ADAPTIVE CASCADE CONTROL OF A HYDRAULIC ACTUATOR WITH AN ADAPTIVE DEAD-ZONE COMPENSATION

Mauro A. B. Cunha
Pelotas Federal Center for Technological Education
Praça Vinte de Setembro, 455 – CEP 96015-360 – Pelotas – RS - Brazil
mauro@cefetrs.tche.br

Raul Guenther
Federal University of Santa Catarina
Campus Universitário - Trindade - CEP 88040-900 - Florianópolis – SC - Brazil
guenther@emc.ufsc.br

Abstract. This work presents a cascade controller for the trajectory tracking control of a hydraulic actuator by using adaptive algorithms to deal with parametric uncertainties together with an adaptive dead-zone compensation scheme. The hydraulic actuator is modeled as a forth order nonlinear system including the valve dynamic as a first order linear system and considering a valve with a dead-zone. Some hydraulic actuator parameters are difficult to be determined or measured, leading to the necessity of considering parametric uncertaities in the design process. Furthermore, the valve dead-zone causes trajectory tracking errors. In the cascade strategy, the hydraulic actuator mathematical model is interpreted as a mechanical subsystem (mass and damper) driven by a hydraulic one (valve and cylinder). From this interpretation, one uses a cascade controller by combining two different adaptive control laws for each subsystem and an adaptive algorithm to compensate the dead-zone. Simulation results illustrate the main characteristics of the proposed controller.

Keywords: hydraulic actuator, adaptive control, adaptive dead-zone compensation, parametric uncertainties, trajectory tracking.

1. Introduction

Hydraulic actuators are very attractive for applications that require a high force/size ratio. Their use has increased in the last few years mainly due to the electronics development. Such a development allowed that elaborated control laws could be used to overcome the limitations imposed by the hydraulic actuators. These limitations arise because the hydraulic actuator is a highly nonlinear system and has lightly damped dynamics. Furthermore, some parameters of the hydraulic actuator mathematical model are difficult to be obtained from experimental measurements.

Due to the outlined characteristics, there has been a research effort to develop controllers that can overcome the hydraulic actuator limitations for high performance applications by using different control techniques (Selemic and Lewis, 2000, Sirouspour and Salcudean, 2001, Bu and Yao, 2000, Virvalo, 2002).

This work's authors have developed controllers in a cascade strategy that is based on the reduction order proposed by Utkin (1987). In a gradual manner, from the work of Guenther and De Pieri (1997), many cascade controllers have been proposed in order to overcome different limitation of the hydraulic actuators. Here, one outlines some of them: parametric uncertainties in the mechanical subsystem (Cunha et al., 1998), parametric uncertainties in the mechanical and hydraulic subsystems (Guenther et al., 1998, 2000), inclusion of the valve dynamic (Cunha et al., 2000), design methodology (Cunha et al., 2002). More details about such developments can be found in Cunha (2001).

In Cunha et al. (2004), a fixed cascade controller NFCC (Cunha, 2001) was combined with an adaptive dead-zone compensation. However, the parameters of the mechanical and hydraulic subsystems were considered known. In Cunha (2005), considering the valve dynamic, a cascade controller with adaptive algorithms for the mechanical and hydraulic subsystems and was proposed. However, the valve considered in that work did not have a dead-zone.

In this work, in order to deal with parametric uncertainties in both subsystems and with the valve dead-zone nonlinearity, one proposes a controller that combines FACC (Cunha, 2005) together with an adaptive dead-zone compensation scheme (Cunha et al., 2004). This controller is referred as FACCADZC. By using adaptive control in the mechanical and hydraulic subsystems, it is not necessary to know the exact values of the system parameters. Furthermore, some parameters are difficult to be obtained and others vary according to the operation point or due to the application. By using an adaptive compensation scheme for the valve dead-zone, it is not necessary to do tests to obtain the dead-zone breakpoints. The dry friction (static + Coulomb) is another source of trajectory thacking errors and it will be taking into account in future works.

This work is divided as follows. Section 2 presents the mathematical model of a hydraulic actuator with a valve that presents a dead-zone. In section 3, a combination of the FACC with an adaptive dead-zone compensation algorithm is presented (FACCADZC). Section 4 presents the simulation results and, in section 5, the conclusions and perspective are outlined.

