Force/Position Tracking for Electrohydraulic Systems of a Robotic Excavator

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Abstract— **This paper presents the sliding mode axis control of the electrohydraulic servo systems of a robotic excavator. The dynamic systems are described by a comprehensive model that accounts for nonlinearities and friction effects of asymmetric hydraulic cylinders. A sliding mode controller is developed for the electrohydraulic actuator of each working axis to provide stable force tracking. With a proper choice of the reference force, the control law can also allow tracking of a desired piston displacement trajectory. Numerical simulation demonstrates the validity of the proposed technique. Experimental results show good performance and strong robustness in both force and position following.**

I. Introduction

Autonomous execution of common excavation tasks (Figure 1) such as digging a footing or loading a haul truck confronts the problem of realising efficient tool-soil interactions. As the bucket interacts with the material to be excavated, the contact force should be regulated so that it remains within a specified range determined by the strength of the material and the power of the machine. To this end, bucket impedance control has been proposed ([5], [7]) to achieve a desired dynamic relationship between the digging force and the bucket tip position. Following this approach, the force required of each hydraulic cylinder can be determined. One needs to design a control law that causes the cylinder to track both a desired ram force and a desired position.

For electrohydraulic systems, feedback linearisation control has been proposed ([15], [1], [12]) to address hydraulic nonlinearities. In [14], a control law is obtained using the feedback linearisation framework. This approach has the advantage that the linearised system can be designed using the well-established tools from linear control theory to deal with nonlinearities in the system. The need for accurate system parameter and model information remains, however, a major disadvantage of feedback linearisation control.

For high control performance, robustness against non-

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Fig. 1. Robotic Excavator

linearities and uncertainties can be attained via the use of variable structure systems ([11], [2]) or adaptive control systems [16]. Several variable structure controllers have been proposed for hydraulic systems. Variable structure control offers robustness to unstructured uncertainties and global stability. A new sliding controller has been developed in the present work, to give high control performance in a simple implementation. Our approach separates the force control subsystem from the position tracking subsystem using a technique similar to integrator backstepping ([14], [1]). A fuzzy turning technique proposed in [8] is used to minimise the control chattering often associated with classical sliding controllers. To demonstrate system robustness against parameter uncertainties, extensive simulation has been carried out when the system is subjected to load, supply pressure and oil bulk modulus variations. Field tests have been conducted and experimental results are presented to validate the proposed technique.

The paper is organised as follows. After the introduction, modelling of electrohydraulic systems is briefly described in Section 2. The controller design is presented in Section 3. Friction estimation is described in Section 4. Simulation results for the boom cylinder hydraulic system are provided in Section 5. Experimental results obtained during teleoperated excavations are shown in Section 6, whilst conclusions and discussions are given in the final section.

II. Electrohydraulic servo system model

The model presented in this section is intended to emphasise the nonlinear nature of hydraulic actuators, to promote insight into the various physical phenomena that play an important role in determining the behaviour of hydraulic servosystems. A similar approach to the modelling of asymmetric hydraulic actuators can be found in [2].

The mathematical model of a hydraulic system is formulated from the laws of physics, such as mass balance, equations of motion for moving parts, equations for turbulent flow through small restrictions, and so on. Note that the hydraulic pump is modelled simply as a source of constant pressure, independent of the fluid flow. By following the oil flow in the different subsystems of the actuator and valve, comprehensive models of electrohydraulic system elements are derived in [9]. Let us first define the following state vector:

$$[y_1 \quad y_2 \quad y_3 \quad y_4]^{\mathsf{T}} = [y \quad \dot{y} \quad p_1 \quad p_2]^{\mathsf{T}}, \tag{1}$$

where y is the piston displacement and p_1, p_2 are the pressures in the compartments of the actuator. By combining the flow equations, the leakage flow equation, the oil compressibility equation, the equations of continuity, and the load equations of motion, one arrives at the following set of nonlinear state space equations for a hydraulic cylinder:

