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TRACKING CONTROL IN HYDRAULIC ACTUATORS USING SLOW PROPORTIONAL DIRECTIONAL VALVES

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Abstract. *This paper discusses the influence of the proportional directional valve dynamics in the design of tracking controllers for hydraulic actuators. Particularly, its influence on the behavior of the closed loop is addressed considering linear (P, PI and State Controller) and non-linear (Fixed Cascade Controller-CC) controllers. By considering a fifth-order model to represent the actuator dynamics, it is shown, through a theoretical analysis, that when the hydraulic actuator dynamics is dominated by the valve dynamics with a strongly damped characteristic, the closed loop system damping is increased. Furthermore, the linear and non-linear controller gains are limited by the damped valve poles. In this regard, we conclude that, using linear controllers, the best closed loop performance is achieved using a simple proportional (P) controller. Our analysis outlines that the contribution of the integral term in a proportional-integral (PI) controller to the closed loop performance is very small. Also, it was found that the use of a state control does not improve the system performance significantly. This is mainly because the system is dominated by the valve dynamics and so, using the velocity and acceleration of the cylinder as state variables, the feedback of these variables can not be used to change the location of the valve poles. The analysis of the Fixed Cascade Controller shows that using this strategy the closed loop results in the best performance observed for the controllers investigated in this study. However, it has some disadvantages, such as sensitivity to unknown external disturbances and to supply pressure drops. This analysis allows some controller design guidelines to be established. All the theoretical results are confirmed by experimental results, using a test rig where it is possible apply external forces and variable loads simulating real situations encountered by hydraulic actuators in the industrial field.*

Keywords: *hydraulic control, cascade strategy, valve dynamics, disturbance influence*

1. INTRODUCTION

Hydraulic actuators are commonly used in applications that demand high levels of power. The main advantage of these components is their high power/size ratio, which is attractive when high levels of power are necessary and the available space is small. However, these actuators have some undesirable characteristics, such as variable natural frequency, lightly damped dynamics and highly nonlinear behavior, which hinder their closed loop control in applications that demand high performance.

According to Virvalo (2002), the main goal of designing a controller for a hydraulic actuator is to increase the damping. In Virvalo (2002), the author uses a fast servo valve (ω_s=100 Hz) to control the position of a hydraulic cylinder and concludes that the state controller allows this goal to be achieved by feeding back the position, the velocity and the acceleration of the cylinder.

But when the hydraulic actuator has a proportional directional valve with slow and damping dynamics, the closed loop system damping is increased. This reduces the oscillation in the actuator closed loop response and brings some peculiarities to the controller design. In this regard, one of the particularities is that the controller gains are limited by the valve poles and that the system dynamic is dominated by the valve.

In this paper, linear (P, PI and State Controller) and non-linear (Fixed Cascade Controller-CC) controllers are tested in a hydraulic actuator with a slow proportional directional valve. The objective is to show how the valve dynamics influences the controller design and the position and trajectory tracking control of a hydraulic actuator. All the theoretical analysis is carried out using linear and non-linear fifth-order models, where the parameters are easily obtained in catalogues. The experiments were carried out in a rig test where it is possible to apply external forces and variable loads, simulating real situations encountered by hydraulic actuators in the industrial field.

Section 2 describes the test rig and the system parameters. In section 3, based on the test rig components, the linear and non-linear fifth-order models are presented, considering the valve dynamics as a second-order system. The controller designs and their theoretical analysis are given in sections 4 and 5. Details regarding the experimental implementation are shown in section 6. The experimental results are presented in section 7. In section 8, the conclusions are outlined.

2. TEST RIG

The test rig used in this research is shown in Fig. 1. It is composed of a double action differential cylinder; a two stage asymmetrical proportional directional valve, where the first stage is a proportional solenoid pressure reducing valve and the second stage is a 4/3 hydraulically operated proportional directional valve; variable masses; one spring with variable pre-load to represent the action of an external disturbance; one transmitter and two pressure transducers and one position transducer.

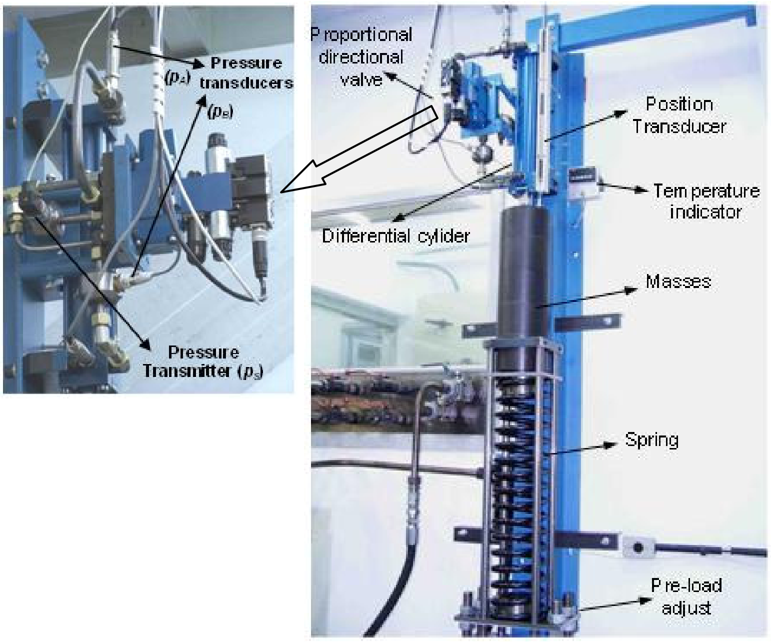

Figure 1. Test rig

The system parameters and specifications are:

