

Synchronous Motion Control of Dual-Cylinder Electrohydraulic Actuators Through a Non-Time Based Scheme

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Abstract: This paper presents a novel non-time based scheme for the synchronous motion control of dual-axis hydraulic systems. The technique, named Delayed Reference Synchronization Control (DRSC), is suitable for those applications where the time of execution of the task can be modified, while axis synchronization must be ensured. In practice, the DRSC allows achieving the synchronous motion by shifting in time the axis position reference on the basis of a scalar action reference parameter. Such a variable is computed on-the-fly on the basis of the sensed positions and of the estimated disturbances due to fluid compressibility, leakage flow and external forces. A multi-loop scheme is obtained, which is suitable for implementation in industrial controllers. The numerical results obtained demonstrate the stability and the effectiveness of the approach.

Keywords: Electrohydraulic actuators; synchronous motion control; non-time based control; delayed reference control.

1. INTRODUCTION

Many production systems often employ electrohydraulic actuators because of their high payload capability and high force-to-size ratio. On the other hand however, the dynamic of these systems is highly nonlinear and relatively difficult to control (Sohl et al., 1999). In order to fully exploit the advantages of hydraulic actuation, the synthesis of advanced and effective control techniques is therefore of the utmost importance for high-performance applications. Despite the extensive research on the position control of single-axis systems (see e.g. (Eryilmaz et al., 2001; Sohl et al., 1999; Vossoughi et al. 1995)), the synchronized motion control of multi-axis hydraulic systems is still an open and challenging field. Generally speaking, the problem of coordinated motion generation has been becoming over the last years a relevant issue in automation and robotics (Florin et al., 2001) to be properly tackled in the synthesis of effective control schemes. The problem of ensuring coordinated motion in multi-axis system is exacerbated when hydraulic actuators are employed, because of their non-ideal behaviour. This issue is relevant even when identical actuators are considered as a consequence of the presence of uneven loads, friction, hysteresis, variable environmental conditions and fluid behaviour, asymmetric lay-out with respect to the hydraulic power source, variability in the supply oil pressure (Chen, 2007).

Several approaches have been proposed in literature for the synchronous motion control of generic multi-axis systems (see e.g. (Lorenz et al., 1989; Sun, 2003; Xiao et al., 2005) and the references therein). Nevertheless, only few works proposes solutions explicitly accounting for the specific model of hydraulic actuators in the development of the

control scheme. Besides the traditional “open loop” approaches making use of hydraulic devices (e.g fluid flow dividers) or mechanical linkages, feedback solutions based on an additional synchronization control loop have been recently developed (see e.g. (Chen, 2007; Chen et al., 2008; Sun et al., 2001; Xiao et al., 2012)). In (Sun et al., 2001) a cascade control scheme is proposed, which comprises two independent SISO (single input, single output) pressure controllers in the inner loops of each axis, and a MIMO (multi input, multi output) linear robust controller in the outer loop. The latter is introduced to perform the synchronous motion control by generating the pressure references for the two SISO controllers. Fuzzy logic has also been adopted in (Chen, 2007; Chen et al., 2008) to synthesize both the controllers of each actuator and the motion synchronization controller. In (Xiao et al., 2012), the synchronization error is embedded in a performance index based on the model predictive control (MPC) theory, and the optimal synchronization control law is obtained through minimization of such an index.

In the quoted references, multi-loop schemes are developed by implementing synchronization control loops coupling the axes. By taking advantage of the idea of multi-loop control, a novel and original non-time based scheme is proposed in this paper for the synchronous motion control of dual-cylinder electrohydraulic actuators. In non-time based control strategies the desired path to be followed is not defined as an explicit function of the time, but of a scalar time-varying action reference parameter (Kang et al., 1999). The action reference parameter is computed on-the-fly on the basis of the variables to be controlled, for properly shaping the time history of the controller references in accordance with the control specifications. Reference planners assume therefore a new role: while in traditional time-based control schemes

trajectory planners are responsible for pre-planning input references as functions of just the time, in non-time based control schemes reference planners are a primary element of the feedback loop since the state trajectory affect the planned references. Generally speaking, non-time based strategies are well suited when following a path through space is of the utmost importance and when the coordinated motion of multi-axis system must be always ensured, even in the presence of uncertain dynamics or interaction with the environment (Kang et al., 1999; Richiedei et al., 2009). Given their effectiveness and ease of implementation, successful examples of non-time based controllers, sometimes referred to as “event based controllers”, have already been applied to several problems, such as telerobotic (Gasparetto et al., 2007), force control (Gasparetto et al., 2009; Gallina, 2009), vibration control (Gallina et al., 2004, 2005; Richiedei et al., 2008, 2009; Boschetti et al., 2011, 2012).

The approach proposed in this work takes advantage of the idea of the Delayed Reference Control (DRC), firstly proposed in (Gallina et al., 2004) and then exploited and validated experimentally in (Gallina et al., 2005; Richiedei et al., 2008, 2009; Boschetti et al., 2011, 2012). This control scheme has been originally developed for the simultaneous motion and vibration control in elastic systems. In DRC schemes the action reference parameter behaves as a “delayed time”, which delays or speeds up the rigid-body motion position reference to minimize the superimposed elastic displacements. On the basis of this successful idea, in this work a similar approach is proposed and validated numerically. Such an approach, named Delayed Reference Synchronization Control (DRSC), ensures the synchronous motion by properly delaying the axis position reference on the basis of the capability of the axes to track the reference position, which is related to the perturbations due to fluid compressibility, leakage flow and disturbance forces.

The paper is organized as follows: Section 2 briefly discusses the nonlinear model of an electrohydraulic actuator. On the basis of such a model, Section 3 describes the DRSC theory and synthesis. The numerical validation of the method is proposed in Section 4, by means of two significant test cases. Finally, concluding remarks are given in Section 5.