$$\begin{aligned}
\dot{y}_1 &= y_2 \qquad (2)\\
\dot{y}_2 &= \frac{1}{M}[A_1 y_3 - A_2 y_4 - w y_2 - F_{\mathsf{f}}]\\
\dot{y}_3 &= \frac{\beta}{A_1 y_1 + V_{\mathsf{L1}}}[-y_2 A_1 + C_{\mathsf{ip}}(y_3 - y_4)] +\\
&\quad + K_{\mathsf{v}} C_{\mathsf{d}}\left[\sqrt{2\frac{P_{\mathsf{s}} - y_3}{\rho}}\,\delta(u) - \sqrt{2\frac{y_3}{\rho}}\,\delta(-u)\right]u\\
\dot{y}_4 &= \frac{\beta}{A_2(L - y_1) + V_{\mathsf{L2}}}[y_2 A_1 + C_{\mathsf{ip}}(y_3 - y_4)] +\\
&\quad + K_{\mathsf{v}} C_{\mathsf{d}}\left[\sqrt{2\frac{y_4}{\rho}}\,\delta(u) - \sqrt{2\frac{P_{\mathsf{S}} - y_4}{\rho}}\,\delta(-u)\right]u\\
\delta(u) &= 1,\ u \geq 0\\
&= 0,\ u < 0, \qquad (3)
\end{aligned}$$

where u is the control voltage applied to the servo valve, P_{S} is the supply pressure from the pump set, L is the piston stroke length, w is linear viscous friction, β is the hydraulic bulk modulus, C_{d} is the flow discharge coefficient for a sharp-edged orifice, K_{v} is the valve coefficient, and F_{f} is the total opposing forces including nonlinear friction and external forces. Further nomenclature and numerical values of the parameters of the boom hydraulic cylinder are given in [12].

III. Controller design

A. Force tracking with a sliding mode control

The idea of force tracking follows from the time derivative of the fluid force acting on the piston:

$$\dot{F} = \dot{p}_1 A_1 - \dot{p}_2 A_2, \tag{4}$$

where F is the total hydraulic force generated by the cylinder. After substituting $\dot{p}_1$and $\dot{p}_2$ from (2) and we have

$$\dot{F} = A(y_1, y_2, p_1, p_2) + C_{\mathsf{gain}} Z(y_1, y_2, p_1, p_2)u, \tag{5}$$

where

$$\begin{aligned}
A(y) &= \frac{\beta}{y_1}[-y_2 A_1 - C_{\mathsf{ip}}(p_1 - p_2)]\\
&\quad - \frac{\beta}{L - y_1}[y_2 A_2 + C_{\mathsf{ip}}(p_1 - p_2)], \qquad (6)
\end{aligned}$$

$$\begin{aligned}
Z &= \frac{\sqrt{(P_{\mathsf{S}} - p_1)}}{y_1} + \frac{\sqrt{p_2}}{L - y_1} \quad if \quad u > 0\\
Z &= \frac{\sqrt{(P_{\mathsf{S}} - p_2)}}{L - y_1} + \frac{\sqrt{p_1}}{y_1} \quad if \quad u \leq 0 \qquad (7)\\
C_{\mathsf{gain}} &= K_{\mathsf{c}} C_{\mathsf{d}} \frac{\sqrt{2}}{\sqrt{\rho}}\beta, \qquad (8)
\end{aligned}$$

and where K_{c} is a constant.

As a result, the model (5) used for force tracking is simplified to a third-order model. Note that encoders and pressure transducers can readily be used to measure Z. The dynamic gain C_{gain}, which is the product of the oil bulk modulus and the flow discharge coefficient, is known to be very sensitive to system parameters. In general we assume that C_{gain} and $A(y)$ are bounded according to:

$$C_{\mathsf{gain\,min}} < C_{\mathsf{gain}} < C_{\mathsf{gain\,max}}, \tag{9}$$

and

$$\left| A(y) - \hat{A}(y) \right| \leq \alpha,$$

where $\hat{A}(y)$ denotes the value of $A(y)$ at the nominal regime of the system.

Assuming that the piston areas A_1 and A_2 and the external leakage coefficient C_{ip} are known with sufficient accuracy, a variable structure controller can then be developed. The sliding control method proposed in [13] is used in this paper.

