A FIXED CASCADE CONTROLLER WITH AN ADAPTIVE DEAD-ZONE COMPENSATION SCHEME APPLIED TO A HYDRAULIC ACTUATOR

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Abstract

This paper addresses the position trajectory control of a hydraulic actuator by combining a fixed cascade controller including the valve dynamic with an adaptive dead-zone compensation scheme. Amongst the hydraulic actuators nonlinearities, the valve dead-zone is one of the main position error sources in the hydraulic actuator closed loop control. The dead-zone compensation scheme used here is based on the dead-zone inverse. As the dead-zone breakpoints depend on the system operation point and they vary for each valve, one uses an adaptation law to estimate those points, in such a way that the proposed algorithm does not need manual procedures to identify the dead-zone breakpoints. In addition, the adaptation scheme allows the system to work in different points of operation. These estimated breakpoints are also used in the dead-zone which has to be intentionally added to the valve spool position measured in the electronic card. Simulation results illustrate the main characteristics of the proposed scheme.

1 Introduction

Hydraulic actuators are widely used in industrial applications, mainly due to their ability to generate high forces (or torques) with small dimension actuators. Unfortunately, these actuators present some undesirable characteristics: lightly damped dynamics, highly nonlinear behaviour, difficulties in obtaining the parameter values, amongst others. Such undesirable characteristics limit the closed loop performance that can be obtained with a classical controller. In order to improve the hydraulic actuators closed loop performance, many different control techniques have been proposed in the literature. These work’s authors have been developing cascade controllers to overcome such limitations [2].

Amongst the hydraulic actuators nonlinearities, the valve dead-zone and the dry friction force are the main sources of position error in the hydraulic actuator closed loop control. The amount of dead-zone and friction force are inversely related to the components price. This work deals with the use of the fixed cascade controller (NFCC) proposed in [3] with an adaptive algorithm that compensates the dead-zone [8]; in such a way that it can work with low price proportional valves.

Dead-zone is a static input-output relationship which for a range of input values gives no output [8]. The range of a dead-zone is stated by its left and right breakpoints. If the breakpoints are known, one can cancel the valve dead-zone by using a dead-zone inverse [8]. Most of valve electronic cards have a dead-zone compensation scheme manually adjusted that is based on the dead-zone inverse. In [3], a fixed scheme similar to the used in electronic cards is described. Bu and Yao [1] propose a dead-zone compensation scheme by measuring the relationship between the valve spool position and the flowrate, approaching this flowrate curve by straight lines and using the inverse of that curve to compensate the valve dead-zone. These fixed schemes need experimental tests and are effective in a certain region around the point where the dead-zone compensation was adjusted. In order to overcome this problem, algorithms that on-line estimate the breakpoint values must be used [7,8].

In this work, one combines an adaptive scheme based on the adaptive dead-zone compensation algorithms proposed in [8] combined with NFCC [3]. By using this algorithm, manual adjusts are not needed.

In this work, section 2 presents the hydraulic actuator mathematical model; section 3 describes the fixed cascade controller (NFCC); section 4 addresses the adaptive dead-zone compensation scheme; section 5 discusses the simulation results and, in section 6, one presents the conclusions.

2 Hydraulic actuator mathematical model

Consider the hydraulic actuator shown in Fig. 1, where M represents the system total mass, B is the viscous friction coefficient, \( p_0 \) is the supply pressure, \( p_r \) 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, \( Q_f \) is the flowrate from the valve to chamber 1, \( Q_s \) is the flowrate from chamber 2 to the valve and \( u \) is the electrical voltage applied to the electronic card.

In this work, the valve under consideration is an overlapped four-way valve. In these kinds of valves, the lands of the spool are greater than the annular parts of the valve body, in
such a way that when the spool is displaced from the central position, there will be a region where there is not flowrate (dead-zone).

\[ \text{Figure 1 – Hydraulic Actuator} \]