2. MODEL OF AN HYDRAULIC ACTUATOR

The scheme of electrohydraulic actuator discussed in this paper is the common valve controlled piston, with a double-acting hydraulic cylinder and a servo-controlled proportional valve modulating the oil flow from the pump to the cylinder (Balea et al., 2010). The model of the system components is described in the following, on the basis of the theory developed in (Merritt, 1967).

2.1. Fluid compressibility model

Hydraulic fluid compressibility in a chamber is governed by the equation of state:

$$\beta = -V \frac{dp}{dv} \quad (1)$$

where β is the fluid bulk modulus, V is the chamber initial volume and p the fluid pressure in the chamber, v its volume. In accordance with Eq. (1), the two continuity equations of the two cylinder chambers are written as follows (Avram et Al., 2011):

$$\begin{cases} \frac{V_A}{\beta_A} \dot{p}_A = Q_A - \dot{V}_A - Q_{L_A}^{ext} - Q_L^{AB} \\ \frac{V_B}{\beta_B} \dot{p}_B = Q_B - \dot{V}_B - Q_{L_B}^{ext} + Q_L^{AB} \end{cases} \quad (2)$$

where:

- Q_A and Q_B are the flows into the cylinder chambers;
- $Q_{L_A}^{ext}$ and $Q_{L_B}^{ext}$ are the external leakage flows, which are linear functions of the chamber pressure, on the basis of the external leakage coefficients $C_{L_A}^{ext}$ and $C_{L_B}^{ext}$ ($Q_{L_A}^{ext} = C_{L_A}^{ext} p_A$, $Q_{L_B}^{ext} = C_{L_B}^{ext} p_B$);
- Q_L^{AB} is the leakage flow across the two cylinder chambers (cross-port or internal leakage), and is a linear function of the pressure differential across the piston, on the basis of the internal leakage coefficient C_L^{AB} ($Q_L^{AB} = C_L^{AB} (p_A - p_B)$);
- V_A and V_B are the total fluid volumes of the two chambers, including piston volume and connecting lines:

$$V_A = V_{A0} + A_A x \quad (3)$$

$$V_B = V_{B0} + A_B (L - x)$$

where x is the piston position, L is the cylinder stroke, V_{A0} and V_{B0} are the initial volumes of chambers (i.e. when $x=0$), A_A and A_B are the piston areas;

- β_A and β_B are the effective (total) bulk modulus in the cylinder chambers, accounting for the liquid compressibility, the presence of entrapped air and the mechanical compliance (e.g. the flexibility of the hydraulic lines connecting the servovalve and the cylinder):

$$\frac{1}{\beta} = \frac{1}{\beta_l} + \frac{1}{\beta_c} + \frac{1}{\beta_g} \frac{V_g}{V_t} \quad (4)$$

In Eq. (4), β_l is the bulk modulus of the liquid, β_c is the bulk modulus of the container with respect to the total volume V_t , β_g is the bulk modulus of air ($\beta_g \approx 1.4p$, p is the fluid pressure), and V_g is the air volume.

In accordance with Eq. (3), the continuity equations can be written so as to highlight the relation between the flow rates into the cylinder chambers and the piston velocity:

$$\begin{cases} Q_A = A_A \dot{x} + \frac{V_A}{\beta_A} \dot{p}_A + (Q_{L_A}^{ext} + Q_L^{AB}) \\ Q_B = -A_B \dot{x} + \frac{V_B}{\beta_B} \dot{p}_B + (Q_{L_B}^{ext} - Q_L^{AB}) \end{cases} \quad (5)$$

The first term on the right side of both equations in Eq. (5) describes the relation between flow rate and piston speed in an ideal cylinder. The second term is the compressibility flow and describe the flow resulting from pressure changes in a compressible fluid. The third term is the total flow due to leakage.

2.2. Servovalve model

The flow rates Q_A and Q_B are linearly related to the servovalve spool displacement on the basis of the pressure-flow curve for the valve:

$$\begin{aligned} Q_A &= c_a y \sqrt{\Delta p_a} & \Delta p_a &= \begin{cases} p_s - p_A & \text{if } y > 0 \\ p_A - p_R & \text{if } y < 0 \end{cases} \\ Q_B &= c_b y \sqrt{\Delta p_b} & \Delta p_b &= \begin{cases} p_B - p_R & \text{if } y > 0 \\ p_s - p_B & \text{if } y < 0 \end{cases} \end{aligned} \quad (6)$$

where:

- c_a, c_b are the flow (constant) gain coefficients of the servovalve;
- p_s is the fluid supply pressure;
- p_R is the reservoir tank or reference pressure.

Equation (6) can be written in a more compact form by introducing the variable gain coefficients $\tilde{c}_a$ and $\tilde{c}_b$:

$$Q_A = \tilde{c}_a (\text{sign}(y), p_A, p_s, p_R) y \quad (7)$$

$$Q_B = \tilde{c}_b (\text{sign}(y), p_B, p_s, p_R) y$$

The valve spool position is related to the control input y_{ref} through a dynamic model, which can be modelled as a second order system, with natural frequency ω_v and damping ratio ξ_v (without any lack of generality, in Eq. (8) the same scaling is assumed for both y and y_{ref} , to provide a clearer representation):

$$\ddot{y} + 2\xi_v \omega_v \dot{y} + \omega_v^2 y = \omega_v^2 y_{ref} \quad (8)$$

$(|\dot{y}| \leq \dot{y}_{max}, |y| \leq y_{max})$

In practice, the valve dynamics is usually significantly faster than the cylinder dynamics and adequately damped (as a typical value $\xi_v > 0.7$), and therefore is commonly neglected in the controller synthesis (see e.g. (Eryilmaz et al., 2001; Sohl et al, 1999)).