First, let us define a scalar time-varying function

$$S(t) = F - F_{\mathsf{d}}, \tag{10}$$

where F_{d} is a desired force. The equivalent control u_{eq} is determined by the necessary condition for existence of a sliding mode $\dot{S} = 0$ with nominal values, denoted with the hat notation $(\hat{\cdot})$, of equation (5). This results in

$$\dot{S}(t) = \dot{F} - \dot{F}_d = \hat{A}(y) + \hat{C}_{gain} Z u_{eq} - \dot{F}_d = 0 \qquad (11)$$

$$u_{eq} = \frac{\dot{F}_d - \hat{A}(y)}{\hat{C}_{gain} Z} = \hat{C}_{gain}^{-1} Z^{-1}(\dot{F}_d - \hat{A}(y)), \qquad (12)$$

where $\dot{F}_d$ is obtained from F_d by differentiation.

The control law must satisfy a sliding condition regardless of uncertainties in the values of A and C_{gain}. A discontinuous term is therefore added to drive the system state to the sliding surface $S = 0$. Thus, the control law takes the following form:

$$u = u_{eq} - \hat{C}_{gain}^{-1} Z^{-1} K sgn(S), \qquad (13)$$

where K is a positive constant to be chosen.

B. Stability Analysis

In essence, to achieve a zero tracking error all system trajectories must be forced to reach the sliding surface $S(t) = 0$ in a finite time, and to remain on this surface once it is reached. A sufficient condition can be expressed as

$$\frac{1}{2}\frac{d}{dt} S^2(t) \le -\eta \left| S(t) \right|, \qquad (14)$$

where η is a strictly positive value. This leads to

$$S(t)\dot{S}(t) \le -\eta \left| S(t) \right|. \qquad (15)$$

We now compute K from (13) such that the above sliding condition is met. Substituting u from (13) into (10) yields

$$\dot{S} = A(y) + C_{gain} Z \hat{C}_{gain}^{-1} Z^{-1} (\dot{F}_d - \hat{A}(y) - K sgn(S)) - \dot{F}_d \qquad (16)$$

$$= A(y) + C_{gain} \hat{C}_{gain}^{-1} (\dot{F}_d - \hat{A}(y) - K sgn(S)) - \dot{F}_d$$

$$= A(y) - C_{gain} \hat{C}_{gain}^{-1} \hat{A}(y) + (C_{gain} \hat{C}_{gain}^{-1} - 1)\dot{F}_d$$

$$- C_{gain} \hat{C}_{gain}^{-1} K sgn(S).$$

By substituting (16) into the sliding condition (15), we obtain

$$[A(y) - C_{gain}\hat{C}_{gain}^{-1}\hat{A}(y) + (C_{gain}\hat{C}_{gain}^{-1} - 1)\dot{F}_d$$
$$-C_{gain}\hat{C}_{gain}^{-1} K sgn(S)] S \le -\eta \left| S(t) \right|$$
$$\eta + [A(y) - C_{gain}\hat{C}_{gain}^{-1}\hat{A}(y) +$$
$$+(C_{gain}\hat{C}_{gain}^{-1} - 1)\dot{F}_d] \frac{S}{|S(t)|} \le C_{gain}\hat{C}_{gain}^{-1} K.$$

The above expression may be rearranged to give

$$K \ge \eta C_{gain}^{-1}\hat{C}_{gain} +$$
$$+ \left| C_{gain}^{-1}\hat{C}_{gain} A(y) - \hat{A}(y) + (1 - C_{gain}^{-1}\hat{C}_{gain})\dot{F}_d \right|$$
$$\ge \eta C_{gain}^{-1}\hat{C}_{gain} +$$
$$+ \left| C_{gain}^{-1}\hat{C}_{gain} A(y) - C_{gain}^{-1}\hat{C}_{gain}\dot{F}_d + \dot{F}_d - \hat{A}(y) \right|$$
$$\ge C_{gain}^{-1}\hat{C}_{gain}(\eta +$$
$$+ \left| (A(y) - \hat{A}(y)) - (\dot{F}_d - \hat{A}(y)) + \dot{F}_d - \hat{A}(y) \right|).$$