- Cylinder: 63/45-400 (piston diameter/rod diameter-stroke, [mm]), natural frequency (ω_{cyl}) = 405 rad/s.
- Valve characteristics: 85 L/min nominal flow with a 10 bar valve pressure drop, flow ratio (2:1), input signal range= ±10 V, bandwidth (ω_v) =90 rad/s, natural damping (ξ_v) = 1, flow coefficients Kv_A=2E-6 $m^3/(sPa^{0.5})$ and Kv_B=1E-6 $m^3/(sPa^{0.5})$, flow gains Kq_{UA}=3.79E-4 $m^3/(sV)$ and Kq_{UB}=1.89E-4 $m^3/(sV)$, flow-pressure gains Kc_A=1.58E-10 $m^5/(sN)$ and Kc_B=9.24E-11 $m^5/(sN)$.
- Spring characteristics: elastic constant (K_m)=27560 N/m, free length (L_0)=790 mm, block length (L_{Bl}) = 321 mm.
- Data acquisition and control board: DSPACE DS1104.
- Fluid temperature=40º C ± 2º C; Fluid viscosity=32 cSt; Total mass=108.5 kg; Viscous friction coefficient (B) = 488 Ns/m; Effective bulk modulus (β_e) = $0.8*10^9$ Pa; Total volume (V_T) = $1*10^{-3}$ m^3.
- Position transducer accuracy = ± 0.23 mm.

Due to its construction characteristics, the proportional directional valve (PDV) has a significant dead-zone and a relatively small bandwidth, around 90 rad/s. Moreover, as the PDV does not have spool position feedback, the valve hysteresis is high (hysteresis ≤ 6%).

A hydraulic power unit (HPU), not shown in Fig. 1, is used to supply a fluid with constant temperature and supply pressure to the actuation system (hydraulic actuator). Since the pressure control valves in the HPU, responsible for controlling the supply pressure, do not have pressure feedback, there occurs a significant pressure drop when the cylinder piston moves (Schwartz, 2004). Moreover, the pressure control valves have a high time response, increasing the pressure drop in the initial instants of the cylinder piston displacement.

The supply pressure drop hinders the hydraulic actuator performance in the trajectory tracking control. In the absence of an accumulator, it is necessary work with small velocities in the actuation system to slow down the pressure variation (Pereira, 2006). In this way, the pressure control valves are able to maintain the supply pressure approximately constant.

3. HYDRAULIC ACTUATOR MATHEMATICAL MODEL

The model of the hydraulic actuator is based on the schematic drawing shown in Fig. 2, where p_T is the return pressure, p_S is the supply pressure, p_A is the pressure in line A, p_B is the pressure in line B, q_{VA} is the flow in line A, q_{VB} is the flow in line B, A_A is the cylinder piston area, A_B is the cylinder cross-section area, F_L represents the spring force (F_L=$K_m y$), F_G is the gravitational force (F_G=mg), F_{at} is the viscous friction ($F_{at} = B\dot{y}$), x_v represents the spool displacement and y is the cylinder piston position. In what follows it is considered that the hydraulic power unit (HPU) delivers constant supply pressure p_S .

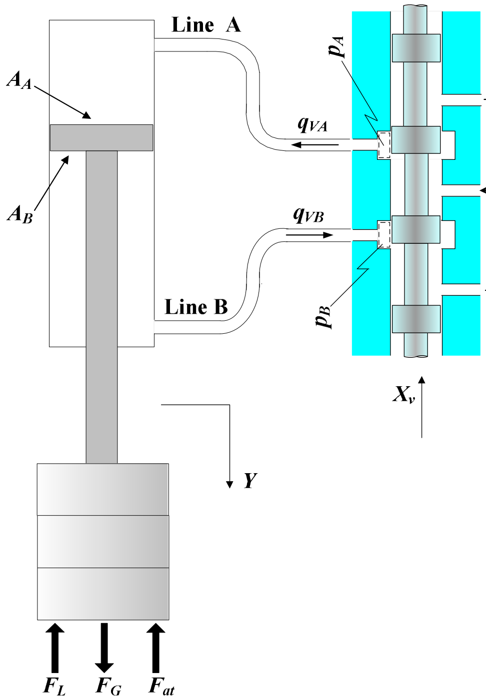

Figure 2. Hydraulic actuator schematic drawing

In sections 3.1 and 3.2 a non-linear and a linear fifth-order model, considering the valve dynamics as a second order system, are presented, respectively. These models are used in the design and analysis of linear (P, PI and State Controller) and non-linear (Fixed Cascade Controller-CC) controllers.