Assuming the valve dynamic as a first order system, the hydraulic actuator mathematical model is given by

\[ \begin{align*}
\text{My} + By &= Ap_A, \\
\hat{p}_A &= -fA\dot{y} + fK_v g x_v, \\
x_v &= DZ_1(x_{vb}), \\
x_{vb} &= -\omega_v x_{vb} + K_{em}\omega_c u.
\end{align*} \tag{1} \tag{2} \tag{3} \tag{4} \tag{5} \]

where \( f = f(y) = \frac{\beta v}{(0.5v)^2 - (Ay)^2} \), \( x_v \) is the valve spool position, \( p_a = p_1 - p_2 \) is the cylinder chambers pressure difference, \( \beta \) is the bulk modulus, \( v = v_1 + v_2 \), \( K_v \) is the hydraulic constant, \( g = g(x_v, p_a) = \sqrt{p_a - \text{sgn}(x_v)p_a} \). \( K_{em} \) is the valve constant, \( \omega_v \) is the valve bandwidth, \( x_{vb} \) is the valve spool position before the dead-zone. The signal \( x_{vb} \) is the signal that is measured by an internal transducer in the valve and is available in the electronic card. The relationship between \( x_v \) and \( x_{vb} \) is given by

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

where \( b_r \) is the right breakpoint and \( b_l \) is the left breakpoint.

Figure 2 shows a block diagram of the Equations (3), (4) and (5).

\[ \text{Figure 2 – Valve with dead-zone block diagram} \]

3 New fixed cascade control (NFCC)

The NFCC was proposed in [3]. Here, it is briefly presented for completeness.

The cascade strategy consists in interpreting the hydraulic actuator mathematical model as two subsystems: a mechanical subsystem driven by a hydraulic one. From this interpretation, the control law is calculated in two steps [6]:

(i) Compute a control law \( p_{ad} \) (desired pressure difference) for the mechanical subsystem such that the output “\( y \)” tracks the desired trajectory \( y_d \) as close as possible;

(ii) Compute a control law “\( u \)” for the hydraulic subsystem such that \( p_a \) tracks \( p_{ad} \) as close as possible.

The NFCC mechanical subsystem control law is given by

\[ p_{ad} = \frac{1}{A} (M\ddot{y} + B\dot{y} - K_p x_v) \tag{6} \]

where \( \ddot{y} = y - y_d \) is the position trajectory tracking error, \( y_d = \ddot{y} - \lambda \dot{y} \), \( z = y - \ddot{y} = \dot{y} + \lambda \ddot{y} \) and \( K_p \) and \( \lambda \) are positive constants.

The control law \( u = u_{NFCC} \) for the hydraulic subsystem is given by

\[ u_{NFCC} = \frac{1}{K_{em}g} \left[ \dot{\tilde{p}}_A - \dot{\tilde{p}}_0 + \dot{\lambda} \right] \tag{7} \]

where \( \tilde{p}_A = p_A - p_{ad} \) is the pressure difference trajectory tracking error, \( x_{vb} \) is the desired spool position trajectory, \( \dot{\tilde{p}}_A = p_A - p_{ad} \) is the pressure difference trajectory tracking error and \( K_p, K_v, \phi_1, \phi_2 \) are positive constants.

Remark 1 – In [3], without considering the valve dead-zone, i.e. \( x_v = x_{vb} \), one demonstrates that the closed loop system \( \{1\}(2)(4)(6)(7)(8) \) is exponentially stable when the system parameters are known. A methodology to tune the controller’s gains was presented in [5].

Remark 2 – If the valve has a dead-zone, \( \tilde{p}_A \) does not tend to zero, i.e. there is a trajectory tracking error in the hydraulic subsystem and, consequently, the position trajectory tracking error \( \ddot{y} \) will tend to a bounded set.

4 Adaptive dead-zone compensation

In order to propose the schemes to compensate the dead-zone and to read the valve spool position, one assumes a static relation between \( x_v \) and \( x_{vb} \), such that \( x_v = K_{em}u \). Therefore, Equations (3), (4) and (5) can be written as

\[ x_v = DZ_1(K_{em}u) = DZ_2(u) \tag{9} \]

\[ x_v = DZ_2(u) = \begin{cases} K_{em}u - b_r, & u > b_r \\
0, & b_l \leq u \leq b_r, \\
K_{em}u - b_l, & u < b_l \end{cases} \tag{10} \]

A dead-zone inverse (DZI)[8] that can be used to compensate the valve dead-zone is given by
K is different from zero, the estimated parameters and $u$ in the tends to a residual set NFCC included as shown in figure 4 [3]. Thus, in order to obtain the signal $x_v$, Figure 3 illustrates the proposed scheme compensation in Equation (11). As it was mentioned before, the valve spool position that is read in the electronic card is the signal $x_v$. Thus, in order to obtain the signal $x_v$ that is used in the control law $u_{NFCC}$, a dead-zone equal to the valve dead-zone must be included as shown in figure 4 [3].