2. Hydraulic actuator

2.1 Mathematical model

The hydraulic actuator considered in this work is shown in Fig. 1 (Cunha et al., 2004), where M represents the system total mass, B is the viscous friction coefficient, p_s is the supply pressure, p_0 is the return pressure, p_1 and p_2 are the pressure in lines 1 and 2, v_1 and v_2 are the volume in lines 1 and 2, A is the cylinder piston cross sectional area, Q_1 is the flowrate from the valve to chamber 1, Q_2 is the flowrate from chamber 2 to the valve and u is the electrical voltage applied to the electronic card.

The main difference of the model presented below (Cunha et al., 2004) and the model considered in Cunha (2005) is the presence of a valve dead-zone in the relationship between the valve spool displacement and the flowrate. It is caused by the fact that the lands of the spool are greater than the annular parts of the valve body. Valves with dead-zones require less precision in the fabrication and are less expensive than those with a null overlap.
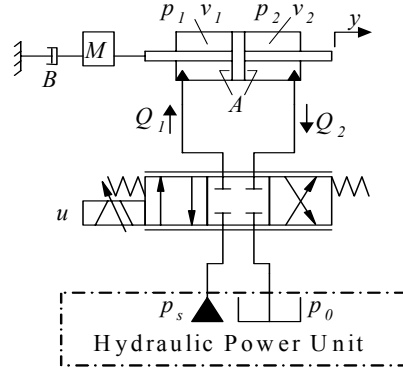

Figure 1 – Hydraulic Actuator

The hydraulic actuator mathematical model can be written as (Cunha et al., 2004)

$$M\ddot{y} + B\dot{y} = Ap_\Delta \tag{1}$$

$$\dot{p}_\Delta = -fA\dot{y} + fK_h g x_v \tag{2}$$

$$x_v = DZ_1(x_{vb}) \tag{3}$$

$$\dot{x}_{vb} = -\omega_v x_{vb} + K_{em}\omega_v u , \tag{4}$$

$$x_v = DZ_1(x_{vb}) = \begin{cases} x_{vb} - b_r, & x_{vb} > b_r \\ 0, & b_l \leq x_{vb} \leq b_r \\ x_{vb} - b_l, & x_{vb} < b_l \end{cases} \tag{5}$$

where $f = f(y) = \dfrac{\beta v}{(0.5v)^2 - (Ay)^2}$, x_v is the valve spool position, $p_\Delta = p_1 - p_2$ is the cylinder chambers pressure difference, β is the bulk modulus, $v = v_1 + v_2$, K_h is the hydraulic constant, $g = g(p_\Delta, x_v) = \sqrt{p_S - sgn(x_v)p_\Delta}$, K_{em} is the valve constant, ω_v is the valve bandwidth, x_{vb} is the valve spool position before the dead-zone, b_r is the right breakpoint and b_l is the left breakpoint. Figure 2 shows a block diagram of the relation between the control input u and the spool position (x_v). The signal x_{vb} is the signal that is measured by an internal transducer in the valve and is available in the electronic card.
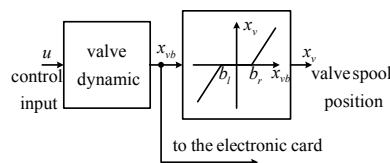

Figure 2 – Block diagram of a valve with dead-zone

2.2 Comments on parametric uncertaities and nonlinearities

There are many parameters in a hydraulic actuator that are intrinsically difficult to be measured. This is the case of the bulk modulus (β). Some others depend on the application, like the system total mass (M).

The bulk mudulus varies with temperature, pressure and air inside the actuator. The parameter K_h can be calculated by using some data from the valve manufacturer, but it also depends on some variables, in such a way that the calculated value is only an approximation. The dynamic of a valve is very nonlinear and complex, then the use of a first order linear system is also an approximation. The valve manufacturers normally provide different time constants for differents values of u. The area (A) and the volume (v) can usually be determined with a good precision.

The mass (M) will be constant in applications where the load is always the same. In case of different loads must be manipulated, like in robots, the mass will vary. The viscous friction coefficient (B) is a linearization of the friction force and, therefore, its value depend on the operation point.

The valve electronic cards normally have adjusts that can be used to decrease the effective valve dead-zone. However, like the other hydraulic parameters, the breakpoint values depend on the operation point. Therefore, the use of an algorithm that can track the breakpoints values is a good way to try to cancel the dead-zone.

The dry friction (static + Coulomb) is another source of trajectory thacking errors. This is not being taking into account in this work.