2.3. Piston motion model

The application of the Newton's law in the piston motion direction leads to the following equation of motion:

$$M\ddot{x} + D\dot{x} = p_A A_A - p_B A_B - F_{ext} \quad (9)$$

where:

- M is the total moving mass in the x direction, resulting as the sum of the piston mass and the load mass;
- D is the viscous friction coefficient of piston and load;
- F_{ext} collects the external disturbance forces (except for those arising from viscous friction), such as for example elastic forces due to the interaction with the environment or Coulomb and stick-slip friction.

3. SYNTHESIS OF THE CONTROL STRATEGY

3.1. Model analysis

Differentiating Eq. (9) and substituting for $\dot{p}_A$ and $\dot{p}_B$ from Eq. (5) allows casting the nonlinear model of an hydraulic actuator as follows:

$$\ddot{x} + 2\xi_p \omega_p \dot{x} + \omega_p^2 x = \omega_p^2 k_{vy} y + \Omega_L - \frac{1}{M} \dot{F}_{ext} \quad (10)$$

In Eq. (10) the position-dependent cylinder natural frequency, ω_p , and damping ratio, ξ_p have been introduced:

$$\omega_p(x) = \sqrt{\frac{\beta_A A_A^2 + \beta_B A_B^2}{V_A(x) + V_B(x)}} \quad (11)$$

$$\xi_p(x) = \frac{D_p}{2M \omega_p(x)} \quad (12)$$

The term k_{vy} is the spool-position to cylinder-velocity gain:

$$k_{vy} = \left(\tilde{c}_a \frac{\beta_A A_A}{M V_A} - \tilde{c}_b \frac{\beta_B A_B}{M V_B} \right) \quad (13)$$

Finally, the effect of leakage flows, both internal and external, has been collected into the term $\Omega_L = \Omega_L(x, p_A, p_B)$, defined as follows:

$$\Omega_L = -Q_L^{AB} \left(\frac{\beta_A A_A}{MV_A} + \frac{\beta_B A_B}{MV_B} \right) - Q_{L_A}^{ext} \frac{\beta_A A_A}{MV_A} + Q_{L_B}^{ext} \frac{\beta_B A_B}{MV_B} \quad (14)$$

If an ideal actuator is considered, whose fluid compressibility and leakage flows are negligible, Eq. (10) is simplified as a proportional relation between the spool position and the piston velocity:

$$\dot{x} = k_{vy} y \quad (15)$$

Equation (10) can be therefore rearranged to highlight how the ideal behaviour in Eq. (15) is perturbed by the presence of three nonlinear disturbance terms, $\delta_{v,\beta}$, $\delta_{v,\Omega}$ and $\delta_{v,F}$:

$$\dot{x} = k_{vy} y - (\delta_{v,\beta} + \delta_{v,\Omega} + \delta_{v,F}) \quad (16)$$

These terms are the velocity perturbations due, respectively, to the fluid compressibility, to leakage flow, and to the presence of time-varying external forces:

$$\begin{aligned} \delta_{v,\beta} &= \frac{\ddot{x}}{\omega_p^2} + \frac{2\xi_p \dot{x}}{\omega_p} \\ \delta_{v,\Omega} &= \frac{\Omega}{\omega_p^2} \\ \delta_{v,F} &= -\frac{1}{M\omega_p^2} \dot{F}_{ext} \end{aligned} \quad (17)$$

The three aforementioned disturbance terms are highly nonlinear and uncertain, and are the main responsible of the tracking error. Therefore they should be properly accounted for in the synthesis of effective synchronous motion controllers.

3.2. The DRSC idea

Let us consider a dual-cylinder electrohydraulic system, consisting of two actuators (axes) which should track the same position reference, whose time history is referred to as $x_{ref}^*(t)$. Let x_i ($i=1,2$) be the absolute actual displacement of the i -th actuator. On the basis of the basic idea of non-time based control schemes, the axis position reference is not expressed as an explicit and pre-planned function of the time t , but as function of a scalar time-varying action reference parameter $l(t) \in \mathbb{R}$:

$$x_{ref}^*(t) = x_{ref}(l(t)) \quad (18)$$

In Eq. (18) $x_{ref}(l)$ is the pre-planned reference, which is known a-priori as an explicit function of l . The action reference parameter is, in turn, a function of the system state trajectory (or the measured output) and in particular of the variables to be controlled.

The control specification is therefore tracking the desired “path through space” defined through the two-dimensional

vector $\{x_{1,ref}(l) = x_{ref}(l), x_{2,ref}(l) = x_{ref}(l)\}$, rather than a trajectory in time as in the classical time-based control strategies.

In particular, following the basic DRC idea proposed in (Gallina et al., 2004) for the simultaneous motion and vibration control of elastic systems, l is defined as

$$l(t) = t - \tau(t) \quad (19)$$

The time-varying scalar $\tau(t) \in \mathbb{R}$ is named time delay; it is the key variable for achieving the synchronous motion by delaying the time histories of the axis position reference. The action reference parameter can be therefore thought of as a delayed time, which provides a shift in the time of execution of the motion on the basis of the capability of each axis to track the reference position. Clearly, such an approach is suitable for those application in which the time of execution of the task can be modified to ensure precise synchronization. Which often occurs in many processes (Lorenz et al., 1989).

As it is in the original DRC formulation, the strategy proposed in this paper leads to a cascade control scheme (Fig. 1). It comprises a unique MISO (multi input, single output) outer loop for the synchronous motion control (black lines in Fig. 1), and two independent SISO (single input, single output) inner loops for the position control of each axis (grey lines in Fig. 1). The outer loop includes the action reference block and the position planner. The action reference block computes on-the-fly the action reference parameter on the basis of the feedback measurements. l is then fed into the planner to generate the delayed reference for the inner position loops. No specific structure on the inner position controllers is required: DRSC schemes can be implemented by simply adding the outer loop to standard position controllers, as long as they ensure stable inner loops.