3.1. Non-linear model

Considering the valve dynamics as a second order system, the hydraulic actuator non-linear model is given by (Pereira, 2006):

$$\ddot{U}_c = \omega_v^2 U_v - 2\xi_v \omega_v \dot{U}_c - \omega_v^2 U_c \tag{1}$$

$$\ddot{y} = \frac{1}{m}\cdot(F_H - F_{at} - F_L + F_G) \tag{2}$$

$$\dot{F}_H = \frac{A_A \beta_e}{V_A}\cdot q_{VA} + \frac{A_B \beta_e}{V_B}\cdot q_{VB} - \left(\frac{A_A^2 \beta_e}{V_A} + \frac{A_B^2 \beta_e}{V_B}\right)\cdot \dot{y} \tag{3}$$

where Eqs. (1), (2) and (3) represent, respectively, the second-order valve dynamics, the resultant acceleration from a hydraulic force and the hydraulic force variation. The parameter U_v is the valve input signal in volts, U_c is the spool displacement in volts, ω_v is the valve bandwidth, ξ_v is the valve natural damping, m is the total mass and F_H is the hydraulic force ($F_H = p_A A_A - p_B A_B$). The cylinder chamber volumes, $V_A = V_{A0} + A_A y$ and $V_B = V_{B0} - A_B y$, depend on the initial volumes V_{A0} and V_{B0} and on the cylinder piston position (y). The flows in the valve notches q_{VA} and q_{VB} can be given by (Furst, 2001):

- For $x_v \geq 0$: $q_{VA} = Kv_A \cdot \frac{U_c}{U_{cn}} \cdot \sqrt{p_S - p_A}$ and $q_{VB} = Kv_B \cdot \frac{U_c}{U_{cn}} \cdot \sqrt{p_B - p_T}$ (4)

- For $x_v \leq 0$: $q_{VA} = Kv_A \cdot \frac{U_c}{U_{cn}} \cdot \sqrt{p_A - p_T}$ and $q_{VB} = Kv_B \cdot \frac{U_c}{U_{cn}} \cdot \sqrt{p_S - p_B}$ (5)

It should be noted that this flow representation is very practical, since the partial flow coefficients (Kv_A and Kv_B) can be easily obtained in the valve catalogue as shown in Eq. (6):

$$Kv_A = \frac{q_{VAn}}{\sqrt{\Delta p_{An}}} \quad \text{and} \quad Kv_B = \frac{q_{VBn}}{\sqrt{\Delta p_{Bn}}} \tag{6}$$

where q_{VAn} is the nominal flow and Δp_{An} is the nominal pressure drop in the valve A notch and q_{VBn} and Δp_{Bn} are, respectively, the nominal flow and the nominal pressure drop in the valve B notch.

Although they are considered constant, in practice the partial flow coefficients (Kv_A and Kv_B) have variations for valve opening of 0 to 50%. For valve openings between 60% and 100%, the variations in the partial flow coefficients values are small and can be considered constant (Valdiero, 2005). This characteristic supports findings that the proportional directional valves have a higher non-linear behavior in small opening ranges, since the leakage and the dead-zone of the valve are more significant in this work range (Valdiero, 2005; Virvalo, 1999). Furthermore, the smaller the valve opening, the lower the cylinder piston velocity and, consequently, the more significant will be the non-linear friction behavior influence (Stribeck curve) (Valdiero, 2005).

3.2. Linear model

Linearising the system around the point where $V_A = V_B = V_T/2$, $U_c = 0$ and considering F_G=0, F_L=0, one obtains the hydraulic actuator fifth-order transfer function relating the control input U_v and the cylinder piston position y (Pereira, 2006):

$$G(s) = \frac{\omega_v^2}{s^2 + 2\xi_v \omega_v s + \omega_v^2} \cdot \frac{\dfrac{2\beta_e}{V_T m}\left(Kq_{UA}A_A + Kq_{UB}A_B\right)}{s\left[s^2 + \left(\dfrac{B}{m} + \dfrac{2\beta_e Kc_p}{V_T}\right)s + \dfrac{2\beta_e (BKc_p + A_A^2 + A_B^2)}{V_T m}\right]} \tag{7}$$

where the first term represents the valve dynamics, the second term represents the cylinder dynamics and $Kc_P=(Kc_A+Kc_B)/2$ is the partial flow-pressure gain.

The non-linear and linear models presented in this section will be used below.

4. LINEAR CONTROLLERS

In this section we present the design of the proportional (P), proportional integral (PI) and state controllers in the position and trajectory tracking control, using the hydraulic actuator transfer function presented in section 3.2 and the parameter values shown in section 2.

4.1. Proportional controller

In the P controller, the position error is multiplied by a proportional gain (*KP*). The structure block diagram of the closed loop system, using a P controller, is shown in Fig. 3.

