be properly chosen, and $\hat{D}_r$ is the output value of each one of the N fuzzy rules.

Considering that each rule defines a numerical value as output $\hat{D}_r$, the final output $\hat{d}$ an be computed by a weighted average:

$$\hat{d}(\tilde{x}, \dot{\tilde{x}}) = \frac{\sum_{r=1}^{N} w_r \cdot \hat{D}_r}{\sum_{r=1}^{N} w_r} \tag{18}$$

or, similarly,

$$\hat{d}(\tilde{x}, \dot{\tilde{x}}) = \hat{\mathbf{D}}^{\mathrm{T}} \mathbf{\Psi}(\tilde{x}, \dot{\tilde{x}}) \tag{19}$$

where, $\hat{\mathbf{D}} = [\hat{D}_1, \hat{D}_2, \ldots, \hat{D}_N]$ is the vector containing the attributed values $\hat{D}_r$ to each rule r, $\mathbf{\Psi}(\tilde{x}, \dot{\tilde{x}}) = [\psi_1, \psi_2, \ldots, \psi_N]$ is a vector with components $\psi_r(\tilde{x}, \dot{\tilde{x}}) = w_r / \sum_{r=1}^{N} w_r$ and w_r is the firing strength of each rule, which can be computed from the membership values with any fuzzy intersection operator (t-norm).

Thus, the control law with the fuzzy compensation scheme can be stated as follows

$$u = b^{-1}(\mathbf{a}^{\mathrm{T}}\mathbf{x} + \dddot{x}_d - 3\lambda\ddot{\tilde{x}} - 3\lambda^2\dot{\tilde{x}} - \lambda^3\tilde{x}) + \hat{d}(\tilde{x}, \dot{\tilde{x}}) \tag{20}$$

In this work, the adopted fuzzy rule base is presented in Table 1, where NB, NM, NS, ZO, PS, PM and PB represents, respectively, Negative–Big, Negative–Medium, Negative–Small, Zero, Positive–Small, Positive–Medium and Positive–Big. The central values of the membership functions, related to the tracking error and its derivative, are $C_{\tilde{x}} = \{-0.032; -0.008; -0.002; 0; 0.002; 0.008; 0.032\}$ and $C_{\dot{\tilde{x}}} = \{-0.025; -0.005; -0.001; 0; 0.001; 0.005; 0.025\}$. With respect to the output of each rule, the following values were heuristically adopted for NB to PB: $\hat{D}_r = \{-2.0\ ;\ -0.4\ ;\ -0.2\ ;\ 0.0\ ;\ 0.2\ ;\ 0.4\ ;\ 2.0\}$.

Table 1. Adopted fuzzy rule base.

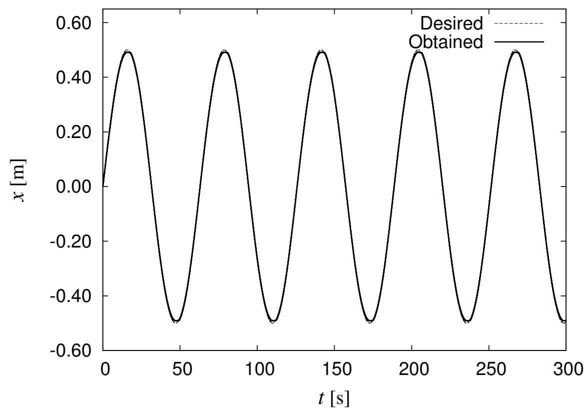
$\tilde{x}$ / $\dot{\tilde{x}}$	NB	NM	NS	ZO	PS	PM	PB
NB	PB	PB	PB	PB	PB	PB	PM
NM	PB	PB	PM	PM	PM	PM	PS
NS	PB	PM	PM	PS	PS	PS	NS
ZO	PM	PS	PS	ZO	NS	NS	NM
PS	PS	NS	NS	NS	NM	NM	NB
PM	NS	NM	NM	NM	NM	NB	NB
PB	NM	NB	NB	NB	NB	NB	NB

In order to evaluate the performance of the proposed control law, Eq. (20), some numerical simulations were carried out. In the first case, it was assumed that the model parameters were perfectly known, but a dead-zone input was considered for the electro-hydraulic system. Figure 5 shows the obtained results for the tracking of $x_d = 0.5\sin(0.1t)$ m.

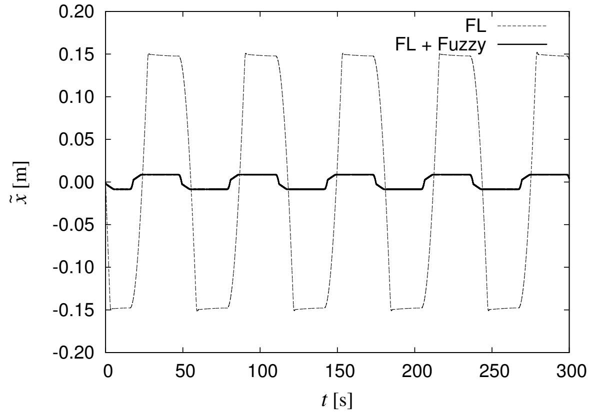
As observed in Fig. 5, despite the dead-zone input, the proposed control scheme allows the electro-hydraulic actuated system to track the desired trajectory with a small tracking error. Through the comparative analysis showed in Fig. 5(b), the improved performance of the proposed controller over the uncompensated counterpart can be easily ascertained.

Now, the capability of the proposed scheme to deal with uncertainties was appraised by choosing the parameters for the controller based on the assumption that exact values are not known but with a maximal uncertainty of ±10% over previous adopted values. Figures 6 and 7 show the obtained results.

