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Boris Vidolov. A Discrete-Time Design of Time Delay Control Focus on Control Experimentations of coordinated motion control of hydraulic excavator. 10th International IFAC Symposium on Robot Control (SYROCO 2012), Sep 2012, Dubrovnik, Croatia. hal-00757797

HAL Id: hal-00757797

https://hal.science/hal-00757797v1

Submitted on 27 Nov 2012

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A Discrete-Time Design of Time Delay Control Focus on Control Experimentations of coordinated motion control of hydraulic excavator

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Abstract: In this paper a discrete-time design of continuous-time delay (TDC) is presented. The main goals are: (i) to eliminate the TDC assumption of accessibility to all the delayed derivative of the plant state variables, and (ii) to design discrete-time controllers based on approximate discrete-time models. Simulations results are given, for an articulated arm actuated with linear hydraulic motors with variable payloads, to illustrate the robustness of the proposed control design. The new proposed controller has similar results as that obtained by TDC without the use of a delay element which has become a time constant. Moreover, there is no need of measuring the joints jerk. The controller robustness is guaranteed despite the uncertainties in dynamic model caused by payload variation, imprecise modeling, etc. Other uncertainties such as friction and external disturbances are rejected in the controller automatically. Real time experimentations, with a small scale Liebherr excavator, of coordinated motion control of implements are presented.

Keywords: discrete-time control, tuning parameter, coordinated motion control, hydraulic excavator

1. INTRODUCTION

In 1987, Youcef-Toumi and Ito (Youcef-Toumi et al. (1987)) proposed a robust control method for a class of nonlinear continuous-time systems with unknown dynamics and unexpected disturbances, i.e. $\dot{x} = f(x) + B(x)u + h(x,t)$ where $h(x,t)$ represents both unknown dynamics and unexpected disturbances, which is called Time Delay Control (TDC). Under the assumption of accessibility to all the state variables and estimates of their delayed derivative, TDC is characterized by a direct estimation of the function $h(x,t)$. This estimation is accomplished using time delay. This delayed observation of uncertainties is used to modify the control action directly, rather than to adjust the controller parameters like in gain scheduling or adaptive control techniques. Although TDC has been recognized as an efficient technique, see e.g. Youcef-Toumi et al. (1992.a) – Chang et al. (2003), it requires a good estimate of the system state derivative, and it is also inherently a continuous-time-based design introducing a delay in the control loop.

TDC implementations are thus carried out as time discretized functions of continuous-time functions, and this can lead to unstable closed-loop behavior as illustrated in the following example. Consider the first-order nonlinear plant proposed in Youcef-Toumi et al. (1990). This plant is exactly represented by $\dot{x} = \cos(x) + u + d$, $\cos(x)$ is treated to be the unknown dynamics, and $d(t) = 0.5\mathrm{I}(t-0.5)$ is the unexpected disturbance acting at t = 0.5s. Note that $\mathrm{I}(t)$ represents a unit step function. As expressed in Youcef-Toumi et al. (1990), the TDC is given by:

$$u(t) = u(t-L) - \dot{x}(t-L) - x(t) + \mathrm{I}(t)$$

Let the sampling period be equal to the time delay L, then the discretized TDC is

$$u(k) = u(k-1) - x(k) + 1 - \frac{x(k-1) - x(k-2)}{L}$$

Fig.1 shows simulation results where the plant output converges to the desired set-point under continuous-time control with L set to 0.05 s., and diverges under discretized control.

In this paper, in opposition to the standard TDC, we first focus on the fact that the design model has to be a discrete-time model, which is necessarily, an approximate model of the real system.

Performance of the control system was evaluated through simulations. Through simulations for a first order nonlinear plant, good model following and disturbance rejection property were obtained within a large range of sampling periods. The good model following and disturbance rejection property was confirmed through simulations for a more complicated non-linear system — a two-link manipulator carrying an unknown payload.

Real time experimentations, with a small scale Liebherr excavator, of coordinated motion control are presented. Proposed discrete-time TDC law is applied for the tracking control. Only the hydraulic cylinders positions are measured and only two tuning parameters are used, i.e. the sampling period *T*, and the dynamic of the desired state trajectory.

The required motion will be achieved via a two-stage sampled data system: the first one solves the inverse kinematics problem and send joint velocity set points to a tracking control system.

