PERGAMON

Automatica 37 (2001) 589–595

automatica

www.elsevier.com/locate/automatica

Brief Paper

Variable structure methods in hydraulic servo systems control☆

A. Bonchis[a,*,1], P. I. Corke[b], D. C. Rye[a,1], Q. P. Ha[a,1]

[a]CSIRO Manufacturing Science and Technology, P.O. Box 883, Kenmore Qld 4069, Australia
[b]Australian Centre for Field Robotics, The University of Sydney, NSW 2006, Australia

Received 2 August 1999; revised 1 August 2000; received in final form 16 October 2000

Abstract

In the general framework of hydraulic servo systems, this paper addresses the problem of position control in the presence of important friction nonlinearities. The accent falls on the variable structure methodology, as we try to use its intrinsic robustness properties. Several friction observers, including the one based on a variable structure approach, were incorporated and tested in an acceleration feedback control. Next, we present a novel implementation of a variable structure controller, which lumps friction and load as an external disturbance. Results of extensive experimental testing encourage the use of variable structure methods in a class of highly nonlinear hydraulic servo systems. © 2001 Elsevier Science Ltd. All rights reserved.

Keywords: Variable structure; Friction compensation; Observers; Robust control; Hydraulic servo systems

1. Introduction

In hydraulic servo systems friction is an important source of nonlinearity, considerably diminishing the force or torque available at the actuators. In addition, at motion reversal and low velocity, a host of dynamic effects have been observed, and appropriate friction models have been developed (Armstrong-Hèlouvry, Dupont, & Canudas de Wit, 1994). Increasing the positioning accuracy in such systems requires adequate measures to alleviate the adverse effects of friction. One of the most common ways is to provide the controller with quantitative information on friction, achieving what is commonly referred to as model-based friction compensation. As direct measurement of friction is not possible, two options are models based on experimental friction identification or the use of nonlinear friction observers. For the system at hand, experimental friction identification resulted in a pressure-dependent model capable of describing friction over the entire velocity range (Bonchis, Corke, & Rye, 1999). Nonlinear reduced-order friction observers require at least position and external force measurements (Friedland & Mentzelopoulou, 1992; Tafazoli, de Silva, & Lawrence, 1995). In order to increase robustness of estimates, an observer based on a variable structure systems approach was suggested (Ha, Bonchis, Rye, & Durrant-Whyte, 2000). Its application to position control for hydraulic servo systems is highlighted in this paper.

In electrical servo systems, friction compensation is a straightforward technique, due to the proportionality between the control current and the output torque. This is hardly the situation in their hydraulic counterparts. Acceleration feedback has been used in order to achieve friction compensation in such systems (Tafazoli, de Silva, & Lawrence, 1998), on the ground that the estimated acceleration bears friction information.

The use of friction models or compensation is not a requisite condition for improving the positioning performance of the system. In essence, any robust control technique should provide a solution to the problem. We will focus here on a variable structure control with sliding mode, which proved its potential in electro-hydraulic servo systems (Hwang & Lan, 1994; Fung, Wang, Yang, & Huang, 1997). Most results have been reported for systems operated by hydraulic motors, while our case deals with an asymmetric hydraulic cylinder. The approach of Slotine and Sastry (1983) combined with a fuzzy logic reasoning reported by Ha (1997) was used in

☆This paper was not presented at any IFAC meeting. This paper was recommended for publication in revised form by Associate Editor L.-C. Fu under the direction of Editor M. Araki.

* Corresponding author. Tel.: + 61-7-3327-4464; fax: + 61-7-3327-4455.

E-mail address: adrian@cat.csiro.au (A. Bonchis).

[1] Supported by the Centre for Mining Technology and Equipment, Australia.

PII: S0005-1098(00)00192-8

order to minimise chattering and to determine the control in the boundary layer neighbouring the switching surface.

We analyse model-based compensation in Section 2, where the discussion focuses on friction estimators and on the method used to compensate for friction in hydraulic servo systems. Section 3 presents the development and stability analysis for a variable structure controller in the case of a hydraulic servo system with asymmetric cylinder which is of interest to us. Due to its widespread use, a PD control was implemented and used as a test benchmark. The experimental set-up and results obtained follow in Sections 4 and 5, respectively. Finally, conclusions are presented in Section 6.