Substitution of u_{eq} from equation (11) into the above expression gives

$$K \ge C_{gain}^{-1}\hat{C}_{gain}(\alpha + \eta) + \qquad (17)$$
$$+ \hat{C}_{gain} Z |C_{gain}^{-1}\hat{C}_{gain} - 1| \left| u_{eq} \right|.$$

By introducing the gain margin

$$\beta = \sqrt{\frac{C_{gain\,max}}{C_{gain\,min}}} > 1, \qquad (18)$$

with $0 < C_{gain\,min} < C_{gain} < C_{gain\,max}$, we have $C_{gain}^{-1}\hat{C}_{gain} \le \beta$. The discontinuous gain K in (17) can then be rewritten as:

$$K \ge \beta(\alpha + \eta) + \hat{C}_{gain} Z(\beta - 1) \left| u_{eq} \right|. \qquad (19)$$

As a result, if the control gain K satisfies equation (19), the sliding condition will hold. The force tracking error will then be reduced to 0 in a finite time, given the bounds on C_{gain} and $A(y)$ as in the assumptions (9). For practical implementation of the proposed controller, the *signum* function in (13) may be replaced by a tanhyperbolic function, which is derived from a fuzzy reasoning technique [8]:

$$u = u_{eq} - \hat{C}_{gain}^{-1} Z^{-1} k \tanh(\frac{S}{\phi}), \qquad (20)$$

where the positive constant ϕ can be adjusted in practice to eliminate chattering associated with the sliding controller.

C. Position tracking

To achieve position tracking, the desired force is chosen to have the form presented in [14]:

$$F_d = m\ddot{y}_d(t) - k_v(\dot{y}(t) - \dot{y}_d(t)) - k_p(y(t) - y_d(t)) + \hat{g}(\dot{y}), \qquad (21)$$

where $\hat{g}(\dot{y})$ is an estimate of friction in the hydraulic cylinder. Letting $e = y - y_d$, the equation of motion (2) becomes

$$m\ddot{e} + k_v\dot{e} + k_p e = (F - F_d) + (g(\dot{y}) - \hat{g}(\dot{y})), \qquad (22)$$

where $g(\dot{y}) - \hat{g}(\dot{y})$ is the friction estimation error which can be considered as a disturbance. Equation (22) represents a second-order linear system in $e(t)$ driven by $F - F_d$.

D. Remarks:

- If the force error e_F approaches to zero, the position error e_P approaches to zero and vice versa, according to a specified dynamic relationship defined by the selection of the gain k_v and k_t in equation (21).
- F_d is a refrence force that is generated by the computed system. We need to design a force controller for the servo-hydraulic system that could track the set-point value provided by (21) as close as possible [6].
- In the industrial hydraulic system, friction may reduce the force at the actuator up to 30% [3], so that the accurate estimation of friction needs to be addressed. That is the subject of the next section.

IV. Friction estimation

Based on the variable structure method, a new friction and velocity observer has been developed in our laboratory . A detailed derivation and comprehensive experimental results can be found in [4]. The main advantage of a new observer is to enhance its robustness against plant parameter variations. The discontinuous observer is described by

$$\begin{aligned} \tilde{\dot{v}} &= -(F_\mathsf{f} + w + L_1\sigma)/m \qquad (23)\\ F_\mathsf{f} &= -L_2 \cdot \sigma \\ \tilde{\dot{y}} &= \tilde{v} - \sigma/m \\ \sigma &= M \cdot sgn(x - \tilde{x}), \qquad (24) \end{aligned}$$

where $M > 0, L_1$ and L_2 are respectively the observer parameters, x, v, and F_f representing position, velocity and friction, and the tilda notation $\tilde{(\cdot)}$ denotes the estimates. The switching action in the observer dynamics can be alleviated by replacing the signum function with [4]:

$$\sigma = M \cdot \tanh(\frac{y - \tilde{y}}{\gamma_{\mathsf{e_y}}}), \qquad (25)$$

where $\gamma_{\mathsf{e_y}}$ is some positive constant.