3. Adaptive cascade controller with an adaptive dead-zone compensation

The cascade strategy applied to a hydraulic actuator consists in interpreting the hydraulic actuator mathematical model as two interconnected subsystems (see Fig. 3): a hydraulic subsystem and mechanical one. Such a strategy can be summarized as (Guenther and De Pieri, 1997):

(i) Compute a control law $p_{\Delta d}$ (desired pressure difference) for the mechanical subsystem such that the output y tracks the desired trajectory y_d as closed as possible;

(ii) Compute a control law u for the hydraulic subsystem such that p_Δ tracks $p_{\Delta d}$ as close as possible.
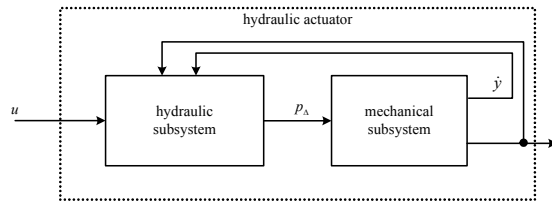

Figure 3 – Interconnected subsystems

Cunha (2005) proposes an adaptive cascade controller (FACC – Full Adaptive Cascade Controller) combining two adaptive control laws for the hydraulic and mechanical subsystems, where the mechanical subsystem adaptive control law is given by (Slotine and Li, 1987)

$$p_{\Delta d} = \left(Y\hat{\theta} - K_D z\right)/A_0 \tag{6}$$

where $()_0$ means the nominal parameter or calculated based on the nominal parameter, K_D is a positive constant, $Y = \begin{bmatrix} \ddot{y}_r & \dot{y}_r \end{bmatrix}$ is the regressor, $\tilde{y} = y - y_d$ is the position trajectory tracking error, $\dot{y}_r = \dot{y}_d - \lambda\tilde{y}$, $z = \dot{y} - \dot{y}_r = \dot{\tilde{y}} + \lambda\tilde{y}$, λ is a positive constant, $\hat{\theta} = \begin{bmatrix} \hat{M} & \hat{B} \end{bmatrix}^T$ is the estimated parameter vector, $\hat{M}$ is the estimated mass, $\hat{B}$ is the estimated viscous friction coefficient.

The mechanical subsystem adaptation law is given by

$$\dot{\hat{\theta}} = \dot{\tilde{\theta}} = -\Gamma Y^T z \tag{7}$$

where Γ is a positive symmetric matrix chosen as $\Gamma = \text{diag}(\gamma_1, \gamma_2)$.

The hydraulic subsystem control law is given by (Cunha, 2005)

$$x_{vd} = (1/g)\left[(\hat{\alpha}\dot{p}_{\Delta d} - K_P\tilde{p}_\Delta)/f_0 + \hat{\alpha}A_0\dot{y}\right] \tag{8}$$

$$u_{FACC} = \left[\hat{\tau}_v\dot{x}_{vd} - (\phi_1/\phi_2)f_0 g\tilde{p}_\Delta - K_V\tilde{x}_v + x_v\right]/K_{em0} \tag{9}$$

where x_{vd} is the spool position desired trajectory, $\widetilde{x}_v = x_v - x_{vd}$ is the spool position trajectory tracking error, $\widetilde{p}_\Delta = p_\Delta - p_{\Delta d}$ is the pressure difference trajectory tracking error, $\alpha = 1/K_h$ is the inverse of the hydraulic constant, $\tau_v = 1/\omega_v$ is the valve time constant, $\hat{\alpha}$ and $\hat{\tau}_v$ are the estimated parameters, ϕ_1 and ϕ_2 are positive constants and K_P is a positive constant.

The hydraulic subsystem adaptation law is given by (Cunha, 2005)

$$\dot{\hat{\alpha}} = -(\phi_1/\phi_3)\widetilde{p}_\Delta(\dot{p}_{\Delta d} + fA_0\ddot{y}) \tag{10}$$

$$\dot{\hat{\tau}}_v = -(\phi_2/\phi_4)\widetilde{x}_v\dot{x}_{vd} \tag{11}$$

where ϕ_3 and ϕ_4 are positive constants.

Remark 1 - By using the Lyapunov's direct method and the Barbalat's Lemma, it is proved for a system without valve dead-zone, that the closed-loop is globally stable and the trajectory error vector converges asymptotically to the origin as t→∞ regardless of the adapted parameters converge to the true values (Cunha, 2005). In the simulations, it was verified that the FACC works very well when applied to a hydraulic actuator with parametric uncertainties and without valve dead-zone.