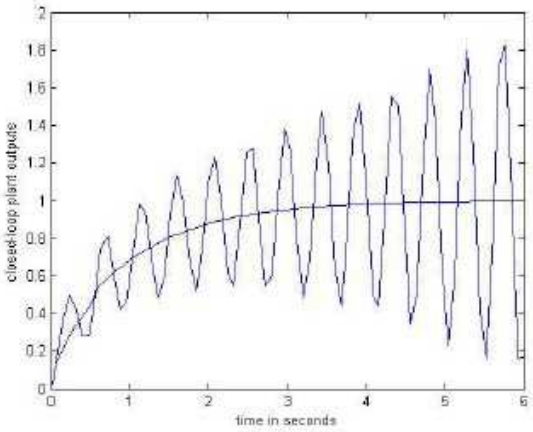


Fig. 1. Response of the first-order nonlinear system under continuous-time TDC versus a time-discretized implementation.

2. PRELIMINARIES AND PROBLEM STATEMENT

The design of any feedback control law requires some a priori knowledge about the dynamics of the plant to be controlled. In this paper, we consider the designer's a priori knowledge is a continuous-time plant model:

$$\dot{x} = f(x) + B(x)(u + \omega) \qquad (1)$$

where the term ω is completely unknown and represents both unmodeled plant dynamics and external unexpected perturbations. This plant model satisfies the matching condition that the unknown term ω enters the plant through the same input distribution matrix $B(x)$ as the control input u. As usual, we use the following notations:

$$x = [x_1, \ldots, x_n]^T \in X \subseteq \Re^n,\ u = [u_1, \ldots, u_p]^T \in U \subset \Re^p$$
$$B(x) = [b_1(x), \ldots, b_p(x)],$$

and assume the nonlinear functions, f, b_i for $i = 1 \ldots p$ are known and are, at least continuous. Given this a priori knowledge, and under the assumption of accessibility to all the state variables and estimates of their delayed derivative, TDC is characterized by a simple estimation of the unknown vector:

$$h(x,t) = B(x)\omega(t), \text{ i.e. } \hat{h}(x,t) = h(x(t-L), t-L)$$

which is directly derived from (1) at $t - L$.

In opposition to TDC, we first require that the design model to be used is a discrete-time model. However, even though the unknown term ω in (1) is ignored, an exact discrete-time model of (1) is impossible to obtain in general. Hence, it is realistic to use approximate discrete-time models in the control design. In the sequel, only the Euler forward integration scheme will be considered. This simple approximation scheme is consistent, and moreover, the $p+1$ known functions $f, b_1, \ldots, b_p$ are evaluated once only at each sampling instant.

Assuming, $x(k-1), u(k-1), x(k)$ are known at $t = kT$, then the consistency error due to the Euler-forward integration scheme is given by:

$$\tilde{e}(k) = \Delta x_k - T[f_{k-1} + B_{k-1}(u(k-1) + \omega(k-1))],$$

where short-hand notations such as $\Delta x_k = x(k+1) - x(k)$, $f_k = f(x(k))$, and $B_k = B(x(k))$ are used. Therefore, since $\omega(k-1)$ is unknown at that time, this later equation is rewritten as:

$$T\varepsilon(k) = \Delta x_k - T[f_{k-1} + B_{k-1}u(k-1)] \qquad (2)$$

where $T\varepsilon(k) = \tilde{e}(k) + TB_{k-1}\omega(k-1)$ represents both the consistency error due to the approximate numerical integration scheme, and the effects of unmodeled dynamics as well as unexpected disturbances over a sampling period. Note that, $\tilde{e}(k), \omega(k-1)$ cannot be separately computed; only the term $T\varepsilon(k)$ can be computed at $t = kT$.

Based on (2), a causal prediction of $\varepsilon(k+1)$ can be defined as $\hat{\varepsilon}(k+1) = \varepsilon(k)$. Hence, for any vector $u_k \in \Re^p$, a prediction $\hat{x}(k+1/u_k)$ of the next plant state is expressed as:

$$\hat{x}(k+1/u_k) = x_k + T[f_k + B_k u_k + \varepsilon(k)] \qquad (3)$$

In summary, given the a priori known continuous-time model (1), the model that will be used to design a control law is the one-step ahead prediction model (3), where $(\hat{x}(k+1/u_k) - \hat{x}(k))/T$ can be viewed as a prediction of the next state derivative under a particular choice of u_k.