V. Simulation study

This section presents results of simulation studies with the controller derived above. Consider the axis control problem for the bucket motion of the robotic excavator actuated by a hydraulic servo system with parameters given in [12]. The control objective is to track a desired sinusoidal input command

$$y_\mathsf{d}(k) = 0.5 * (x_{\mathsf{d\,max}} - x_{\mathsf{d\,min}}) * sin(0.1\pi kT)[m],$$

with a sampling time $T = 0.002s$. The desired position input will be used to compute the desired force and its derivative in equation (21). The set of equations (2) are then used in the simulation to determine the motion of the cylinder and its states ($y, \dot{y}, p_1$ and p_2).

Assuming that the friction here has a Coulomb model. Figure 2 shows the force tracking response. The position tracking response and position error are presented respectively in Figures 3. It can be seen that the force and position tracking performance is quite good, with nearly zero steady-state error despite the frictional disturbance. The simulation is also used to examine the invariant characteristics of the proposed controller in the presence of varying parameters such as valve coefficients and fluid bulk modulus. The tracking errors are slightly changed when C_{gain} is varied by about 40% around its nominal value. As noted earlier, the error in friction modelling has entered as a disturbance in the position tracking response.

VI. Experimental results

Experiments were conducted to validate the simulation results. Data acquisition and control algorithm were written in C++ and executed under the Windows NT operating system. The hardware connections diagram for the

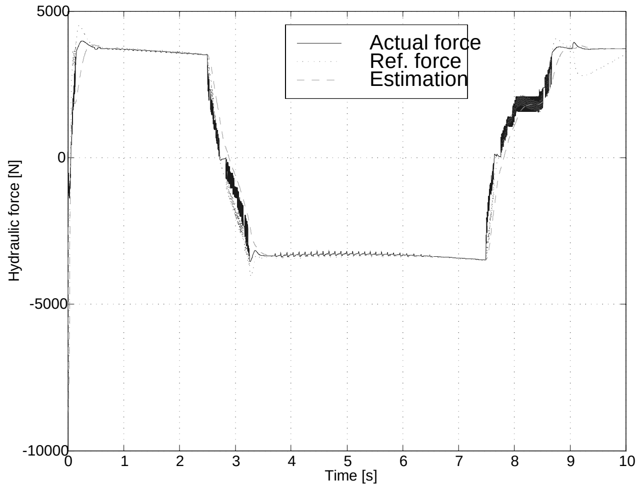

Fig. 2. Simulated force tracking

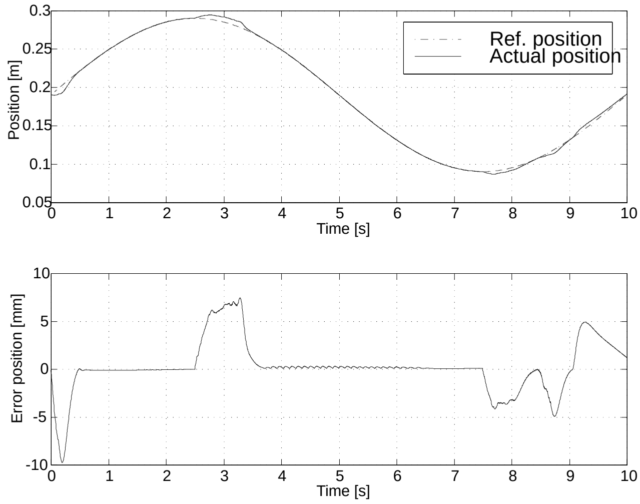

Fig. 3. Simulated position tracking

robotic digger is shown in Figure 4. The programmable servo controllers, together with the necessary power supplies, are installed in a 19 inch rack as can be seen in Figure 5. Further details of hardware and software organisation can be found in ([11], [10]). The force acting on cylinder can be measured by two pressure transducers installed on the head and rod side.