Here, one is dealing with the problem when the hydraulic actuator is composed of a valve with a dead-zone. In this case, FACC will not be able to yield the trajectory tracking errors asymptotically to zero. To overcome this limitation, it is necessary to add an algorithm to deal with the errors caused by the dead-zone nonlinearity.

In Cunha et al. (2004), a fixed cascade controller with an adaptive dead-zone compensation was proposed to deal with the trajectory tracking errors caused by the valve dead-zone. Such a fixed cascade controller was designed under the assumption that the hydraulic and mechanical subsystems parameters were known. The adaptive dead-zone compensation is based on the algorithm proposed by Tao and Kokotovic (1996). To overcome this limitation, it is necessary to add an algorithm to deal with the errors caused by the parametric uncertainties in the hydraulic and mechanical subsystems.

Remark 2 – In Cunha et al. (2004), in the development of the adaptive dead-zone compensation, it is assumed a static relation between x_v and x_{vb} in such a way that $x_{vb} = K_{em}.u$, yielding to the following equations:

$$x_v = DZ_1(K_{em}u) = DZ_2(u) \tag{12}$$

$$x_v = DZ_2(u) = \begin{cases} K_{em}u - b_r, & u > p_r \\ 0, & p_l \leq u \leq p_r \\ K_{em}u - b_l, & u < p_l \end{cases} \tag{13}$$

where $p_r = b_r/K_{em}$ and $p_l = b_l/K_{em}$.

Now, one proposes a control algorithm by combining FACC with the adaptive dead-zone compensation proposed in Cunha et al. (2004). This algorithm is referred as FACCADZC (Full Adaptive Cascade Controller with an Adaptive Dead-Zone Compensation). Coupling the FACC with the adaptive dead-zone compensation scheme, one has

$$u_{FACCADZC} = \hat{D}ZI(u_{FACC}) = \begin{cases} \hat{p}_r + u_{FACC}, & u_{FACC} > 0 \\ 0, & u_{FACC} = 0 \\ \hat{p}_l + u_{FACC}, & u_{FACC} < 0 \end{cases} \tag{14}$$

$$\dot{\hat{p}}_r = \begin{cases} -\eta_1 u_{FACC}\widetilde{p}_\Delta, & 0 < u_{FACC} \leq p_{rmax} \\ 0, & otherwise \end{cases} \tag{15}$$

$$\dot{\hat{p}}_l = \begin{cases} \eta_2 u_{FACC}\widetilde{p}_\Delta, & p_{lmin} \leq u_{FACC} < 0 \\ 0, & otherwise \end{cases} \tag{16}$$

where η_1, η_2 and p_{rmax} are positive constants and p_{lmin} is a negative constant.

Note that the algorithm to compensate the dead-zone uses the dead-zone inverse with the breakpoints being indirectly estimated by the equations (15) and (16).

Remark 3 – A dead-zone is included to read the signal x_v correctly (see Cunha et al. (2000,2004)).