4.1 Adaptation law for the dead-zone breakpoints

The dead-zone breakpoints can be approximated by using experimental tests. However, besides the necessity to adjust the compensation scheme for each valve, the values of $b_l$ and $b_r$ vary according to the operation point (temperature, pressure, amongst others). To overcome this problem, one proposes an adaptive algorithm based on an adaptation law proposed in [8]. In the proposed scheme, the adaptation and control laws are respectively given by

$$u = DZI(u_{NFCC}) = \begin{cases} p_r + u_{NFCC}, & u_{NFCC} > 0 \\ 0, & u_{NFCC} = 0 \\ p_l + u_{NFCC}, & u_{NFCC} < 0 \end{cases}$$

(11)

where $p_r = \frac{b_l}{K_{em}}$ and $p_l = \frac{b_r}{K_{em}}$. Substituting Equation (11) into Equation (10), one obtains

$$x_v = K_{em} u_{NFCC}$$

(12)

Figure 3 illustrates the proposed scheme compensation in Equation (11). As it was mentioned before, the valve spool position that is read in the electronic card is the signal $x_v$. Thus, in order to obtain the signal $x_v$ that is used in the control law $u_{NFCC}$, a dead-zone equal to the valve dead-zone must be included as shown in figure 4 [3].

$$u_{NFCC} = \begin{cases} p_r + u_{NFCC}, & u_{NFCC} > 0 \\ 0, & u_{NFCC} = 0 \\ p_l + u_{NFCC}, & u_{NFCC} < 0 \end{cases}$$

(13)

$$\dot{p}_r = \begin{cases} \eta_1 u_{NFCC} \dot{p}_A, & 0 < u_{NFCC} \leq p_{lmax} \\ 0, & \text{otherwise} \end{cases}$$

(14)

$$\dot{p}_l = \begin{cases} \eta_2 u_{NFCC} \dot{p}_A, & p_{lmin} \leq u_{NFCC} < 0 \\ 0, & \text{otherwise} \end{cases}$$

(15)

where $\eta_1$, $\eta_2$ and $p_{lmax}$ are positive constants and $p_{lmin}$ is a negative constant.

The adaptation laws, Equations (14) and (15), depend on the signals $\dot{p}_A$ and $u_{NFCC}$. From remarks 1 and 2, one concludes that when the hydraulic actuator parameters are known, the hydraulic subsystem trajectory error $\dot{p}_A$ tends to a residual set due to the valve dead-zone. In addition, if the direct inverse dead-zone is implemented with the true values $p_l$ and $p_r$, the dead-zone is completely compensated [8]. Therefore, while $\dot{p}_A$ is different from zero, the estimated parameters $\dot{p}_r$ and $\dot{p}_l$ are different from $p_l$ and $p_r$. It justifies the use of $\dot{p}_A$ in the proposed adaptation laws. The use of $u_{NFCC}$ is to consider the sign of $u_{NFCC}$ and to decrease the adaptation rate when the valve is closing.

4.2 Smoothing the dead-zone inverse

When the estimated breakpoints approach to the true values, the control signal tends to present high-frequency components. Although the valve works like a filter, this high-frequency signal can excite unmodelled dynamics and can also cause actuator vibration.

To overcome this problem, one can smooth the dead-zone inverse by using trigonometric functions. Another way to change Equations (13), (14) and (15) by including a small dead-zone to avoid excessive switching:

$$u = \hat{DZI}(u_{NFCC}) = \begin{cases} \hat{p}_r + u_{NFCC}, & u_{NFCC} > l_s \\ 0, & l_s \leq u_{NFCC} \leq l_b \\ \hat{p}_l + u_{NFCC}, & u_{NFCC} < l_b \end{cases}$$

(16)

$$\dot{p}_r = \begin{cases} -\eta_1 u_{NFCC} \dot{p}_A, & l_s < u_{NFCC} \leq p_{rmax} \\ 0, & \text{otherwise} \end{cases}$$

(17)

$$\dot{p}_l = \begin{cases} \eta_2 u_{NFCC} \dot{p}_A, & p_{lmin} \leq u_{NFCC} < l_b \\ 0, & \text{otherwise} \end{cases}$$

(18)

where $l_b$ is a negative constant and $l_s$ is a positive constant. In both alternatives, smoothing the control signal yields to position errors greater than the errors when a dead-zone direct inverse is used.