3. DERIVATION OF CONTROL LAWS

Assume, the desired *n*-dimensional trajectory is known one step ahead at each sampling instant, i.e. $x_d(k+1)$ is known at $t = kT$ such that $x_d(k+1) = Fx_d(k) + Gr(k)$. Let, $e(k) = x_d(k) - x(k)$, be the tracking error. After some algebra, this leads to:

$$\hat{e}(k+1/u_k) = Fe(k) + Gr(k) + (F - I)x(k) - T\Delta\tilde{x}(u_k),$$

where $\hat{e}(k+1/u_k)$ is the predicted tracking error, and $\Delta\tilde{x}(u_k) = f_k + B_k u_k + \varepsilon(k)$. Hence, if it is possible to determine a control such that the following equation is always met:

$$Ke(k) = Gr_k + (F - I)x_k - T\Delta\tilde{x}(u_k) \qquad (4)$$

This will lead to, $\hat{e}(k+1/u_k) = (F + K)e(k)$ where K is an $(n \times n)$ error feedback matrix. The control u_k, that satisfy

(4), must then be selected in order to obtain a desired error dynamic. However, equation (4) cannot always be satisfied because the number of state variables is generally greater than the number of control inputs. Thus, the least square solution of (4) is adopted to determine the control u_k:

$$u_k = B_k^+(\varphi_k - \varepsilon(k)) \tag{5}$$

where $B_k^+ = \left(B_k^T B_k\right)^{-1} B_k^T$ is the Moore-Penrose pseudo inverse matrix of B_k, and:

$$\varphi(k) = -f_k + \varepsilon(k) + \frac{Gr(k) + (F - I)x(k) - Ke(k)}{T} \tag{6}$$

Control laws derived in this manner are based on the assumption of accessibility to all the state variables; they have only two tuning parameters, i.e. the sampling period *T*, and the dynamic of the desired state trajectory.

Using equations (3), (5), and (6), it is easy to exhibit the general equation governing the dynamics of the tracking error,

$$e(k+1) = (F + K)e(k) + \gamma_k \tag{7}$$

where,

$$\gamma_k = \left(I - B_k B_k^+\right)\varphi_k - T\left(\varepsilon(k+1) - B_k B_k^+ \varepsilon(k)\right) \tag{8}$$

Therefore, the tracking error $e(k)$ will be ultimately bounded if it can be proved that γ_k is bounded, i.e. $\|\gamma_k\| \le \bar{\gamma}$, provided the matrix (*F* + *K*) is a stable matrix, i.e. $\|F + K\| \le \rho < 1$.

The effectiveness of the proposed design method can be briefly illustrated on the example already mentioned in the introduction. The control is now expressed as:

$$u(k) = u(k-1) + \frac{(1-a)(r(k) - x(k)) + \Delta x(k)}{T}$$

where the desired state trajectory $x_d(k+1)$ has been derived from a continuous-time first-order dynamic system, i.e. $\dot{x}_d = x_d + 1(t)$, as in Youcef-Toumi et al. (1990), therefore $a = \exp(-T)$. Fig. 2 shows two cases of behaviors of the tracking error $e(t) = x_d(t) - x(t)$, corresponding respectively to *T* = 0.04 s., and *T* = 0.4 s. respectively. In both cases, the closed-loop plant trajectory converges to the desired trajectory.

4. APPLICATION TO MANIPULATOR CONTROL

The a priori knowledge representation of the dynamics of a *n*-link robot manipulator is:

$$D(q)\ddot{q} + G(q) = u - \omega(\ddot{q}, \dot{q}, q, t) \tag{9}$$

In this representation, the control vector *u* is the *n*-dimensional vector of forces/torques applied on the joint axis, $D(q) = D^T(q) > 0$ is an approximate inertia matrix (e.g. the inertia matrix derived from the assumption that all the links are point masses), $G(q)$ is also an approximation of the gravity terms. Hence, the unknown vector $\omega(\ddot{q}, \dot{q}, q, t)$ is a lumped representation of the centrifugal and Coriolis terms, of the friction torques on joints axis, and of the force/torque acting at the end-effector.