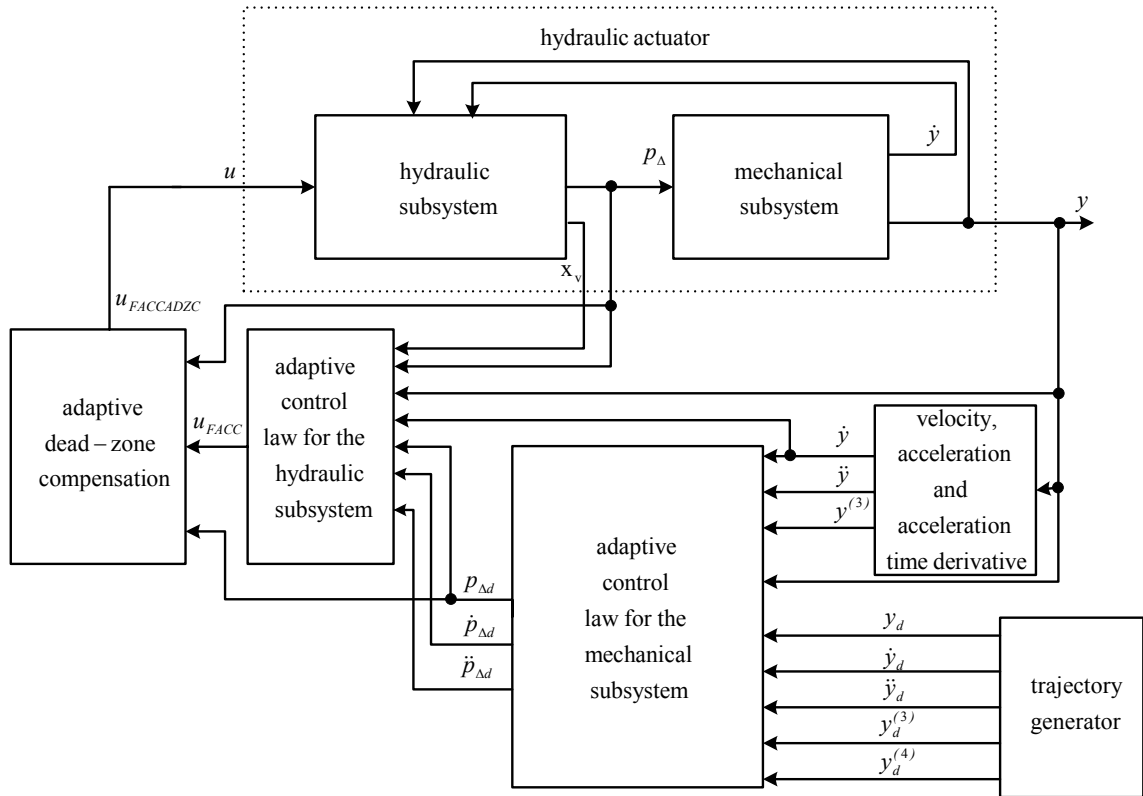Figure 4 shows a block diagram of the closed loop system with FACCADZC.

Figure 4 – Block Diagram of the closed loop system with FACCADZC

4. Simulation results

The closed loop system is composed of the hydraulic actuator mathematical model and the equations related to FACCADZC. The desired trajectory (Fig. 5) is the same proposed in Cunha et al. (2000). It is based on a 7th order polynomial (Eq.(17)) that was chosen in order to allow the choice of y_d, $\dot{y}_d$, $\ddot{y}_d$, $y_d^{(3)}$ at initial and final times.

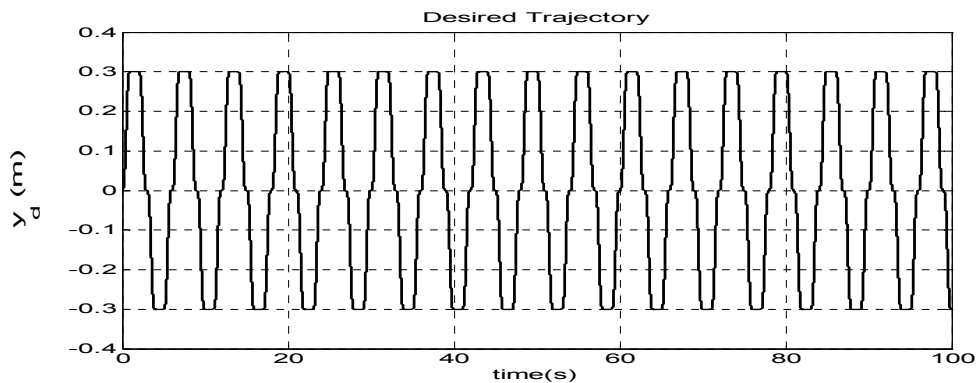$$y_{d1}(t) = -6t^7 + 21t^6 - 25,2t^5 + 10,5t^4 \tag{17}$$

Figure 5 – Desired Trajectory

The hydraulic actuator nominal parameters are $M_0 = 20.66$ Kg, $B_0 = 316.2$ N.s.m^{-1}, $p_{S0} = 100 \times 10^5$ Pa, $A_0 = 7.6576 \times 10^{-4}$ m^2, $v_0 = 9.5583 \times 10^{-4}$ m^3, $K_{h0} = 6.55 \times 10^{-8}$ m^4.V^{-1}.s^{-1}.N$^{\frac{-1}{2}}$, $\beta_0 = 10^9$ Pa, $K_{em0} = 0.76$, $\omega_{v0} = 147$ rad.s^{-1}, $b_r = 0.5$ V and $b_l = -0.8$ V (Cunha, 2001). The controller gains are $K_D = 11000$, $\lambda = 30$, $K_P = 7.6 \times 10^9$, $K_V = 0.61$, $\gamma_1 = 2000$, $\gamma_2 = 2000$, $\phi_1 = 6.5 \times 10^{-18}$, $\phi_2 = 1.5 \times 10^6$, $\phi_3 = 1 \times 10^{-10}$ and $\phi_4 = 5 \times 10^6$ (Cunha, 2005). The adaptation gains of the dead-zone compensation were set at $\eta_1 = 15 \times 10^{-6}$ and $\eta_2 = 15 \times 10^{-6}$.