The multi-loop architecture of the DRSC, which does not require to modify the inner position control loops, enlarges the range of existing applications in which it could be employed. In particular it is also well suited for electrohydraulic actuators with proprietary controllers that cannot be modified to implement additional functions devoted to the synchronous motion control, as is common in practice in many industrial systems.

3.3. Controller synthesis

Since hydraulic actuators show under-damped elastic dynamics, as a consequence of the spring-like behaviour of the compressible fluid, the DRSC formulation can be directly obtained as an extension of the DRC developed for the oscillation control of elastic systems (Gallina et al., 2004; Richiedei et al., 2009). In the delayed reference control of such systems, the time delay is computed by integrating the variable representing the elastic behaviour to be controlled by the outer loop, e.g. the elastic unwanted displacement. For instance, in (Boschetti et al. 2011; Richiedei et al., 2008), the control of the suspended load oscillation in multi degree-of-freedom overhead cranes is performed by computing the time

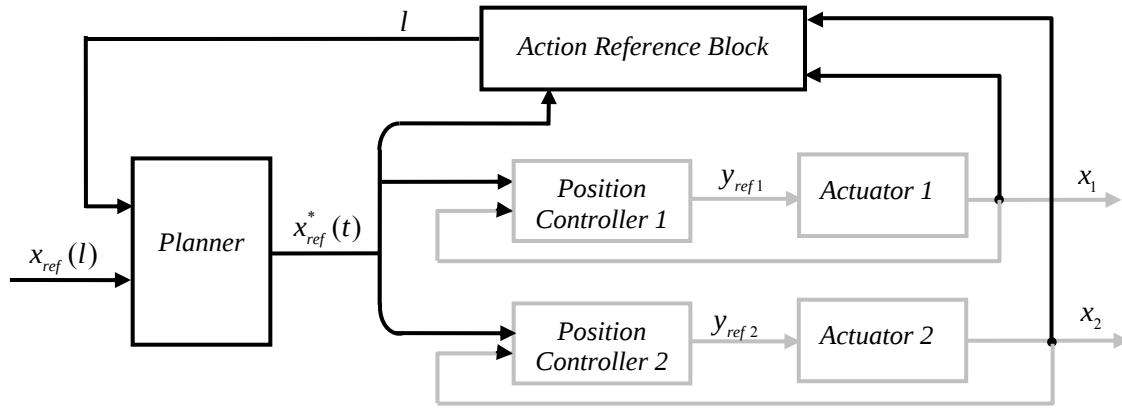


Fig. 1. Multi-loop architecture of the DRSC scheme.

delay as the weighed integral of the swing angle components, which cause the error in tracking the desired path. In (Boschetti et al., 2012) the DRC is extended to perform motion and vibration control in flexible link mechanism. Although control of these mechanisms is challenging, because of the presence of relevant nonlinearities and robustness problem (Caracciolo et al., 2008a), DRC provides an effective and stable control of the unwanted link oscillations. In (Boschetti et al., 2012) the time delay is computed by integrating the link curvatures, since they are the variables providing a significant representation of the elastic phenomena to be controlled (Caracciolo et al., 2006).

In the synchronous motion control scheme developed in this work, the time delay provided by the DRSC should be therefore computed by the following control law:

$$\tau(t) = \int_0^t \chi(e_s(\lambda)) d\lambda \quad (20)$$

where the function χ has the following properties:

- $\chi(e_s = 0) = 0$;
- χ is an increasing function of the synchronization error e_s ;
- χ is computed with a minimal set of sensors.

The first property assures that, if tracking error is null, then the delay does not increase. As a consequence the DRSC affects the system motion only when the actuators show no capability to track the delayed reference. In contrast, when the error increases the time delay should growth. Finally, in order to make the control scheme well suited for industrial application, no additional sensors should be required for just the computation of τ . In practice χ should represent both the synchronization error e_s and the phenomena causing it. A reasonable choice is therefore to consider for each axis both its tracking error and the estimated value of the displacement disturbances due to compressibility, leakage and external forces. The following function $\chi_i(t)$ is therefore introduced for both actuators:

$$\chi_i(t) = \delta_{xi}(t) e_i(t) \quad (21)$$

In Eq. (21) $\delta_{xi}(t)$ is the estimated value of the displacement disturbances affecting the i -th actuator and $e_i(t)$ is the tracking error:

$$e_i(t) = x_{ref}(l(t)) - x_i(t) = x_{ref}^*(t) - x_i(t) \quad (22)$$

In order to evaluate $\delta_{xi}(t)$, the actuator dynamic model formulated in Eq. (16) should be considered, to highlight the velocity disturbances due to fluid compressibility, leakage flow and time-varying external forces. The position disturbance $\delta_{xi}(t)$ is therefore the integral of the sum of these aforementioned terms:

$$\begin{aligned} \delta_{xi}(t) &= \int_0^t (\delta_{v,\beta} + \delta_{v,F} + \delta_{v,\Omega})_i d\lambda \\ &= \int_0^t \left(\frac{\ddot{x}}{\omega_p^2} + \frac{2\xi_p \ddot{x}}{\omega_p} + \frac{\Omega_i}{\omega_p^2} - \frac{1}{M\omega_p^2} \dot{F}_{ext} \right)_i d\lambda \end{aligned} \quad (23)$$

Rather than a direct implementation of the disturbance model defined through the three right-hand side terms in Eq. (23), the disturbance observer for the calculation of $\delta_{xi}(t)$ can take advantage of Eq. (16):

$$\begin{aligned} \delta_{xi}(t) &= \int_0^t (\delta_{v,\beta} + \delta_{v,F} + \delta_{v,\Omega})_i d\lambda \\ &= \int_0^t (k_{vy} y_i(\lambda)) d\lambda - x_i(t) \end{aligned} \quad (24)$$