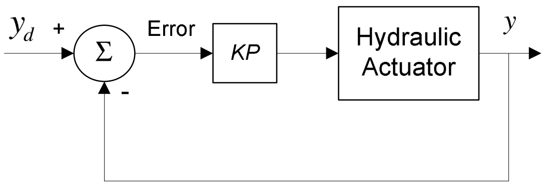

Figure 3. Proportional controller block diagram

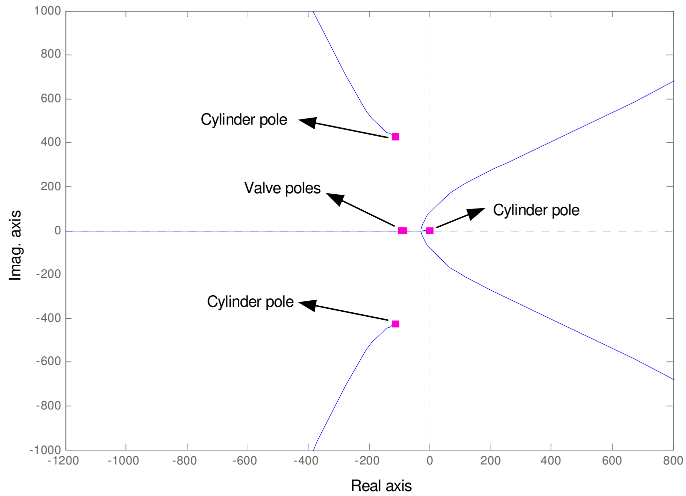

Figure 4. Root locus in relation to *KP* using the P controller

Using the proportional controller, the system closed loop root locus is given in Fig. 4, where it is possible to observe the presence of one pole at the origin, two complex poles and two poles on the negative real axis. Varying the proportional gain (*KP*), the two complex poles move to the left, while the pole at the origin and one of the poles on the negative real axis, initially move towards each other and then go to the right, in the direction of imaginary axis. The other pole on the negative real axis moves to the left.

As shown in Fig. 4, the system stability depends on the valve poles. In other words, the value of *KP* that ensures the system stability is limited by the valve poles. The faster these poles, the higher the value of *KP* that can be used and vice versa.

As the proportional directional valve has a damped behavior and dominates the system dynamics, the system damping is also increased, and the transient periods are less oscillatory. This allows the use of higher gains without causing oscillations or overshoots in the system output.

4.2. Proportional integral controller

In the PI controller, an integral term is added to reduce the steady-state error. This integral term adds one pole at the origin and one zero on the negative real axis near to the origin in the closed loop system. So, the system dynamics is not substantially changed with respect to the proportional controller. The objective of this control strategy is, even with the *KP* variation, to keep one pole close to the origin to actuate at slow frequencies and reduce the steady-state error. But using this control strategy, the transient period can be adversely affected when the system suffers a disturbance (Franklin *et al.*, 1995; Pereira, 2006).

4.3. State controller

When a good performance is not achieved by adjusting the proportional and integral terms, one of the solutions is to use a state control. In the state control, the state variables are fed back in order to assign a set of pole locations for the closed loop system that will correspond to a satisfactory dynamics response in terms of rise time and other measures of transient response (Franklin *et al.*, 1995).

In the hydraulic actuator it is possible to feedback the position, the velocity and the acceleration $([y \quad \dot{y} \quad \ddot{y}])$ or the position, the velocity and the hydraulic force $([y \quad \dot{y} \quad F_H])$. When the hydraulic force is used as a state variable, the system become more sensitive to external disturbances (F_G, F_L), while using the acceleration, the system will be more robust to these disturbances (Cunha, 2001).

In applications where the valve bandwidth is at least 3 times faster than the cylinder natural frequency, the system is dominated by the lightly damped cylinder poles and has an oscillatory behavior in the closed loop. For this situation, it is possible to achieve good results in a position and trajectory tracking control raising the system damping through a state controller that feeds back the position, the velocity and the acceleration of the cylinder (Virvalo, 2002).

However, in cases where the system dynamics is dominated by the valve poles, the performance of the closed loop with a state controller that feeds back the cylinder state variables is reduced, since its poles are not dominant (Pereira, 2006). To change the system dynamics, it is necessary to feed back the valve state variables (spool position and velocity).

5. FIXED CASCADE CONTROLLER (CC)

The fixed cascade controller (CC) is a non-linear control strategy that considers the hydraulic actuator as two interconnected subsystems: a mechanical subsystem driven by a hydraulic one. The idea is to promote a fast loop in the hydraulic subsystem in order to generate a force in the hydraulic subsystem that allows the mechanical subsystem to track the desired trajectory (Guenther and De Pieri, 1997).

The control law for the mechanical subsystem is based on Slotine and Li (1987) and is given by

$$F_{Hd} = m\ddot{y}_r - K_D z + K_m y - mg + B\dot{y} \tag{8}$$

$$\dot{y}_r = \dot{y}_d - \lambda\tilde{y}\,,\ \ \tilde{y} = y - y_d\,,\ \ z = \dot{y} - \dot{y}_r = \dot{\tilde{y}} + \lambda\tilde{y} \tag{9}$$

where F_{Hd} is the desired hydraulic force, $\tilde{y}$ is the position trajectory tracking error, $\dot{y}_r$ is a reference velocity, z is a measure of the velocity error, and K_D>0 and λ >0 are the mechanical subsystem control law gains.