As observed in Figs. 6 and 7, even in the presence of parametric uncertainties and a dead-zone input the proposed controller is able to provide trajectory tracking with a small associated error. By comparing the phase portrait obtained without the fuzzy compensation scheme (Fig. 4 and with compensation (Fig. 4), the superior performance of the proposed controller is noticeable.

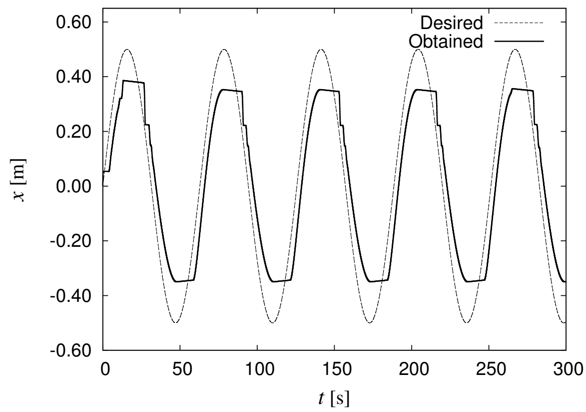


(a) Tracking with fuzzy compensation.

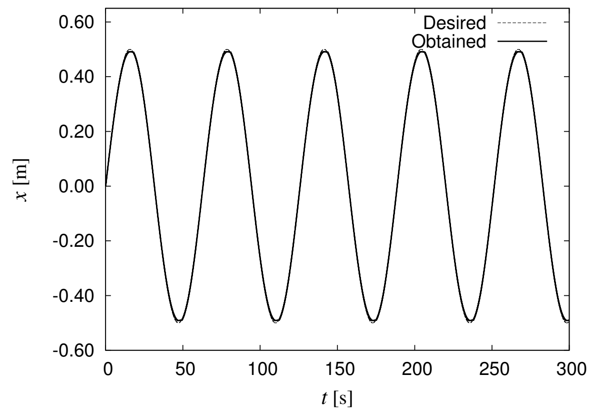


(b) Error with and without fuzzy compensation.

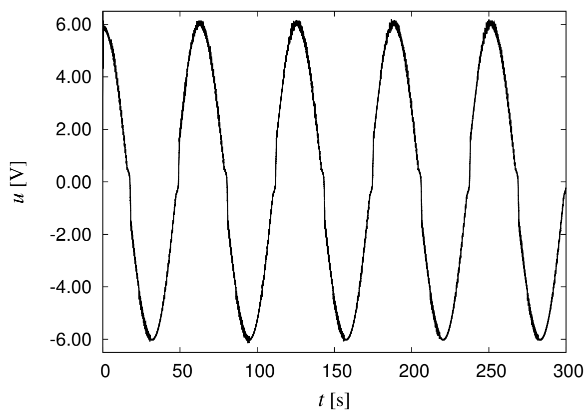
Figure 5. Tracking performance with $x_d = 0.5\sin(0.1t)$ m and considering a dead-zone input.



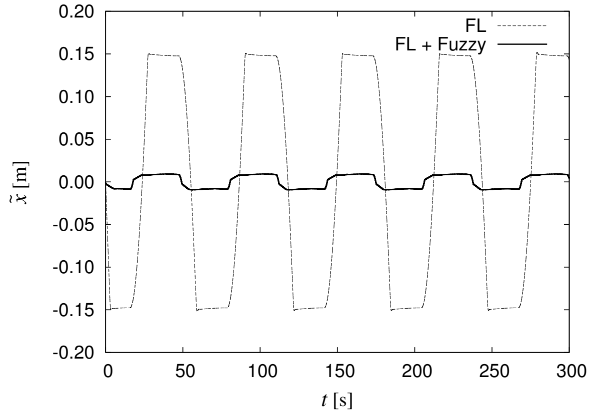
(a) Tracking without fuzzy compensation.



(b) Tracking with fuzzy compensation.



(c) Control voltage with fuzzy compensation.



(d) Error with and without fuzzy compensation.

Figure 6. Tracking performance with $x_d = 0.5\sin(0.1t)$ m and considering a dead-zone input and uncertainties.

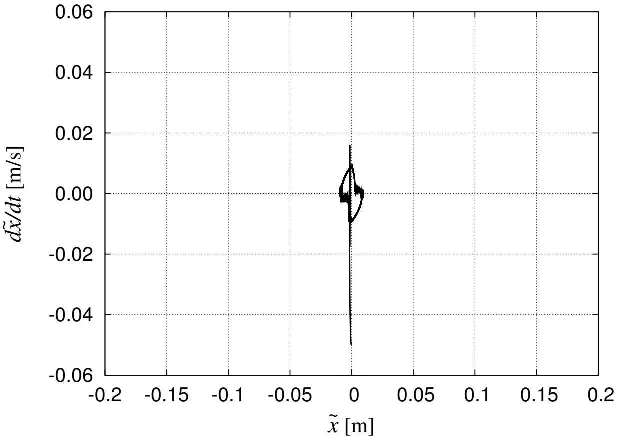


Figure 7. Phase portrait of the trajectory tracking with fuzzy compensation and considering a dead-zone input.

5. CONCLUDING REMARKS

The present work addressed the problem of controlling electro-hydraulic servosystems with unknown dead-zone. A nonlinear controller based on the feedback linearization method and enhanced by a fuzzy inference system was implemented to deal with the position trajectory tracking problem. The control system performance was confirmed by means of numerical simulations. The adoption of a fuzzy algorithm provided an smaller tracking error due to its ability to compensate for the dead-zone nonlinearity, as well as the uncertainties with respect to model parameters.

6. ACKNOWLEDGEMENTS

The authors would like to acknowledge the support of the Brazilian National Research Council (CNPq), the Brazilian Coordination for the Improvement of Higher Education Personnel (CAPES) and the German Academic Exchange Service (DAAD).

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