Since measuring the servovalve spool position $y(t)$ would require additional sensors, $y_i(t)$ can be replaced in Eq. (24) by its estimated value $\hat{y}_i(t)$, which is computed on the basis of the command signal generated by the inner position loop, y_{ref_i} , and on the servovalve dynamic model (see Eq. (10)). The comparison of Eqs. (21), (22) and (24) leads to the final expression of the function χ_i to be integrated in the calculation of τ :

$$\chi_i(t) = \left(\int_0^t (\hat{k}_{vy} \hat{y}_i(\lambda)) d\lambda - x_i(t) \right) (x_{ref}^*(t) - x_i(t)) \quad (25)$$

In Eq. (25) the estimated, or nominal, value of k_{vy} ($\hat{k}_{vy}$) has

been introduced, in accordance with the manufacturer specification or with the experimental identification.

In order consider both actuators performing the same motion task, χ is defined as the scaled sum of χ_1 and χ_2 :

$$\chi(t) = k(\chi_1(t) + \chi_2(t)) \quad (26)$$

where $k \in \mathbb{R}^+$ is the DRSC gain.

In a less compact form, the following equation is obtained:

$$\tau(t) = k \int_0^t (\delta_{x_1}(\lambda) e_1(\lambda) + \delta_{x_2}(\lambda) e_2(\lambda)) d\lambda \quad (27)$$

It should be pointed out that Eq. (27) just involves the computation of an integral, and some simple algebraic operations. All these operations may be effectively performed by any real-time controller without incurring significant computational costs. This feature makes the DRSC very suited for commercial implementations also in industrial hardware.

In order to account for the actuator maximum accuracy, as well as for some amount of acceptable synchronization error, a dead-band can be applied to e_i so as to prevent from uncontrolled increase of τ due to errors that cannot be compensated. In Eq. (26) e_i is therefore replaced by $\tilde{e}_i$, which is computed on the basis of the dead-band width e_w

$$\tilde{e}_i = \begin{cases} e_i - e_w & e_i > e_w \\ 0 & |e_i| \leq e_w \\ e_i + e_w & e_i < -e_w \end{cases} \quad (28)$$

4. NUMERICAL ASSESSMENT

The theory developed above has been assessed by means of numerical simulations. The system recalls a two-axis testing machine, which interacts with an elastic environment (see Fig. 2). The two single-rod cylinders have the same nominal parameters, and therefore are theoretically equal (see Tab. 1). Nevertheless some parameter perturbations affect the second actuator (axis 2), in order to reproduce a critical condition for the achievement of accurate coordinated motion. In particular, 20% higher leakage coefficients and a double percentage of air in the oil (i.e. lower effective bulk modulus, see Eq. (4)) have been set. The models of the two actuators have been implemented on a complete nonlinear simulator, developed in Matlab-Simulink, which includes the model equations described in Section 2. Runge-Kutta integrator has been adopted to solve the nonlinear differential equations.

As far as the inner position controllers is concerned, a nonlinear control strategy is adopted, including position dependent gain-scheduled position and acceleration feedback, integral term, speed and acceleration feedforward:

$$y_{ref}(t) = k_p(x) e(t) + k_A(x) \ddot{e}(t) + \frac{1}{T_i} \int_0^t e(\lambda) d\lambda + k_{ffV} \dot{x}_{ref}^*(t) + k_{ffA} \ddot{x}_{ref}^*(t) \quad (29)$$

where the index i ($i=1,2$) has been omitted for clarity of representation. In Eq. (29) k_p and k_A are, respectively, the position and acceleration feedback gains, T_i is the integral action time, k_{ffV} and k_{ffA} are respectively the velocity and acceleration feedforward gains. This control scheme recalls the one commonly employed in industrial controllers. The position-acceleration feedback is employed to damp the system underdamped behaviour while increasing its frequency. The velocity feedforward, which is based on the ideal system model in Eq. (15), is aimed at reducing the transient tracking error, while the acceleration feedforward is aimed at reducing the effects of fluid compressibility. Only this brief description on the inner loop position controller is provided since it goes beyond the aims of this paper. As a matter of fact, DRSC schemes do not require any specific structure of the inner position loops, as long as they provide stability.

Two tests are proposed to validate the theory developed in the paper. They both consist on rest-to-rest motion, with symmetric piecewise-constant acceleration (defined in the l -domain), in the presence of interaction with an elastic environment. Two different motion amplitudes and environment stiffnesses are simulated in the two tests, so as to show some peculiar features of the DRSC. In particular, test 1 requires a smaller, and both cylinders are able to perform the whole desired motion. In contrast, in test 2 a larger displacement and a stiffer environment are simulated. This prevents one of the two axes from executing the whole desired displacement.

4.1. Path 1

The total displacement required in the first test is equal to 45 mm, from the initial piston position $x_i = -55$ mm (defined with respect to the midpoint cylinder position $x_i = 0$, $i=1,2$). When $x_i \geq x_i^*$ ($x_i^* = -30$ mm) both actuators interact with an elastic environment, whose stiffness k_e is set to 1.7 kN/m:

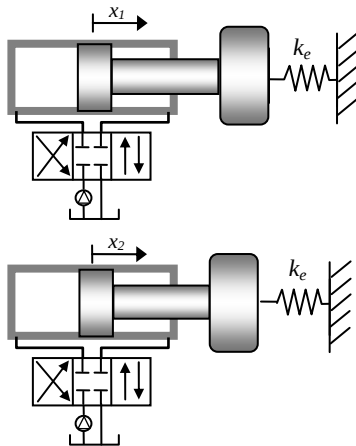
$$F_{ext} = \begin{cases} 0 & x_i < x_i^* \\ k_e (x_i - x_i^*) & x_i \geq x_i^* \end{cases} \quad (30)$$