The control law for the hydraulic subsystem uses the feedback linearization method based on Slotine and Li (1991), and can be written as

$$U_v = \frac{1}{A_A f_A \dfrac{Kv_A}{U_{vn}} g_{A0} + A_B f_B \dfrac{Kv_B}{U_{vn}} g_{B0}} \cdot \left[\dot{F}_{Hd} - K_P \tilde{F}_H + \left(A_A^2 f_A + A_B^2 f_B\right) \cdot \dot{y}\right] \tag{10}$$

where K_P is the hydraulic subsystem positive gain, $\dot{F}_{Hd}$ is the time derivative of the desired hydraulic force, $\tilde{F}_H = F_H - F_{Hd}$ is the hydraulic force error, $f_A=\beta_e/(V_{A0}+A_A y)$, $f_B=\beta_e/(V_{B0}-A_B y)$ and

$$g_{A0} = \begin{cases} \sqrt{p_S - p_A}\ , \text{para } x_v \geq 0 \\ \sqrt{p_A - p_T}\ , \text{para } x_v < 0 \end{cases} \qquad g_{B0} = \begin{cases} \sqrt{p_B - p_T}\ , \text{para } x_v \geq 0 \\ \sqrt{p_S - p_B}\ , \text{para } x_v < 0 \end{cases} \tag{11}$$

A block diagram displaying the CC control structure is shown in Fig. 5.

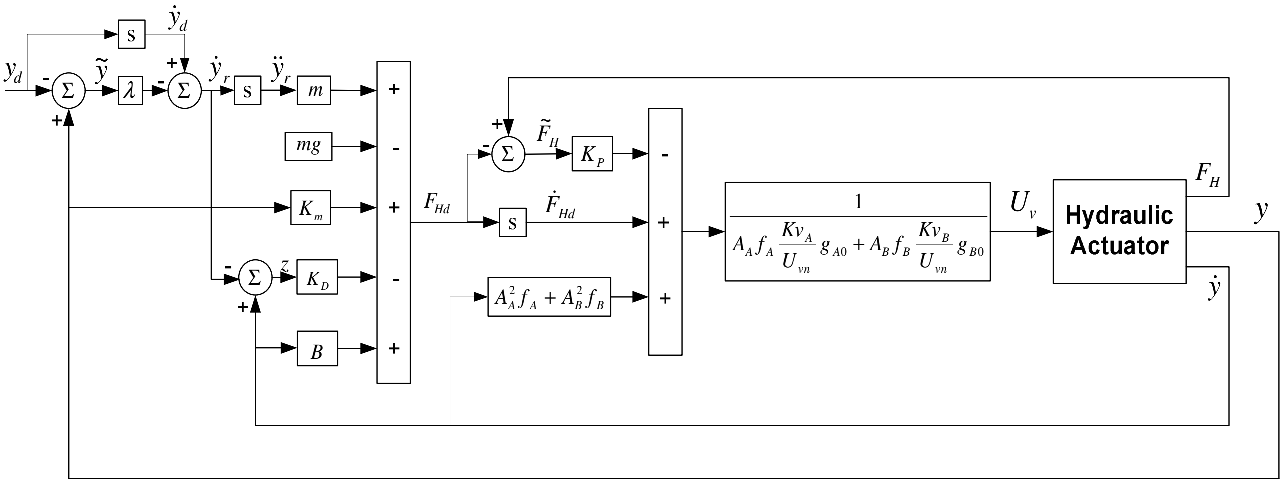

Figure 5. CC control structure block diagram

When all the system parameters and external disturbances are known, the exponential stability of the whole system can be demonstrated by using the Lyapunov's direct method. In cases where there are uncertainties in the system parameters and in the external disturbances, the position and the trajectory tracking error go to a bounded region that can be decreased by increasing the CC controller gains (Cunha *et al.*, 1997; Pereira, 2006). In Cunha *et al.* (2002) some guidelines to tuning the CC controller gains by analyzing the closed loop performance, are recommended.

However, when the valve dynamics is not compensated in the CC control law, because of the absence of a valve spool position transducer, the CC controller gain K_P is limited. This occurs because the valve dynamics introduces a third-order transfer function between F_H and F_{Hd}, as can be seen in Eq. (12), where $0 < K_P < 2\xi_v\omega_v$ to guarantee the system stability. This causes an error in the hydraulic subsystem tracking control that reflects in the mechanical subsystem performance (Pereira, 2006). The faster the valve bandwidth (ω_v), the higher the value of K_P which can be used and consequently the dynamics between F_H and F_{Hd} will be faster and the error in the hydraulic subsystem will decrease.

$$F_H(s) = \frac{\omega_v^2 (s + K_P)}{s^3 + 2\xi_v \omega_v s^2 + \omega_v^2 s + K_P \omega_v^2} \cdot F_{Hd}(s) \tag{12}$$

