Adaptive fuzzy control of electrohydraulic servosystems

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Abstract

Electrohydraulic servosystems 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 electrohydraulic actuated 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 an adaptive fuzzy controller for electrohydraulic actuated systems with unknown dead-zone. The stability properties of the closed-loop systems was proven using Lyapunov stability theory and Barbalat’s lemma. Numerical results are presented in order to demonstrate the control system performance.

I. INTRODUCTION

Electrohydraulic 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 electrohydraulic 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 (Knohl and Unbehauen 2000; Yao et al. 2000) and variable structure (Bouchis et al. 2001; Jerouane et al. 2004; Liu and Handroos 1999; Mihajlov et al. 2002) methodologies, but nonlinear controllers based on quantitative feedback theory (Sohl and Bobrow 1999; Niksefat and Sepehri 2000), optimal tuning PID control (Liu and Daley 2000), integrator backstepping method (Chen et al. 2002) and fuzzy model reference learning control (Testi et al. 2003) were also presented.

In addition to the common nonlinearities that originate from the compressibility of the hydraulic fluid and valve flow-pressure properties, most electrohydraulic 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.

Intelligent control has proven to be a very attractive approach to cope with uncertain nonlinear systems (Bessa et al. 2005a, 2005b, 2017, 2018, 2019; Dos Santos and Bessa 2019; Lima et al. 2018, 2020; Tanaka et al. 2013). By combining nonlinear control techniques, such as feedback linearization or sliding modes, with adaptive intelligent algorithms, for example fuzzy logic or artificial neural networks, the resulting intelligent control strategies can deal with the nonlinear characteristics as well as with modeling imprecisions and external disturbances that can arise.

In this work, an adaptive fuzzy controller is developed for electrohydraulic actuated systems with unknown dead-zone to deal with the position trajectory tracking problem. The adopted approach is based on a recently proposed strategy (Bessa and Dutra 2005), that does not requires previous knowledge of dead-zone parameters. The global stability of the closed-loop system was proven using Lyapunov stability theory and Barbalat’s lemma. Some numerical results are also presented in order to demonstrate the control system performance.

II. ELECTROHYDRAULIC 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 (Walters 1967; Merritt 1967).

The electrohydraulic 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.

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

$$F_p = A_1P_1 - A_2P_2 = M_t\ddot{x} + B\dot{x} + Kx$$

(1)

where $F_p$ 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,
\( B_p \) is the viscous damping coefficient of piston and load, \( K \) 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. \( (1) \) can be rewritten as

\[
M \ddot{x} + B_p \dot{x} + K x = A_p P_L \quad (2)
\]

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

\[
Q_L = A_p \dot{x} + C_{tp} + V_t \frac{4\beta_e}{\rho} P_L \quad (3)
\]

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 \( \beta_e \) 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

\[
Q_L = C_d w \bar{x}_{sp} \sqrt{\frac{1}{\rho} (P_s - \text{sgn}(\bar{x}_{sp}) P_L)} \quad (4)
\]

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

\[
\text{sgn}(z) = \begin{cases} 
-1 & \text{if } z < 0 \\
0 & \text{if } z = 0 \\
1 & \text{if } z > 0 
\end{cases} \quad (5)
\]

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. \( 2 \).

\[
\bar{x}_{sp}(t) = \begin{cases} 
k_v (u(t) - \delta_l) & \text{if } u(t) \leq \delta_l \\
0 & \text{if } \delta_l < u(t) < \delta_r \\
k_v (u(t) - \delta_r) & \text{if } u(t) \geq \delta_r 
\end{cases} \quad (6)
\]
where \( k_v \) is the valve gain and the parameters \( \delta_1 \) and \( \delta_r \) depends on the size of the overlap region.

For control purposes, as shown by [Bessa and Dutra (2005)], Eq. (6) can be rewritten in a more appropriate form:

\[
x_{\text{eq}}(t) = k_v (u(t) - d(u))
\]

where \( d(u) \) can be obtained from Eq. (6) and Eq. (7):

\[
d(u) = \begin{cases} 
\delta_l & \text{if } u(t) \leq \delta_l \\
\delta_r & \text{if } u(t) \geq \delta_r \\
u(t) & \text{if } \delta_l < u(t) < \delta_r
\end{cases}
\]

Combining equations (2), (3), (4), (7) and (8) leads to a third-order differential equation that represents the dynamic behavior of the electrohydraulic system:

\[
\dot{x} = -a^T x + bu - bd(u)
\]