The state $x^T = \left[q^T, \dot{q}^T\right] = \left[x_1^T, x_2^T\right]$ is a *2n*-dimensional vector, and the design model is thus characterized by:

$$f(x) = \begin{bmatrix} x_2 \\ -D^{-1}(x_1)G(x_1) \end{bmatrix}; \; B(x) = \begin{bmatrix} 0 \\ D^{-1}(x_1) \end{bmatrix} \tag{10}$$

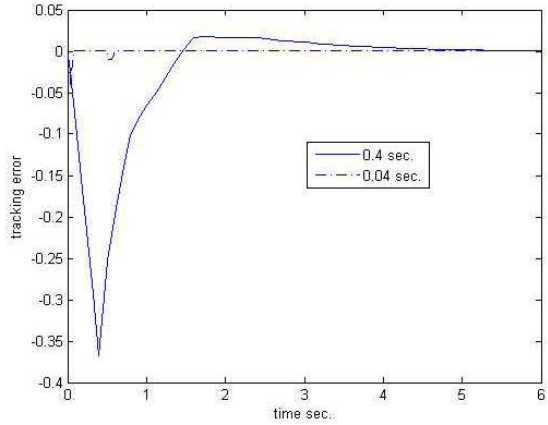


Fig. 2. Tracking error $x_d(t) - x(t)$ of the nonlinear first-order system under the control (7).

It is easy to show that $B^+(x) = [0 \vdots D(x_1)]$. Hence, using (5) and (6), only the *n*-dimensional vector $T\varepsilon_2(k)$ has to be computed at each sampling instant, i.e.

$$T\varepsilon_2(k) = \Delta x_2(k) - D_{k-1}^{-1}(u_{k-1} - G_{k-1}) \tag{11}$$

In order to determine the desired joint trajectory, a continuous-time reference model can be selected as a set of second order linear systems described by $\dot{x}_d = Ax_d + Br$, such that:

$$\begin{bmatrix} \dot{x}_{d1} \\ \dot{x}_{d2} \end{bmatrix} = \begin{bmatrix} 0 & I_n \\ A_{d1} & A_{d2} \end{bmatrix} \begin{bmatrix} x_{d1} \\ x_{d2} \end{bmatrix} + \begin{bmatrix} 0 \\ b_{d2} \end{bmatrix} r$$

Then, using the Euler-forward-integration scheme, this leads to $\hat{x}(k+1) = Fx_d(k) + Gr(k)$, where:

$$F = I + TA; \; G = TB \tag{12}$$

Note that, the error feedback matrix *K*, in (6), can be selected as *K* = 0, provided *F* = *I* + *TA* is a stable matrix. Thus, under this assumption, substituting (10)-(12) into (5), the control u_k can be expressed as:

$$u_k = G_k + D_k(\alpha_2(k) + \beta_2(k)) \tag{13}$$

where

$$\alpha_2(k) = D_{k-1}^{-1}(u_{k-1} - G_{k-1}) - \frac{\Delta x_2(k)}{T} \tag{14}$$

and

$$\beta_2(k) = A_{d1}x_1(k) + A_{d2}x_2(k) + B_{d2}r(k) \tag{15}$$

Note that, the second term in the right-hand side of (14) represents an approximation of the joint acceleration at $t = kT$. Practical implementations of this control, (13)-(15), can be carried out using only sampled joint position information $x_1(k-i), i = 0,1,2$, provided each joint sensor output is suitably filtered before sampling.

Simulation was done for a two-link manipulator, for which all the inertia and center of mass parameters were obtained from an industrial company. These technical data correspond to those of a heavy-duty excavator. The Matlab/Simulink toolbox was used to represent this system, where an additional point mass (m) situated at the end of the second link was used to represent an unknown payload.

In the design model the two links were assimilated as point masses. The same second order reference model was chosen for each link,

$$A_{d1} = \begin{bmatrix} -\omega_n^2 & 0 \\ 0 & -\omega_n^2 \end{bmatrix}; A_{d2} = \begin{bmatrix} -2z\omega_n^2 & 0 \\ 0 & -2z\omega_n^2 \end{bmatrix};$$

$$B_{d2} = \begin{bmatrix} \omega_n^2 & 0 \\ 0 & \omega_n^2 \end{bmatrix};$$

The simulation was performed with a sampling period T=0.05 s., with m = 0 kg, and m = 300 kg, respectively, using the same values of natural frequency (ω_n = 2.0 rad/s) and damping ration (z = 2.0). The input commands r_1 and r_2 where chosen to be step inputs.