Another important characteristic in the CC controller is its sensitivity to a supply pressure drop. This occurs because the supply pressure is considered constant in the design of the CC controller and sometimes, in real applications, it can oscillate reducing the CC performance. The supply pressure drop affects directly the hydraulic subsystem, hindering the tracking control of the desired hydraulic force. Consequently, this drop affects the mechanical subsystem, increasing the trajectory tracking error. Equation (13) shows the influence of the supply pressure drop in the hydraulic force error (Pereira, 2006):

$$\tilde{F}_H(s) = \frac{W_{pert}(s)}{s + K_P W(s)} \tag{13}$$

where,

$$W = \frac{A_A f_A \dfrac{Kv_A}{U_{vn}} g_A + A_B f_B \dfrac{Kv_B}{U_{vn}} g_B}{A_A f_A \dfrac{Kv_A}{U_{vn}} g_{A0} + A_B f_B \dfrac{Kv_B}{U_{vn}} g_{B0}} < 1 \quad (14)$$

$$W_{pert} = (W-1) \cdot \left[\dot{F}_{Hd} + \left(A_A^2 f_A + A_B^2 f_B \right) \cdot \dot{y} \right] \quad (15)$$

When the supply pressure drops, $g_{A0} > g_A$ and $g_{B0} > g_B$. In this case g_{A0} and g_{B0}, which are present in the hydraulic subsystem control law, are calculated considering the supply pressure constant, while g_A and g_B consider the real pressure variation. The higher the supply pressure drop is, the lower the value of W will be. This increases the absolute value of W_{pert}, making the $\tilde{F}_H(s)$ convergence slower.

In Pereira (2006), it is shown that the CC controller is more sensitive to the supply pressure drop when the system volume is small. Due to the cylinder asymmetries, the more recoiled it is, more sensitive the system becomes, because in this region the system natural frequency is high, making the CC controller generate a small control signal. When the cylinder is forward the system volume is high, but its natural frequency is small and to compensate this characteristic a higher control signal is generated. It can be concluded, that when the control signal has high values, the system becomes more robust to the supply pressure drop.

6. EXPERIMENTAL IMPLEMENTATION

In all the tests a seventh-order polynomial was used as the desired trajectory, because in the CC control law the first-, second- and third-order derivatives of the position must be continuous. Figure 6 shows the desired trajectory, where the 0 position refers to the point where the cylinder is centered. To follow this desired trajectory the proportional directional valve opens about 50% (±5 Volts).

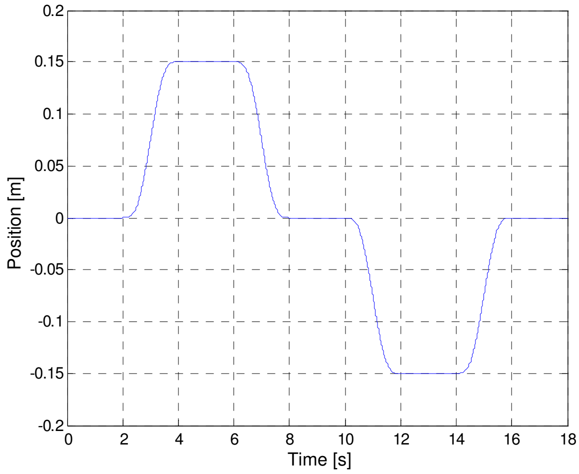

Figure 6. Desired trajectory

All the measured signals (pressures, position, velocity and acceleration) are filtered using first-order low-pass filters to decrease the noise originated in the numerical derivation and/or electromagnetic interference. The filter bandwidths have to be selected considering the necessity for a noise reduction without influencing significantly the system dynamics, because both characteristics limit the control gains and, consequently, the closed loop performance. The filter bandwidths used are: ω_{fp}=200 rad/s (position), ω_{fv}=50 rad/s (velocity), ω_{fa}=40 rad/s (acceleration), ω_{fpa}=300 rad/s (pressure p_A), ω_{fpb}=300 rad/s (pressure p_B), ω_{fps}=50 rad/s (pressure p_S).

The sample period is 1 ms, and consequently, the sample frequency is 6283 rad/s.

The controllers gains values used during the tests are limited by the noise in the control signal. The objective is to achieve the smallest position and trajectory tracking error, without overshoot in the system output and with a low-noise control signal. To achieve this, a maximum noise amplitude of 0.1 Volts in the control signal is considered acceptable (1% of the valve nominal tension U_n= ±10V). This avoids steady state oscillations in the cylinder position and increases the valve life.

The controllers gains used in the experimental tests are: P controller (KP=300), PI controller (KP=300, $1/T_I$=0.1), State controller $[y \quad \dot{y} \quad \ddot{y}]$ (KP=300, KV=1.77, KA=0.03), State controller $[y \quad \dot{y} \quad F_H]$ (KP=300, KV=0.3, KF_H=2.81E-4), CC controller (K_P=170, K_D=12000, λ=250).

In the CC controller implementation the flow coefficients (Kv_A and Kv_B) are adjusted to Kv_A = 1.2E-6 [$m^3/(sPa^{0.5})$] and Kv_B=0.8E-6 [$m^3/(sPa^{0.5})$], since these new values give the best fit to a 50% valve opening (±5 Volts).

The supply pressure and the fluid temperature are adjusted to 5 MPa (50 bar) and 40º C, respectively.

The valve dead-zone is identified by analyzing the pressure behavior and in all the tests it is compensated based on the methodology developed by Valdiero (2005).

In all the tests there is a total mass of 108.5 kg attached in the cylinder and the position error is given by $\tilde{y} = y_d - y$.

7. EXPERIMENTAL RESULTS

This section presents the experimental results obtained with P, PI, state and CC controllers. Firstly, these controllers are tested without the presence of external loads (spring force). The results are analyzed and the controllers that have the best performance are tested under the spring force.