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

\[
a_0 = \frac{4\beta_v p_w k_v}{V_l M_t} \quad ; \quad a_1 = \frac{K}{M_t} + \frac{4\beta_v p_w}{V_l M_t} + \frac{4\beta_w C_{dp}}{V_l M_t} \quad ; \quad a_2 = \frac{B_p}{M_t} + \frac{4\beta_w C_{dp}}{V_l}
\]

and

\[
b = \frac{4\beta_v p_w C_{dp} k_v}{V_l M_t} \sqrt{\frac{1}{p_s} [p_s - \text{sgn}(u) (M_t \ddot{x} + B_p \dot{x} + K x)/A_p]}
\]

Based on the dynamic model presented in Eq. (9), an adaptive fuzzy controller will be developed in the next section.

### III. ADAPTIVE FUZZY CONTROLLER

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

Now, defining a combined tracking error measure \( e = c^T \tilde{x} \), where \( c = [c_0, c_1, 1]^T \) and the coefficients \( c_0 \) and \( c_1 \) chosen in order to make \( c^T \ddot{x} + c_1 p + c_0 \) a Hurwitz polynomial, the following control law can be proposed:

\[
u = b^{-1} (a^T \tilde{x} + \tilde{x}_d - c_1 \dot{\tilde{x}} - c_0 \ddot{\tilde{x}}) + \hat{d}(\hat{u}) - \kappa e \tag{10}
\]

where \( \kappa \) is a strictly positive constant and \( \hat{d}(\hat{u}) \) an estimate of \( d(u) \), that will be computed in terms of the equivalent control

\[
u = b^{-1} (a^T \tilde{x} + \tilde{x}_d - c_1 \dot{\tilde{x}} - c_0 \ddot{\tilde{x}}) + \hat{d}(\hat{u}) - \kappa e
\]

The adopted fuzzy inference system was the zero order TSK (Takagi–Sugeno–Kang), whose rules can be stated in a linguistic manner as follows:

If \( \hat{u} \) is \( \hat{U}_r \) then \( \hat{d}_r = \hat{D}_r \); \( r = 1, 2, \ldots, N \)

where \( \hat{U}_r \) 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 \( \hat{d} \) can be computed by a weighted average:

\[
\hat{d}(\hat{u}) = \sum_{r=1}^{N} w_r \cdot \hat{d}_r
\]

or, similarly,

\[
\hat{d}(\hat{u}) = \hat{D}^T \Psi(\hat{u}) \tag{11}
\]

where, \( \hat{D} = [\hat{D}_1, \hat{D}_2, \ldots, \hat{D}_N]^T \) is the vector containing the attributed values \( \hat{D}_r \) to each rule \( r \), \( \Psi(\hat{u}) = [\psi_1(\hat{u}), \psi_2(\hat{u}), \ldots, \psi_N(\hat{u})]^T \) is a vector with components \( \psi_r(\hat{u}) = w_r / \sum_{r=1}^{N} w_r \) and \( w_r \) is the firing strength of each rule.

To ensure the best possible estimate \( \hat{d}(\hat{u}) \), the vector of adjustable parameters can be automatically updated by the following adaptation law:

\[
\dot{\hat{D}} = -\varphi e \Psi(\hat{u}) \tag{13}
\]

where \( \varphi \) is a strictly positive constant related to the adaptation rate.

Before proving the closed-loop system stability, the following assumptions must be made:

**Assumption 1** The states \( x, \dot{x} \) and \( \ddot{x} \) are available.

**Assumption 2** The desired trajectory \( x_d \) is \( C^2 \). Furthermore \( x_d, \dot{x}_d, \ddot{x}_d \) and \( \dddot{x}_d \) are available and with known bounds.
Theorem 1 Let the electrohydraulic servosystem with a dead-zone at the input be represented by Eq. Then, subject to Assumptions 2, the adaptive fuzzy controller defined by 10 ensures the global stability of the closed-loop system and trajectory tracking.