The obtained results are displayed in figures from 3 through 10. In particular, figures from 3 to 6 show the results obtained by setting the DRSC gain $k = 4$ (1/m) (see Eq. (27)). Fig. 3 shows the time-histories of the actuator positions (x_1 and x_2) when the axes are independent, i.e. with just the inner position controllers (Fig. 3.a), and when the DRSC outer loop is also employed (Fig. 3.b). The resulting synchronization error e_s , which is defined as the difference between the two axis position is shown in Fig. 4:

$$e_s(t) = x_1(t) - x_2(t) \quad (31)$$

Table 1. Main parameters of the simulated actuators

Cylinder stroke	200 mm
Cylinder diameter	63 mm
Rod diameter	45 mm
Total moving mass	18 kg
Nominal bulk modulus	12000 bar
Maximum force	30 kN
Maximum velocity	250 mm/s
Maximum servovalve command voltage	5 V
Servovalve bandwidth	185 Hz

**Fig. 2.** Scheme of the simulated system.

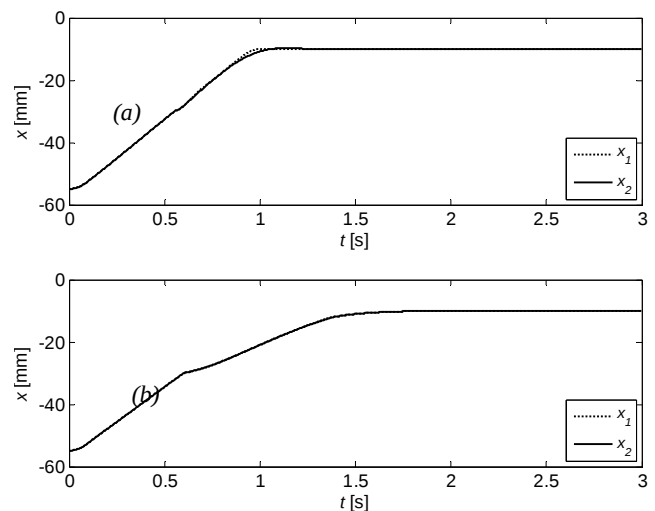
Both figures clearly highlight the DRSC capability to shape the position references so as to provide a significant reduction in the synchronization error, also in the transient. As a consequence, the time histories of x_1 and x_2 displayed in Fig. 3.b are almost overlapped if synchronization control is performed. In contrast, if no synchronization control is implemented, the synchronization error quickly increases when the actuators start interacting with the elastic environment and when the motion stops. This is a consequence of the relevant perturbations affecting axis 2.

Figure 4 also shows the synchronization error attained in the case a master/slave controller is adopted, as it is often done in industries because of its ease of implementation and effectiveness (Lorenz et al., 1989). The slowest actuator (axis 2) is the master, while the fastest one (axis 1) is the slave, which is forced to track the master position. Though reducing e_s compared to the independent axes, the master/slave approach does not provide the best result since it leads to an error higher than the one obtained with the DRSC. In particular the maximum error is 1.26 mm for the independent axes, 0.75 mm for the master/slave control and 0.06 mm for the DRSC. As a further advantage of DRSC over the master/slave approach, it should be mentioned that the latter one imposes the a-priori selection of a (real) master axis. Whenever the tracking error affects the slave axis, the synchronous motion is not preserved. The DRSC overcomes this occurrence: as a matter of fact, the tracking error affecting any axis causes the others to slow down, on the

basis of the scalar action reference parameter. The delayed reference generator, i.e. the planner, can be therefore seen as a virtual master axis, which defines the timing for all the axes by introducing a “virtual stiffness” between the two axes (Lorenz et Al. 1989).

Fig. 5 shows the time histories of the undelayed reference (independent axis reference) and the delayed one (DRSC reference) so as to highlight how the DRSC shapes the position references. Finally, Fig. 6 shows the time history of the time delay τ and of the action reference parameter l which allow achieving the aforementioned results. The time delay increases only during phases which are affected by the fluid elastic behaviour, such as acceleration and deceleration phases and especially when contact with the environment occurs, as it has been required in the controller specifications (see Section 3.3). When both actuators reach the final position, the time delay tends to a constant value. This is a relevant feature under a practical point of view since no uncontrolled growth of the time delay occurs.

The effect of gain variation is shown in figures from 7 through 10, where three different gains are compared: $k=0.25$ (1/m), $k=1$ (1/m) and $k=4$ (1/m). Clearly, the higher the gain, the smaller the synchronization error (see Fig. 7) and the larger the delay (see Fig. 8, Fig. 9, Fig. 10). The DRSC gain k therefore allows trading between the requirement of keeping small both the synchronization error and the delay of the motion execution. However, just a small gain ensures a significant reduction in the synchronization error with a negligible time delay. In fact, the maximum error obtained by setting $k=0.25$ (1/m) is 0.31 mm (see Fig. 7), which is significantly smaller than the one ensured by the master/slave control (0.75 mm), while the final time delay is 0.105 s (see Fig. 9). Finally, it is worth noticing that the DRSC is stable over a wide range of gains, and stability is preserved also in the presence of high gains. This is an important feature of DRSC schemes, since, generally speaking, higher gains often lead to stability problems in many control schemes of under-damped mechanical systems (see e.g. the discussion provided in (Caracciolo et Al., 2008b)).