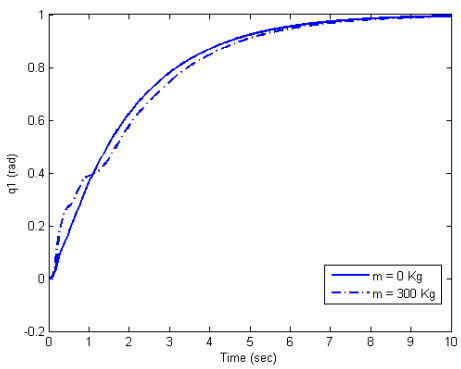


Fig. 3. Response of the first joint variable.

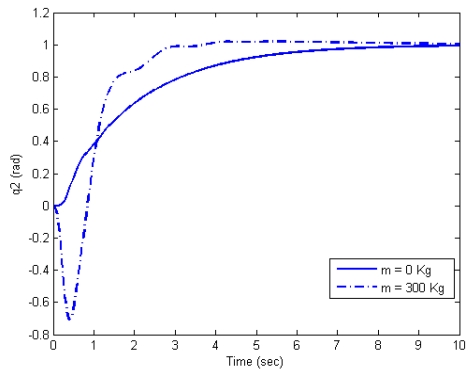


Fig. 4. Response of the second joint variable.

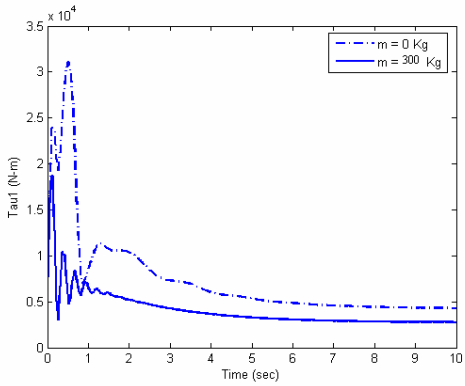


Fig. 5. Torque input on the first joint axis.

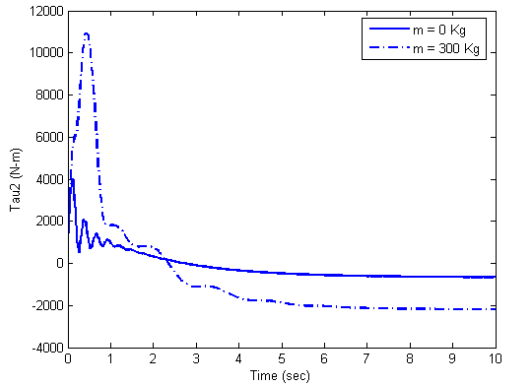


Fig. 6. Torque input on the second joint axis.

5. COORDINATED-MOTION CONTROL OF A HYDRAULIC EXCAVATOR

To illustrate the effectiveness of the proposed control law, we have chosen to consider a highly nonlinear system that is the arm of hydraulic excavator. The testbed (Fig.8.) consists of a small scale hydraulic excavator, xPC targetBox (Matlab/Simulink, Real Time Workshop) and joystick (definition of the Cartesian velocity $[V_x, V_y, \Omega]$). Each hydraulic cylinder is instrumented with linear displacement sensor SLS 095 Penny&Giles (mounted in parallel to the cylinders). The hydraulic system is driven by one hydraulic pump with output pressure of 10 [bars] and constant flow. The flow from the pump to the cylinders is controlled through variables orificies (Fig. 7.) by micro servo Graupner. The control signal u is a digital PWM signal with a 50 Hz frame rate.