2. Model-based friction compensation

A straightforward technique to design a friction estimator is by using an experimentally identified friction model. For the mechanical system shown in Fig. 1, consisting of a hydraulic cylinder moving a load, the equation of motion for the piston is

$$M\ddot{y} = p_1 A_1 - p_2 A_2 - F_{\mathrm{f}}, \tag{1}$$

where p_1, p_2 are the pressures at the cylinder ports, A_1, A_2 the piston areas, F_{f} the friction force, M the load mass (including piston mass), and y its displacement. If the piston moves with constant velocity motion, friction can be computed based on pressure measurements at both ports, assuming that piston areas are also known. For the system under consideration, results of the friction identification experiments are detailed in Section 5.

An observer-based application for friction estimation and compensation was reported in Friedland and Mentzelopoulou (1992). The observer dynamics is postulated in the form

$$\hat{F}_{\mathrm{f}} = \hat{a}\,\mathrm{sgn}(\hat{v}), \tag{2}$$

$$\hat{a} = z_a - k_a|\hat{v}|^{\mu}, \tag{3}$$

$$\dot{z}_a = k_a\mu|\hat{v}|^{\mu-1}(a_{\mathrm{e}} - \hat{a})\,\mathrm{sgn}(\hat{v}), \tag{4}$$

where a represents the piston acceleration, v the velocity, and a_{e} is the acceleration component generated by the external forces. The observer state is z_a and the design parameters are the gain $k_a > 0$ and the exponent $\mu > 0$.

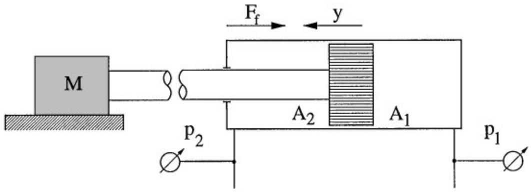

Fig. 1. Mechanical system analysed.

The usual 'hatted' notation is used for estimated values. Velocity is estimated from measured position by

$$\hat{v} = z_v + k_v y, \tag{5}$$

$$\dot{z}_v = -k_v\hat{v} + a_{\mathrm{n}}, \tag{6}$$

$$a_{\mathrm{n}} = a_{\mathrm{e}} - \hat{a}\,\mathrm{sgn}(\hat{v}) \tag{7}$$

with a_{n} representing the net acceleration, z_v the velocity estimator state, and k_v being the design parameter. A slightly changed version has been introduced by Tafazoli et al. (1995), by replacing (6) with

$$\dot{z}_v = -k_v\hat{v}. \tag{8}$$

Based on the theory of discontinuous observers with a variable structure, a new friction and velocity observer was developed (Ha et al., 2000). The main advantage sought was to enhance the robustness of the observer against plant parameter variations. The discontinuous observer can be described by

$$\dot{\hat{y}} = \hat{v} - \sigma/M, \tag{9}$$

$$\dot{\hat{v}} = -(\hat{F}_{\mathrm{f}} + a_{\mathrm{e}} + L_1\sigma)/M, \tag{10}$$

$$\dot{\hat{F}}_{\mathrm{f}} = -L_2\sigma, \tag{11}$$

$$\sigma = \sigma_M\,\mathrm{sgn}(y - \hat{y}) \tag{12}$$

with $\sigma_M > 0$, L_1, and L_2 being the observer parameters. The convergence of the observer has been proved for $L_1 < 0$ and $L_2 > 0$. To avoid chattering generated by (12), a fuzzy technique replaces the signum function with

$$\sigma = \sigma_M \tan h\left(\frac{y - \hat{y}}{\gamma_{\mathrm{e}}}\right), \quad \gamma_{\mathrm{e}} > 0. \tag{13}$$

To test the options for providing friction information discussed so far, an acceleration feedback control (AFB) has been used. The control law for friction compensation is given by

$$u = -K_{\mathrm{AFB}}[(\hat{a} - a_d) + \alpha_{\mathrm{AFB}}(\hat{v} - v_d) + \beta_{\mathrm{AFB}}(\hat{x} - x_d)] \tag{14}$$

with the controller gains chosen in the first instance based on a heuristic method described in Tafazoli et al. (1998), and then tuned to further improve the performance of the positioning system. By using acceleration feedback the controller receives, indirectly, a hint on the friction level in the system.

Using the identified friction model or one of the friction observers described previously, the acceleration estimate $\hat{a}$ needed in (14) can be computed.