**Fig. 3.** Piston positions vs. time with independent axes (a), and with DRSC (b), $k=4$.

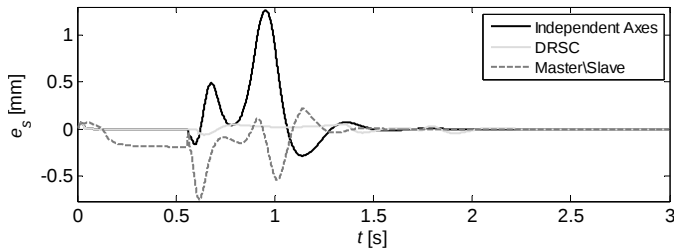


Fig. 4. Synchronization error vs. time. DRC multi-loop cascade structure.

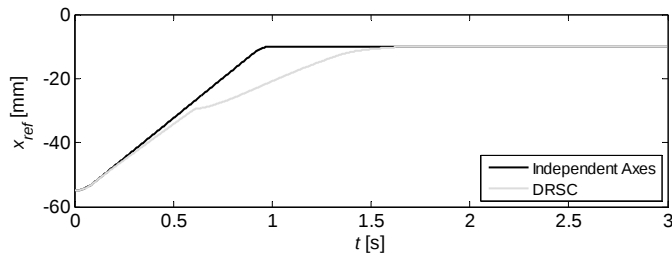


Fig. 5. Position references vs. time.

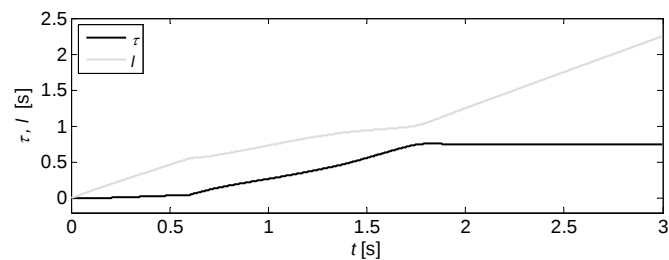


Fig. 6. Time delay and action reference parameter vs. time.

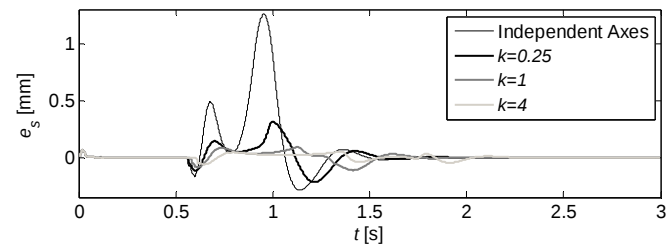


Fig. 7. Synchronization error vs. time.

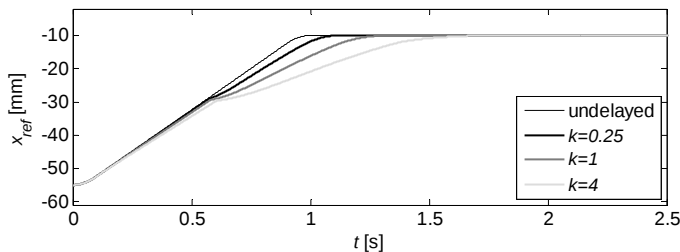


Fig. 8. Position references vs. time.

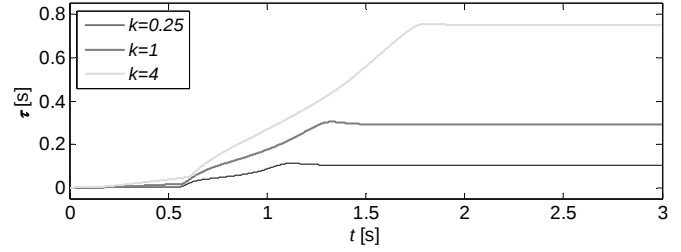


Fig. 9. Time delay vs. time.

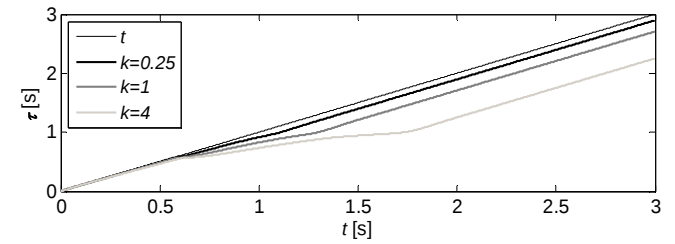


Fig. 10. Action reference parameter vs. time.

4.2. Path 2

A further proof of the DRSC capabilities arises from the second test. Compared with the first test, a larger displacement is required, from the initial piston position $x_i = -55$ mm to the final $x_f = -5$ mm. Once again, when $x_i \geq -30$ mm both actuators interact with an elastic environment with $k_e = 2$ kN/m. Being the environment stiffer and the required displacement larger than the one of first test, higher flow rates are required from the servovalve and higher disturbances affect the axis motion. The results are proposed in figures from 11 through 15. The simulations have been carried by setting $k=4$ (1/m), as it is in the first test proposed in Section 3.1. Fig. 11 and Fig. 12 show, respectively, the time histories of the piston positions and the resulting synchronization error, while Fig. 13 displays the control signals. It can be noticed that axis 2 (solid lines) is not able to execute the whole displacement because of the relevant elastic force due to the contact with the environment and the fluid leakage and compressibility. The control signal saturates at its maximum value (5 V) and the piston reaches a static equilibrium configuration with the external elastic force. If no synchronization control is performed, that causes a significant steady state synchronization error (6.55 mm), since axis 1 is able to execute the whole motion. Clearly this difference in the piston position may lead to damage in the machine mechanics. The introduction of the DRSC outer loop improves both the transient and the steady state behaviour, by leading to a maximum error equal to 0.12 mm. As far as the steady state behaviour is concerned, when actuator 2 approaches its static position the DRSC keeps constant the position references at the highest boundary position that both actuators can reach ($x_2 = 11.2$ mm, Fig. 14), by increasing the

time delay with the same rate of time, $d\tau/dt = 1$ (see Fig. 15, black line). Being $dl/dt = 0$, this leads to a constant value of l (grey line in Fig. 15) and therefore of the position reference, and prevents from uncontrolled increases of the synchronization error.