Figure 6 shows the controller response with a sinusoidal position reference input with the excavator's boom in free space, where the top figure shows the boom position response (in $[m]$) and the bottom figure shows the position error response (in $[10^{-3} \times m]$). It is noted that position tracking is very accurate with the error magnitude smaller than 6mm at the turning point and nearly zero at the other path. The position error may, however, be further improved by tuning the controller gains, in this case the proportional and derivative gains (k_p and k_v in equation (21)). The experimental trials also revealed that a suitable tuning scheme for these controller parameters will depend on whether the cylinder is extending or retracting. Fig-

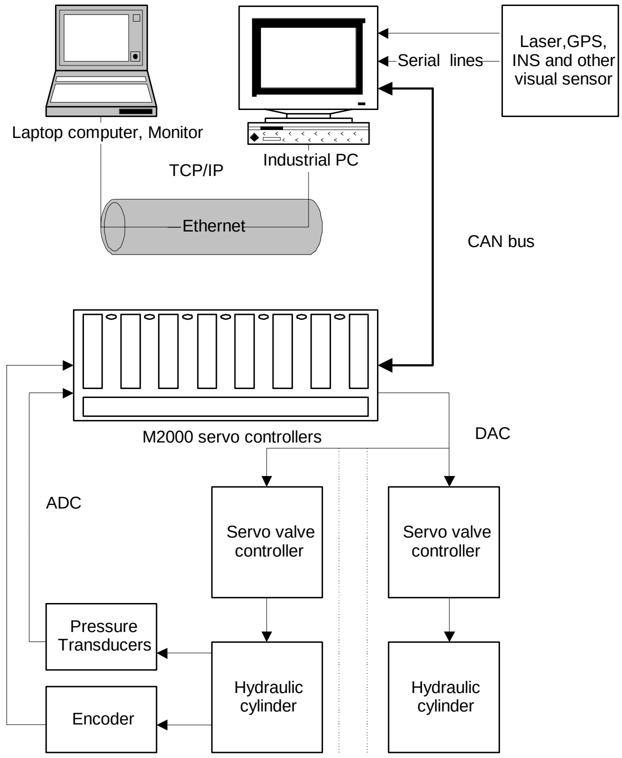

Fig. 4. Excavator hardware connections diagram

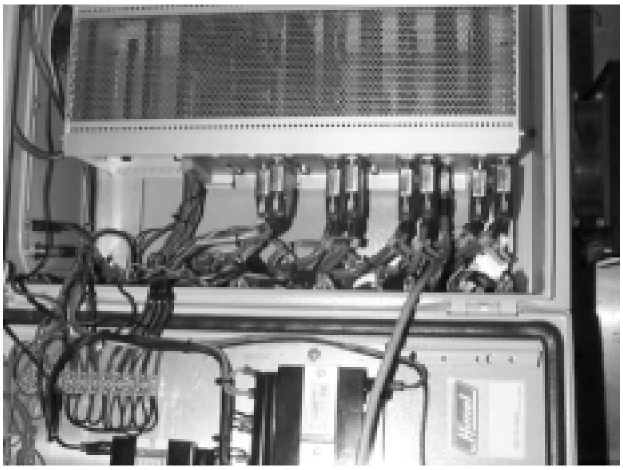

Fig. 5. Programmable servo controllers rack

ure 7 shows the computed control voltage applied to the boom servovalve when the robotic excavator is executing a digging task with a prescribed sinusoidal pattern. Finally, the controller performance without the use of friction compensation is shown on Figure 8, where the top and bottom figures show respectively the boom position and error responses (both in $[m]$). Note that without friction compensation the boom position response exhibits a bigger tracking error. Experimental results indicate that control performance is strongly dependant on selection of the stability margin η in equation (15). That value determines the controller's fast response to the cylinder pressures, which are usually very noisy for industrial hydraulic systems. The ex-