Figure 6 shows the trajectory tracking error and the control input for FACCADZC with the hydraulic actuator with a dead-zone and with the following uncertainties: $M = 1.5M_0$, $B = 1.5B_0$, $K_h = 1.5K_{h0}$, $\omega_v = 1.5\omega_{v0}$ and $\beta = 1.5\beta_0$. The adaptation is turned on after 6 seconds. One can see that the trajectory tracking error tends asymptotically to zero after the adaptation algorithm to have been turned on.
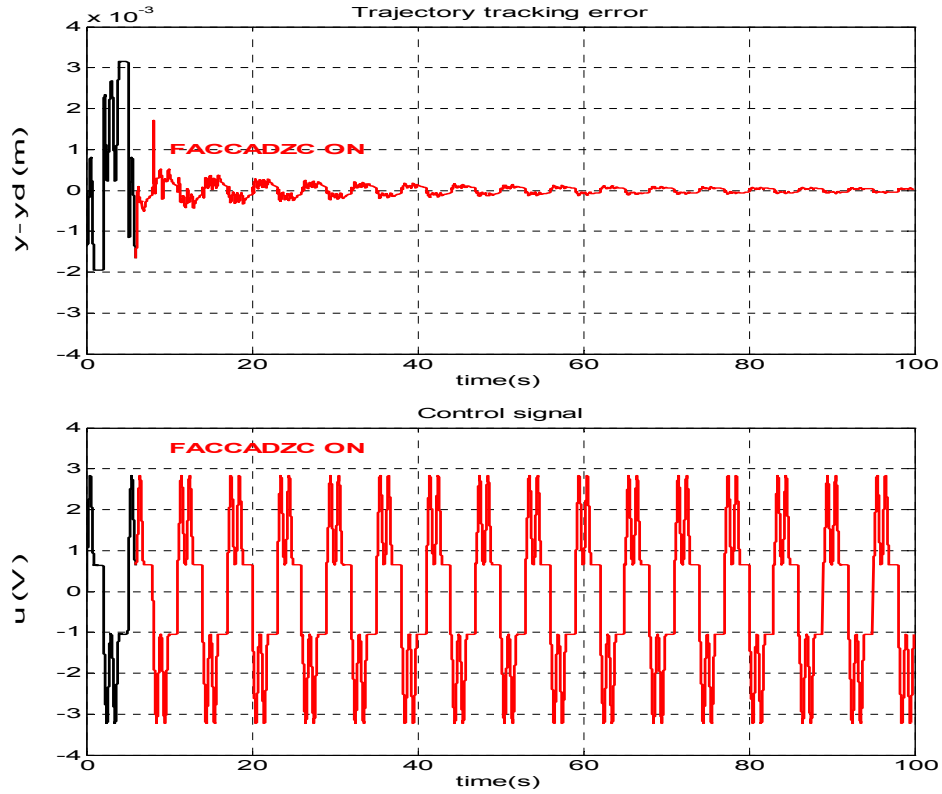

Figure 6 – FACCADZC applied to a hydraulic actuator with a valve dead-zone and parametric uncertainties