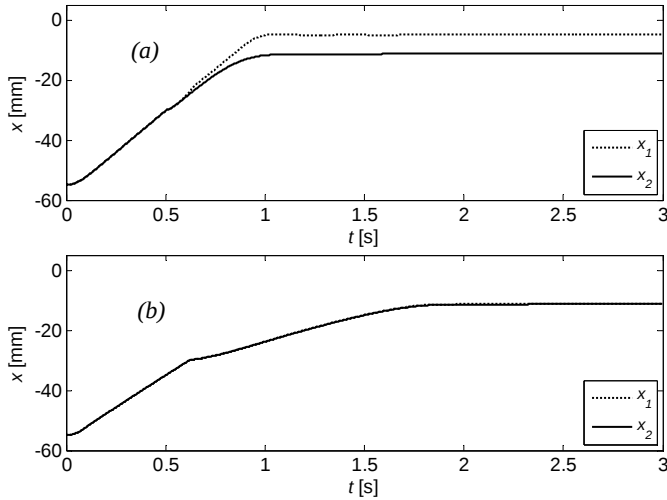


Fig. 11. Piston positions vs. time with independent axes (a), and with DRSC (b).

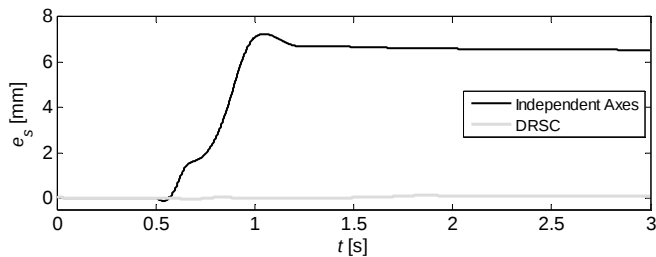


Fig. 12. Synchronization error vs. time.

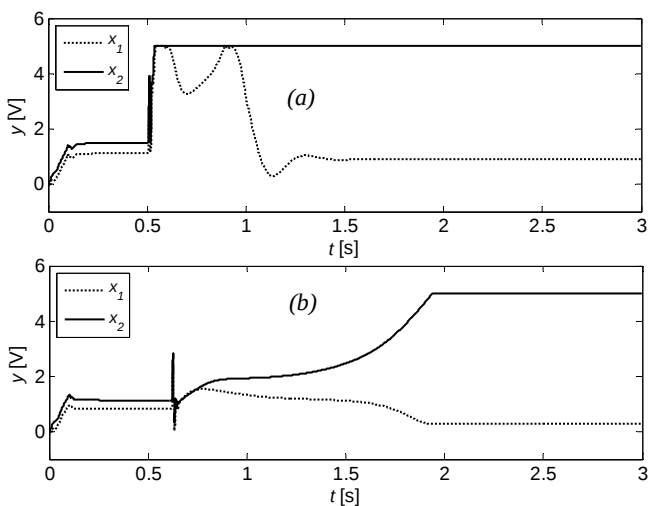


Fig. 13. Control signals vs. time.

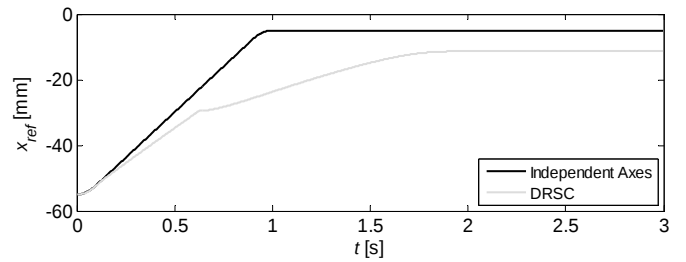


Fig. 14. Position references vs. time.

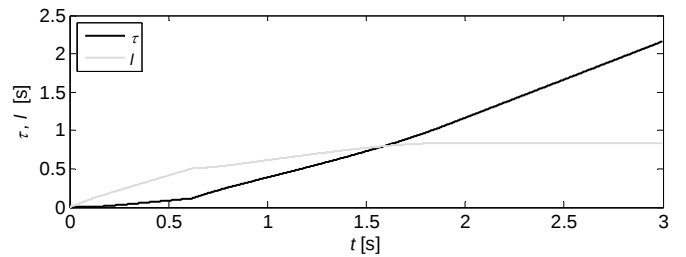


Fig. 15. Time delay and action reference parameter vs. time.

5. CONCLUSIONS

An original and novel non-time based approach, named Delayed Reference Synchronization Control (DRSC), has been proposed for the synchronous motion control of dual-axis hydraulic systems. The DRSC scheme performs the synchronization control by computing a scalar time delay to properly shift in time the axis position references. Such a delay is computed on-the-fly on the basis of the sensed tracking errors and of the estimation of the disturbances due to the fluid behaviour, in order to take into consideration the tracking capabilities of each actuator. A method to estimate the position disturbances due to fluid compressibility, leakage flow and external forces is presented.

The suggested DRSC scheme can be implemented by just adding a MISO outer loop to standard position controllers, as long as they ensure the inner loop stability. As a matter of fact, the implementation of the DRSC does not require specific structures of the inner loop position controllers. This fact, together with the ease of implementation (only position measurements are required) and the small computational effort required, enlarges the range of existing applications in which the DRSC could be employed. In particular, DRSC schemes are suitable for those tasks where the coordinated motion is of the primary importance and the time of execution of the task can be modified.

The simulation results proposed in the paper demonstrate the stability of the control scheme over a wide range of gains and its effectiveness in keeping the synchronization error to a minimum, even in the presence of interaction with the environment.

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