Proof: Let a positive definite Lyapunov function candidate \( V \) be defined as

\[
V(t) = \frac{1}{2} e^2 + \frac{b}{2\varphi}\Delta^T\Delta
\]

where \( \Delta = \hat{D} - \hat{D}^* \) is the optimal parameter vector, associated to the optimal estimate \( \hat{d}^*(u) = d(u) \). Thus, the time derivative of \( V \) is

\[
\dot{V}(t) = e\dot{e} + b\varphi^{-1}\Delta^T\dot{\Delta} = (\ddot{x} + c_1\dot{x} + c_0\dot{x})e + b\varphi^{-1}\Delta^T\Delta \\
= (\ddot{x} + (c_1 + \varphi e)\dot{x} + c_0\dot{x})e + b\varphi^{-1}\Delta^T\Delta
\]

Applying the proposed control law 10 and noting that \( \ddot{x} = \hat{D} \), then

\[
\dot{V}(t) = [b(\hat{d} - d) - \kappa e]e + b\varphi^{-1}\Delta^T\dot{\hat{D}} = [b\Delta^T\Psi(\hat{u}) - \kappa e]e + b\varphi^{-1}\Delta^T\dot{\hat{D}} = -\kappa e^2 + b\varphi^{-1}\Delta^T(\dot{\hat{D}} - \varphi e\Psi(\hat{u}))
\]

Furthermore, defining \( \dot{\hat{D}} \) according to 13, \( \dot{V}(t) \) becomes

\[
\dot{V}(t) = -\kappa e^2
\]

which implies that \( \dot{V}(t) \leq V(0) \) and that \( e \) and \( \Delta \) are bounded. From the definition of \( e \) and considering Assumption 2, it can be easily verified that \( \dot{e} \) is also bounded.

To establish the global stability of the closed-loop system, the time derivative of \( V \) must be analyzed:

\[
\dot{V}(t) = -2\kappa ee
\]

which implies that \( \dot{V}(t) \) is also bounded and, from Barbalat’s lemma, that \( e \to 0 \) as \( t \to \infty \).

For \( e = 0 \), the following error dynamics take place:

\[
\ddot{x} + c_1\dot{x} + c_0\dot{x} = 0
\]

Thus, if the coefficients \( c_0 \) and \( c_1 \) were properly chosen, the associated characteristic polynomial is a Hurwitz polynomial, which ensures the convergence of the tracking error to zero, \( x \to 0 \) as \( t \to \infty \), and completes the proof.

In the following section, some numerical simulations are presented in order to evaluate the performance of the adaptive fuzzy controller.

Some applications may require the use of robust controllers. In such a case, the reader is referred to the adaptive fuzzy sliding mode controllers presented in Bessa (2005) and Bessa et al. (2005b).

IV. Simulation results

The simulation studies were performed with a numerical implementation in C, with sampling rates of 400 Hz for control system and 800 Hz for dynamic model. The adopted parameters for the electrohydraulic 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), \( \beta_s = 700 \) MPa, \( V_1 = 6 \times 10^{-5} \) m\(^3\), \( M_1 = 250 \) kg, \( B_p = 100 \) Ns/m, \( K = 75 \) N/m, \( \delta_s = -1.1 \) V and \( \delta_r = 0.9 \) V. The parameters of the controller were \( \lambda = 8 \), \( \kappa = 1 \) and \( \varphi = 0.5 \). For the fuzzy system were adopted triangular and trapezoidal membership functions for \( U_s \), with the central values defined as \( C = \{ -0.50 ; -0.10 ; -0.05 ; 0.00 ; 0.05 ; 0.10 ; 0.50 \} \).

To evaluate the performance of the proposed control law, Eq. 10, some numerical simulations were carried out. Figure 3 shows the results obtained with \( x_s = 0.5 \sin(0.17t) \) m. In Fig. 4, variations of \( \pm 20\% \) in the supply pressure, \( P_s = 7(1 + 0.2 \sin(x)) \) MPa, were also taken into account. Such variations are very common in real plants.

As observed in Fig. 3(c) and Fig. 4(c), the adopted controller provides good tracking performance and is almost indifferent to variations in the supply pressure. It can be easily verified in Fig. 5(d) and Fig. 4(d) that, in both cases, the chosen adaptive algorithm shows a fast response.

V. Concluding remarks

The present work addressed the problem of controlling electrohydraulic servosystems with unknown dead-zone. An adaptive fuzzy controller was implemented to deal with the position trajectory tracking problem. The stability and convergence properties of the closed-loop systems were proven using Lyapunov stability theory and Barbalat’s lemma. The control system performance was also confirmed by means of numerical simulations. The adaptive algorithm could automatically recognize the dead-zone nonlinearity and previously compensate its undesirable effects.
Figure 3: Tracking performance with $x_d = 0.5 \sin(0.1t)$ m and constant supply pressure.

Figure 4: Tracking performance with $x_d = 0.5 \sin(0.1t)$ m and variable supply pressure.

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