4.3 Obtaining the valve spool position

To obtain the valve spool position signal $x_v$ that is used in the control law, one uses the same scheme proposed in figure 4, with $b_l$ and $b_r$ substituted by

$$\hat{b}_r = K_{em} \hat{p}_r$$

(19)

$$\hat{b}_l = K_{em} \hat{p}_l$$

(20)

5 Simulation results

In this section, the simulation results of the closed loop system with the NFCC with the dead-zone adaptive
compensation are presented. Initially, the dead-zone is not compensated. Then, after 12 seconds the adaptive compensation scheme is turned on.

The system parameters [4] are assumed to be known: $M = 20.66 \text{ Kg}$, $B = 316.2 \text{ Ns/m}$, $\beta = 10^7 \text{ Pa}$, $p_s = 100 \times 10^5 \text{ Pa}$, $\lambda = 7.6576 \times 10^4 \text{ m}^2$, $|u|_{\max} = \pm 10 \text{ V}$, $v = 9.5583 \times 10^5 \text{ m}$, $A = 7.6576 \times 10^{-4} \text{ m}^2$, $\omega_v = 147 \text{ rad/s}$ and $K_{em} = 0.76$. The controller gains are $K_p = 500$, $K_v = 90$, $K_d = 11000$ and $\lambda = 30 [4]$. The valve dead-zone was set with $b_r = 0.5 \text{ V}$ (5%) and $b_l = -0.8 \text{ V}$ (8%). The adaptation gains $\eta_1 = \eta_2 = 5 \times 10^{-6}$ were adjusted in simulations and $p_{r\max} = 2 \text{ V}$ and $p_{l\min} = -2 \text{ V}$.

The desired trajectory $y_d$ is based on a 7th order polynomial, Equation (21), and on straight lines and is given by Equation (22) [2,4]. Figure 5 illustrates the desired trajectory.

$$y_{dl}(t) = \begin{cases} y_{d1}(t) & \text{if } t < 1 \\ 0.3 & \text{if } 1 \leq t \leq 2 \\ -y_{d1}(t-2) + 0.3 & \text{if } 2 < t < 3 \\ -y_{d1}(t-3) & \text{if } 3 \leq t \leq 4 \\ -0.3 & \text{if } 4 < t < 5 \\ y_{d1}(t-5) - 0.3 & \text{if } 5 \leq t \leq 6 \end{cases} (21)$$

$$y_d(t) = \begin{cases} y_{d1}(t) & \text{if } t < 1 \\ 0.3 & \text{if } 1 \leq t \leq 2 \\ -y_{d1}(t-2) + 0.3 & \text{if } 2 < t < 3 \\ -y_{d1}(t-3) & \text{if } 3 \leq t \leq 4 \\ -0.3 & \text{if } 4 < t < 5 \\ y_{d1}(t-5) - 0.3 & \text{if } 5 \leq t \leq 6 \end{cases} (22)$$

Figure 6 shows the system response when the dead-zone adaptive inverse is used. Note that when the compensation adaptive scheme is turned on, the trajectory tracking error $\tilde{y}$ tends to zero. However, the control signal presents high-frequency components that can cause vibrations in the actuator.

Figure 7 shows the system response when the inverse dead-zone is smoothed by Equations (16), (17) and (18) with $l_r = 0.05$ and $l_l = -0.05$. One observes that in this case the trajectory tracking error decreases when the compensation starts to work, but it does tend to zero. In practice, the values of $l_r$ and $l_l$ must be adjusted so that there are not vibrations in the actuator.

Simulation results (not showed here) considering parametric uncertainties showed an increase in the trajectory tracking error. At the moment, a cascade controller taking into account those uncertainties is underdevelopment.
6 Conclusions

In this work, a fixed cascade controller (NFCC) was combined with an adaptive dead-zone compensation scheme. Simulation results showed that when the hydraulic actuator parameters are known, the proposed scheme yields to null trajectory tracking error and yields to small errors when the dead-zone inverse is smoothed. With this desired trajectory, the estimated dead-zone breakpoints tended to the true values, showing that the employed scheme does not need previous or on-line manual adjustments.

Future works include the theoretical proof and experimental implementation of the proposed combination and the development of an algorithm that takes into account the parametric uncertainties in the hydraulic and mechanical subsystems.

References