Figure 7 shows the experimental results obtained with the P and PI controllers. The experimental results obtained with the state controller that feed backs the position, the velocity and the acceleration and the state controller that feed backs the position, the velocity and the hydraulic force, are shown in Fig. 8. The results shown in Figs. 7 and 8 correspond to the case without external force (spring force).

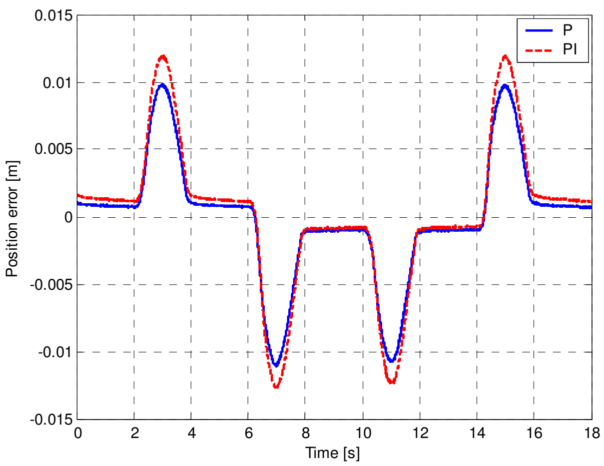

Figure 7. Responses to P and PI controller, without external load.

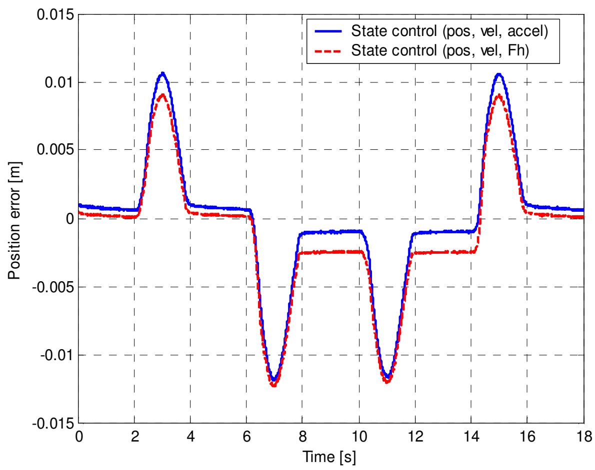

Figure 8. Responses to state controller: [pos, vel, accel] and [pos, vel, F_H], without external load.

Analyzing the experimental results of these four controllers, it can be observed that the P controller, in general, has the best performance. The results obtained with the PI controller, when compared to P controller, show a higher error during the trajectory tracking control, and during the steady state the convergence of the position error is very slow. If the integrative term is increased to accelerate the position error convergence during the steady state, the error during the trajectory tracking greatly increases, generating overshoots in the system response.

The state controllers (pos, vel, accel) and (pos, vel, F_H) are configured to set the cylinder poles at the same location. The first has a performance similar to the P controller, but the second generates an asymmetry in the position errors. This occurs because, as mentioned in section 4.3, when the hydraulic force is fed back the system is more sensitive to external disturbances, in this case the gravitational force ($F_G=mg$). Furthermore, as the state controllers feed back three signals that could have noise, their control signals are noisier than the P control that feed backs just one signal (position). This limits the value of the state control gains hindering their performance and even more, as discussed in section 4.3, the proportional directional valve has a slow and damped dynamic that reduces the efficiency of the state control.

One way to improve the P controller during the trajectory tracking control is to feed forward the desired velocity ($\dot{y}_d$) (Pereira, 2006). In Fig. 9 the results obtained with this strategy, the P control and the CC controller can be compared.

As shown in Figs. 9 and 10, the best results are achieved using the CC controller, because using this control strategy it is possible, among others things, to compensate the external disturbances (F_G and F_L) and the variable cylinder natural frequency. However, in the case of 14s < t < 16s, the CC controller has a poor performance, since in this region the cylinder is recoiled, having a small volume and, as mentioned in section 5, in this situation the system becomes more sensitive to the supply pressure drop.

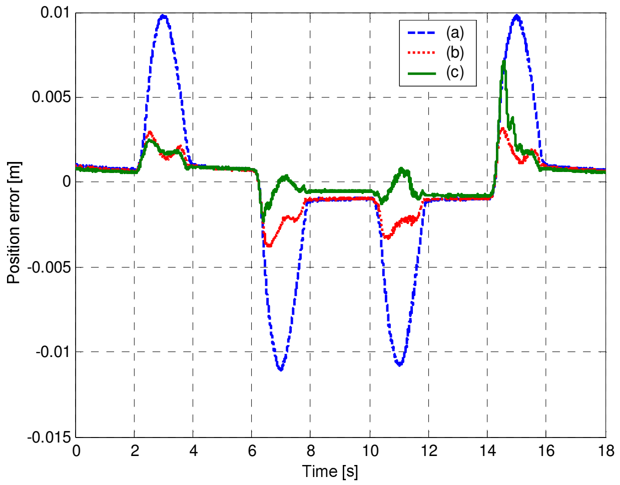

Figure 9. Responses to: (a) P, (b) P with forward loop (KP=300, K_{FL}=15) and (c) CC, without external load.