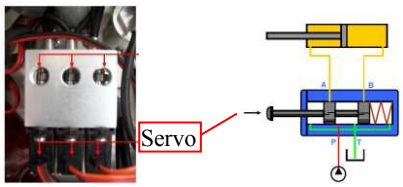


Fig.7. Flow control

5.1 Actuator model

The aim of this stage of model building was to establish a simplest actuator model. Let a highly simplified model be given as:

$$v(k+1) = \varphi(u(k)) + \omega(k+1) \qquad (16)$$

v – is the linear hydraulic cylinder velocity; ω - represents the non-modeled dynamics and all external perturbations and φ – is a static caracreristic of hydraulic cylinder. The latter caractericstic is obtained by some open loop experimentations – with zero initial velocity different control values u are applied at time k and $v(k+1)$ - the stabilized velocity at time $k+1$. Thus the steady states characteristics are obtained (Fig.10.). The control law proposed above is based on this simplest model.

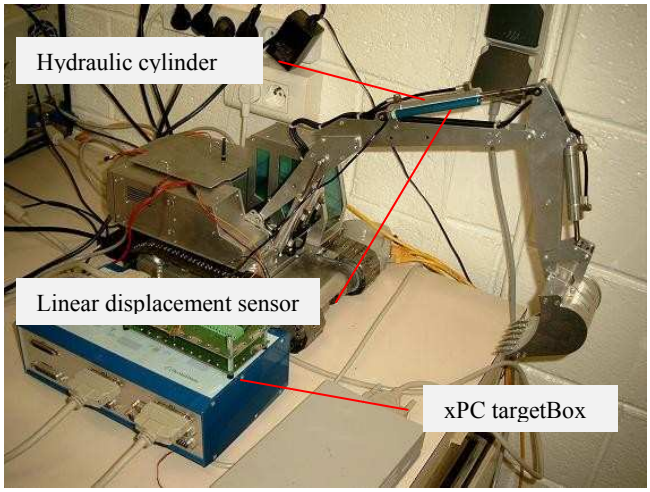


Fig. 8. Experimental testbed

The relationship between joint variables and measured hydraulic (Fig. 9) cylinders displacements can be written as:

$$q_1^2 = L_{BC}^2 + L_{BA}^2 - 2L_{BC}L_{BA}\cos(\theta_1 + \delta_1)$$

$$q_2^2 = L_{FE}^2 + L_{FD}^2 - 2L_{FE}L_{FD}\cos(\theta_2 + \delta_2) \qquad (17)$$

$$q_3^2 = L_{IG}^2 + L_{IH}^2 - 2L_{IG}L_{IH}\cos(\pi - \theta_3 + \delta_3)$$

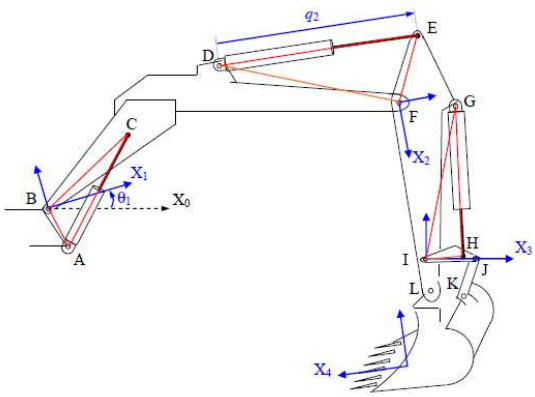
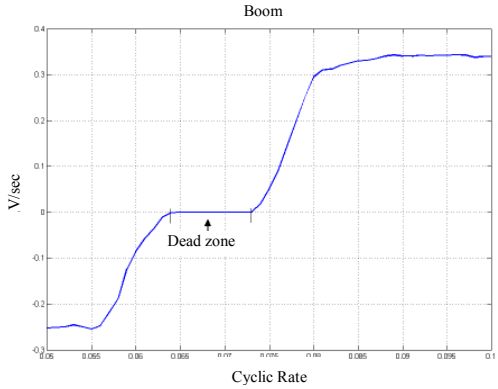
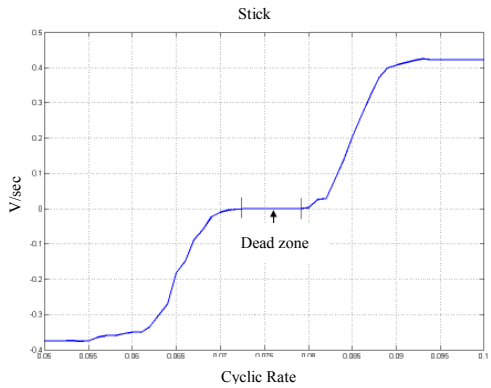