In Fig. 7, one can observe the convergence of the mechanical subsystem parameters. The convergence rate could be increased by using greater values for the mechanical subsystem adaptation gains γ_1 and γ_2. However, the values of γ_1 and γ_2 were set in the same values used in the experimental implementation in Cunha (2001).
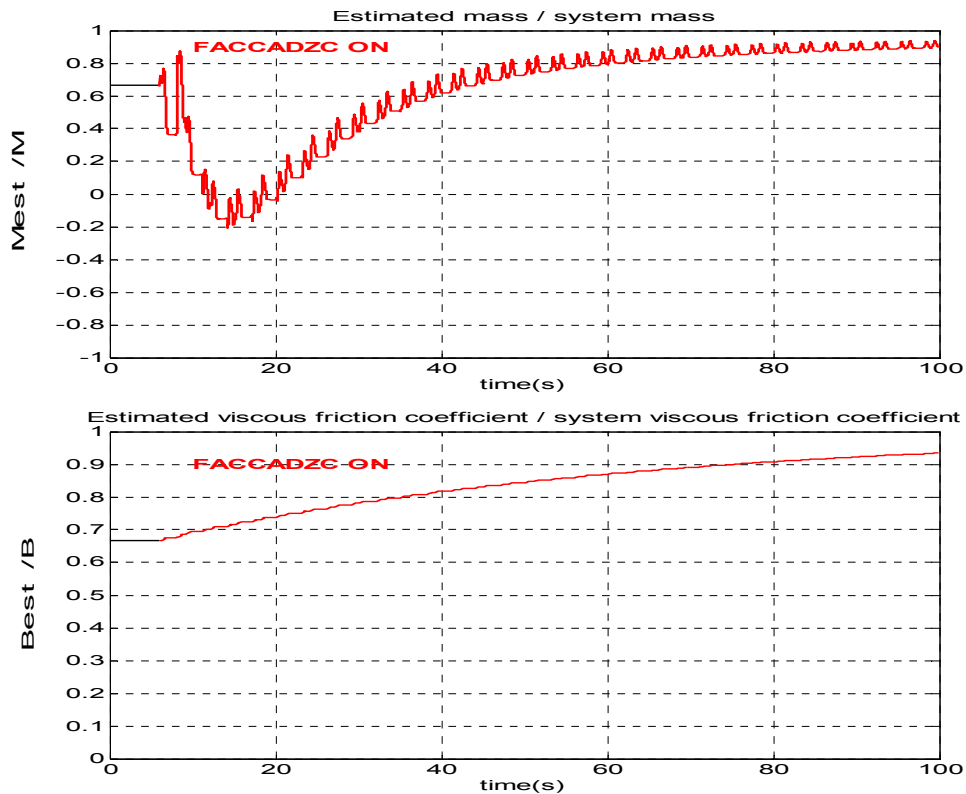

Figure 7 – Estimation of the mechanical subsystem parameters

Figure 8 shows the convergence of the hydraulic subsystem parameters. The convergence rate depends on $\left(\phi_1 / \phi_3\right)$ for $\hat{\alpha}$ and $\left(\phi_2 / \phi_4\right)$ for $\hat{\tau}$.
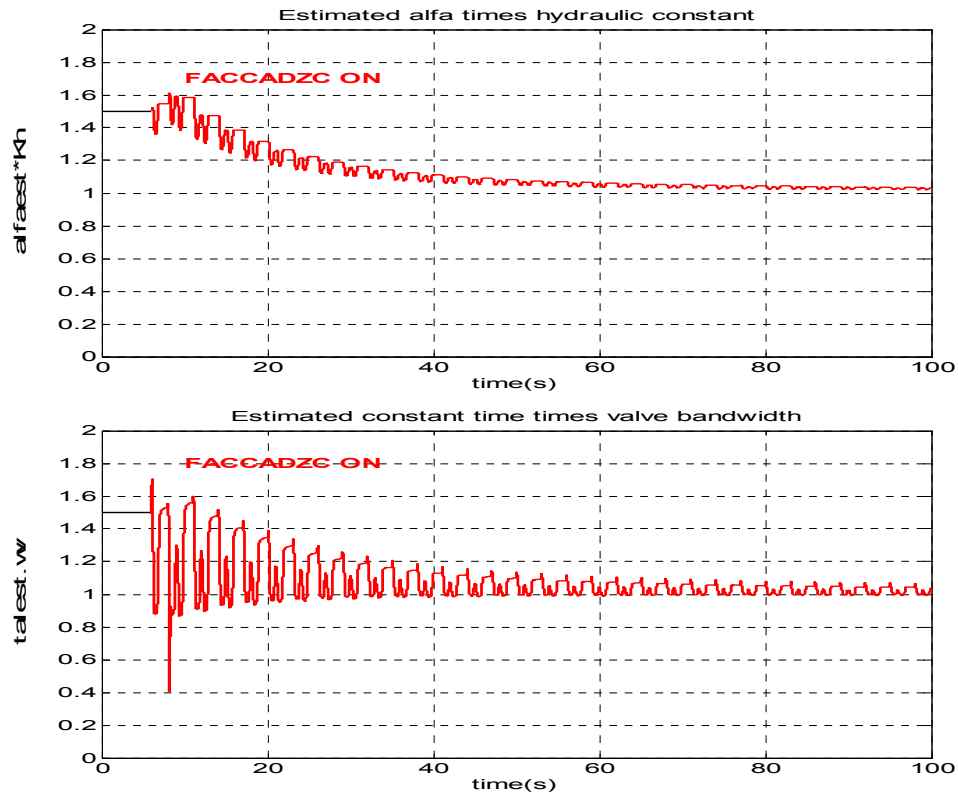
Figure 8 - Estimation of the hydraulic subsystem parameters