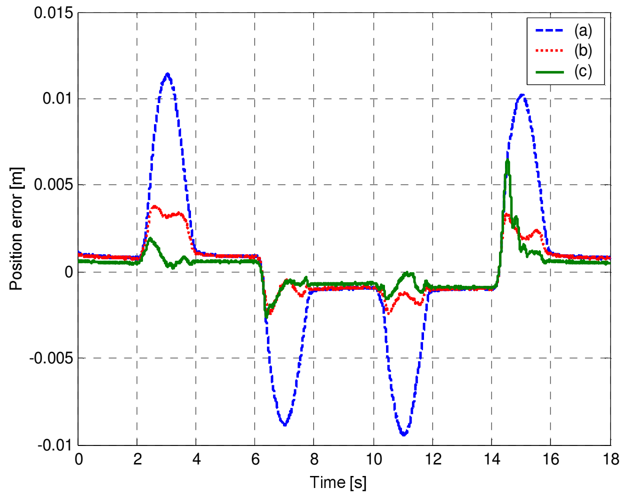

Figure 10. Responses to: (a) P, (b) P with forward loop (KP=300, K_{FL}=15) and (c) CC, with external load.

If the cylinder operates around its center region with small displacements, the system volume has a significant value which is almost constant. This makes the CC controller generates a high control signal to compensate the low natural frequency of the system in this region and, consequently, increases the robustness in relation to the supply pressure drop. Figure 11 shows these characteristics, where a trajectory with an amplitude of 0.05m is used. Analyzing Fig. 11, it can be observed that with the CC controller, the system response has a more homogeneous output and in general the best results are achieved.

Another characteristic about the CC controller, which was discussed in section 5, is its sensitivity to external disturbances, if they are not compensated in the CC control law. In this case, the position error stored by the CC controller increases considerably, as shown in Fig. 12.

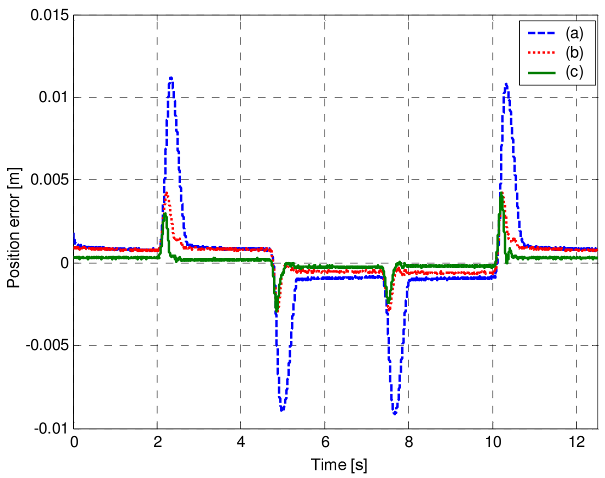

Figure 11. Responses to: (a) P, (b) P with forward loop (KP=300, K_{FL}=15) and (c) CC, with trajectory amplitude=0,05m and external load.

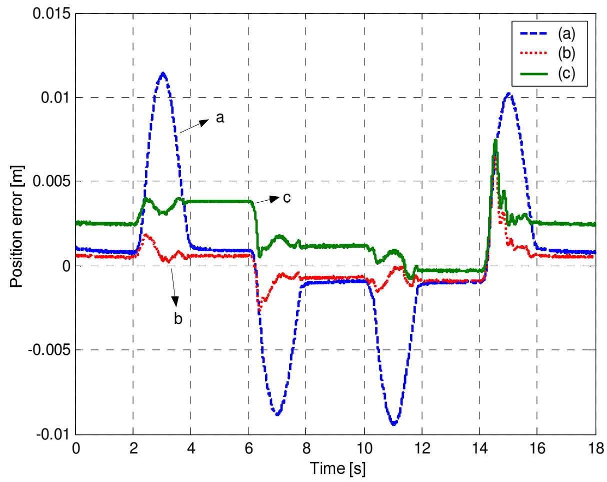

Figure 12. Responses to: (a) P, (b) CC with compensation of the spring force and (c) CC without compensation of the spring force.

8. CONCLUSIONS

The damped valve poles dominate the system dynamic, increasing the system damping and limiting the controllers gains.

The best results in the position and trajectory tracking control are obtained using the P controller with forward loop and, mainly, the CC controller.

One advantage of the linear controllers investigated in this study, in comparison to the CC controller, is that they do not use pressure signals in their control law, making the system more robust to a supply pressure drop.

Although the CC controller gives good results, it is an expensive control strategy with a higher computational cost, when compared with the other controllers (P, P with forward loop, PI, state). The CC controller needs three sensors to obtain the signals used in its control law, while the other controllers need only one sensor to measure the position.

Furthermore, as the control law of the CC controller is more complex, it requires faster computational processing. Thus, during the design of a controller, special attention should be paid to the accuracy specifications and to the budget available, given that, sometimes, a decrease of some millimeters in the position error can be very expensive.

9. ACKNOWLEDGEMENTS

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11. RESPONSIBILITY NOTICE