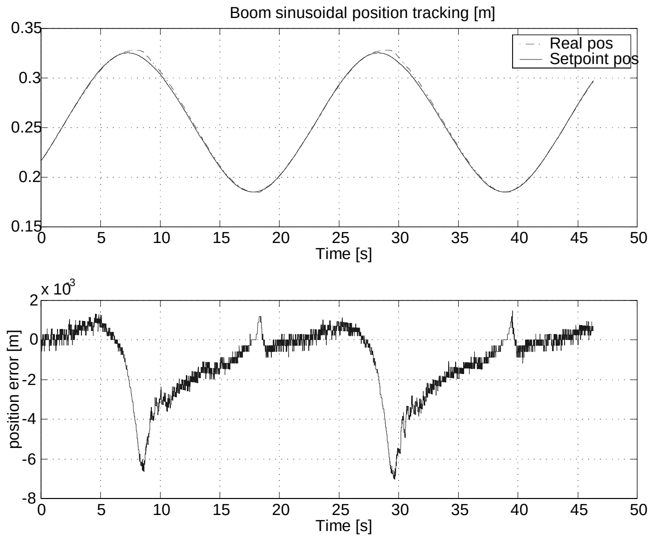

Fig. 6. Experimental path tracking and error

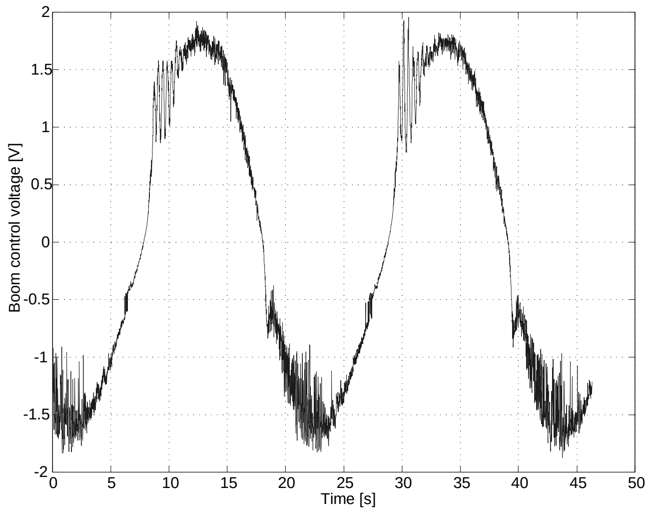

Fig. 7. Experimental computed control voltage

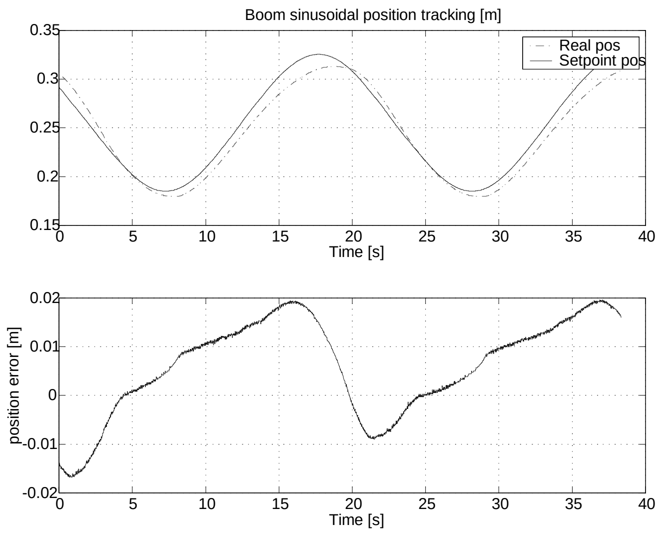

Fig. 8. Experimental tracking without friction compensation

periments verified that even in the conditions of parameters and load uncertainties, the cylinder position can perfectly track the desired time-varying trajectory while the system force approaches to a desired dynamic relationship of the position.

VII. Conclusions and discussions

This paper has presented the derivation, simulation and implementation of a force/position tracking control law for electrohydraulic systems with linear cylinders. The design is based on a variable structure systems method with considerations to reduce chattering. The controller is applied to the axis control problem of a robotic excavator. Simulation and experimental results obtained demonstrate the feasibility and validity of the proposed control scheme in dealing with nonlinearities and disturbances in electrohydraulic systems. Different initial conditions have also been imposed with varying initial pressures in the compartments without observing any degradation in control performance. Unlike our previous work on impedance control, in this study force is assumed as a control input which results from the desired dynamics of the position tracking process via a proper selection of the proportional and derivative gains, k_p and k_v. It should be further noted that hydraulic friction as a disturbance source is an important factor that affects the force tracking accuracy. With friction compensation from a dicontinuous observer, tracking performance has been experimentally verified to exhibit significant enhancement.

VIII. Ackowledgement

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