In Fig. 9, one can observe the convergence of the dead-zone parameters. The fast convergence of the adapted dead-zone parameters is very important since it is normally the major source of trajectory tracking errors. The convergence rate depends on the adaptation gains η_1 and η_2.
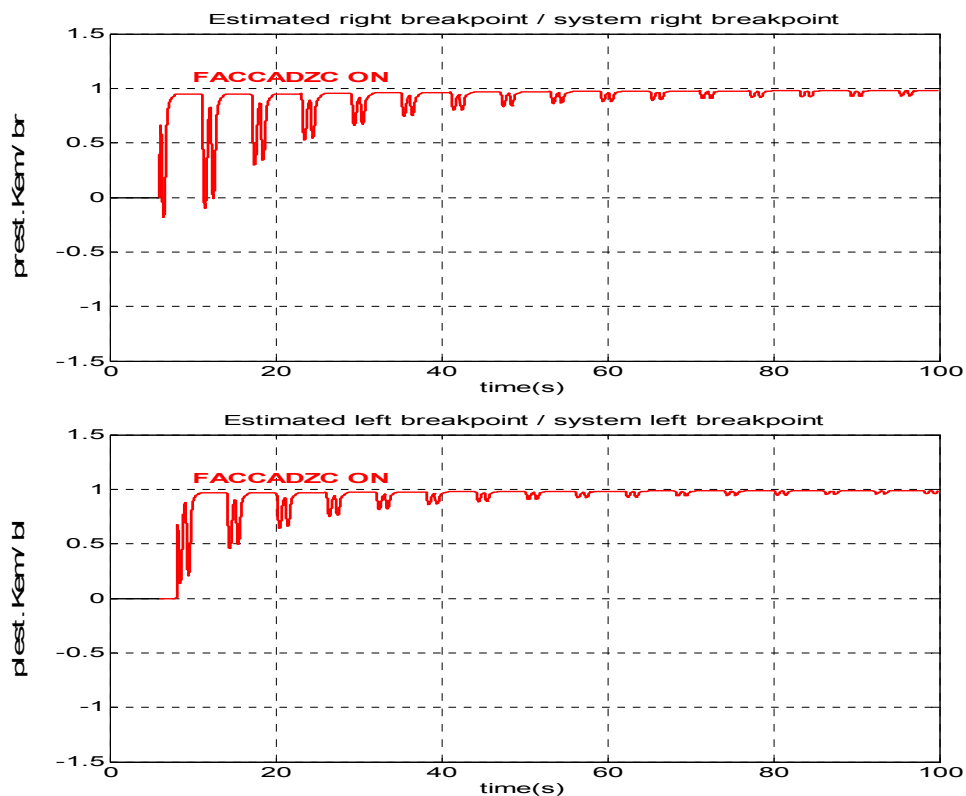
Figure 9 – Estimation of the dead-zone parameters

Although the influence of the adaptation gains for each adapted parameter was separated, it is not so straightforward, since the convergence of each parameter affects the convergence of the other parameters.

5. Conclusions

This work presented an adaptive cascade controller with an adaptive dead-zone compensation scheme, named FACCADZC. To obtain this controller, a cascade control algorithm with adaptive control laws designed to deal with parametric uncertainties in the mechanical and hydraulic subsystems was coupled with an algorithm that compensates the dead-zone by using the dead-zone inverse together with the adapted breakpoint values.

The advantage of the FACCADZC is that it can deal with a system with parametric uncertainties and with a dead-zone without the necessity of manual adjustments. Furthermore, the use of an overlapped valve (valve with a dead-zone) is normally less expensive then the use of a critical center valve. Simulation results illustrated the effectiveness of the proposed algorithm.

The developed algorithm represents one more step in the development of cascade controllers to deal with the limitations imposed by the hydraulic actuators characteristics for obtaining high closed-loop performance.

Future works are the inclusion of an adaptive friction compensation algorithm, a complete stability proof of the presented combination and the experimental implementation of the proposed controller.

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