Fig.9. Detail of the arm links



a. Boom hydraulic cylinder



b. Stick hydraulic cylinder

c. Bucket hydraulic cylinder

Fig. 10. Steady states characteristics

The Jacobian matrix $J_{[3x3]}(\bar{\theta})$ is square and obtaining its inverse is trivial. Thus, with $\dot{q} = B(q)\dot{\theta}$ (derived from (17)), the relationship between Cartesian velocity vector and hydraulic velocities can be presented as:

$$\dot{q} = B(q)J^{-1}(\theta)\begin{bmatrix} V_x \\ V_y \\ \Omega \end{bmatrix} \qquad (18)$$

thus, joint velocity set points to a tracking control system are defined.

5.2 Real time experimentations

The experimentation was performed with $V_y = 0, \Omega = 0$ and V_x reference defined in Fig. 11. Dashed line presents measured bucket velocity. Fig. 12.a shows the closed loop tracking of boom, stick hydraulic cylinders. In Fig.12.b applied control signals are given.

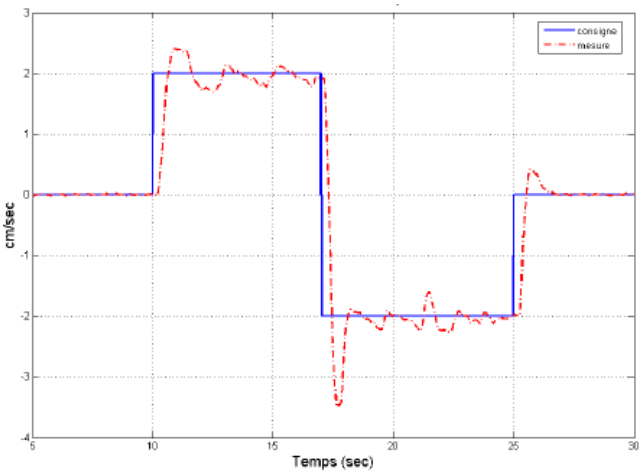
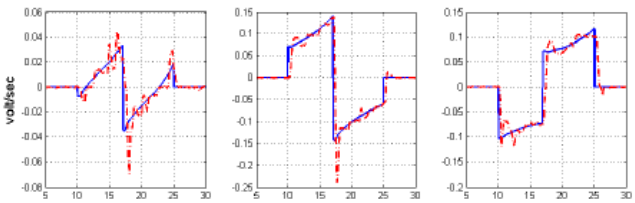
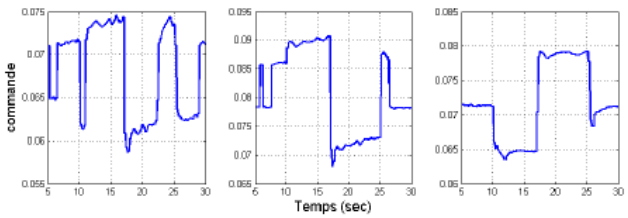


Fig.11. Tracking of reference Cartesian velocity



Boom Stick Bucket

a. Generated trajectory (continuous line), measured velocity (dashed line)



Boom Stick Bucket

b. Control signal

Fig.12. Tracking of reference Cartesian velocity

6. CONCLUSION

The discrete-time design proposed in this paper follows the presentation of TDC given in Youcef-Toumi et al. (1990). However, the design model which is used here is a nonlinear discrete-time model derived from a nonlinear continuous-time model, both being approximate models. In this design model, only the more recent past effects of unmodeled dynamics and unexpected disturbances are estimated at the beginning of each sampling period, together with the last consistency error due to the necessary use of an approximate-integration scheme (in this paper only the Euler-forward-integration scheme was used). This is accomplished under the assumption of accessibility to all the state variables.

Through simulations for a first order nonlinear plant, good model following and disturbance rejection property were obtained within a large range of sampling periods. The good model following and disturbance rejection property was confirmed through simulations for a more complicated non-linear system — a two-link manipulator carrying an unknown payload. In this case, all the joint velocities involved in the control were estimated using first order difference of joint positions. Performance of the control system was evaluated through simulations and real time experimentations with highly nonlinear system.

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