3. Variable structure control for asymmetric cylinders

3.1. Controller design

The variable structure control (VSC) development is exemplified here for the case when the piston extends. The treatment of the retraction case is similar.

A third order model is suggested for the design of the VSC, which will be expressed directly in control canonical form

$$\dot{y} = v, \tag{15}$$

$$\dot{v} = a, \tag{16}$$

$$\dot{a} = (\dot{p}_1 A_1 - \dot{p}_2 A_2)/M. \tag{17}$$

The pressure derivatives $\dot{p}_1$ and $\dot{p}_2$ are given in terms of flows through the directional valve as

$$\dot{p}_1 = \frac{B}{V_{L_1} + yA_1}(K_q\sqrt{p_s - p_1}u - vA_1), \tag{18}$$

$$\dot{p}_2 = \frac{B}{V_{L_2} + (y_{\max} - y)A_2}(-K_q\sqrt{p_2 - p_t}u + vA_2), \tag{19}$$

where B is the oil bulk modulus, $V_{L_{1,2}}$ are the inactive cylinder volumes, $y_{\max}$ is the piston stroke, u is the control signal, $p_{s,r}$ are the supply and return pressures, and K_q is a constant flow coefficient which can be approximated using data supplied by the valve manufacturer.

Substituting (18) and (19) in (17), we obtain in compact form

$$\dot{a} = f(y, v) + b(y, v)u, \tag{20}$$

where

$$f = -\frac{v}{M}[\beta_1 A_1^2 + \beta_2 A_2^2], \tag{21}$$

$$b = \frac{K_q v}{M_p}[\beta_1 A_1\sqrt{\Delta p_1} + \beta_2 A_2\sqrt{\Delta p_4}] \tag{22}$$

with β_1 and β_2 being functions of the piston position y. The system dynamics f is estimated by $\hat{f}$, and we seek a bound for the estimation error

$$|f - \hat{f}| \leq \varepsilon_f, \tag{23}$$

where the bound ε_f could depend on the y, v, and a. Assuming that $A_{1,2}$ and M are known with sufficient accuracy, an expression for ε_f is found of the form

$$\varepsilon_f = -\frac{B}{M}\left(\frac{A_1^2}{V_{L_1}} + \frac{A_2^2}{V_{L_2}}\right)\max(|\dot{y} - \dot{y}_d|), \tag{24}$$

the index d denoting reference values. Due to the multiplicative effect, the control input gain can be estimated as

$$\hat{b} = \sqrt{b_{\min} b_{\max}}. \tag{25}$$

By manipulating Eq. (22), we get

$$\hat{b} = \frac{K_q B}{M}\sqrt{\frac{1}{2}\sqrt{p_{v\min} p_{v\max}}\sqrt{\max(g)\min(g)}}, \tag{26}$$

where

$$g(y) = \frac{A_1}{V_{L_1} + A_1 y} + \frac{A_2}{V_{L_2} + A_2(S - y)} \tag{27}$$

and $p_{v\min}, p_{v\max}$ represent the minimum and maximum valve pressure drops. For the state error vector $\mathbf{e} = [e_y e_v e_a]$ with the components $e_y = y - y_d$, $e_v = v - v_d$, and $e_a = a - a_d$, we define a scalar time-varying surface $S(\mathbf{e}, t) = 0$, with S being

$$S(\mathbf{e}, t) = e_a + 2\lambda e_v + \lambda^2 e_y, \quad \lambda > 0. \tag{28}$$

The equivalent control u_{eq} is determined from the condition $\dot{S} = 0$, resulting in

$$u_{eq} = \hat{b}^{-1}[-\hat{f} + \dot{a}_d - 2\lambda e_a + \lambda^2 e_v]. \tag{29}$$

To accommodate the estimation errors, a discontinuous term is added to (29)

$$u = u_{eq} - \hat{b}^{-1}k\,\mathrm{sgn}(S) \tag{30}$$

3.2. Stability analysis

In essence, to achieve perfect tracking, all system trajectories have to converge to S in finite time and stay on S afterwards, a condition expressed mathematically as

$$\frac{1}{2}\frac{d}{dt}S^2(\mathbf{e}, t) \leq -\eta|S(\mathbf{e}, t)|, \tag{31}$$

where η is a strictly positive design parameter. We have to determine k in (30) such that the above condition is satisfied. From (28) we obtain

$$\dot{S} = \dot{a} - (\dot{a}_d - 2\lambda e_a + \lambda^2 e_v). \tag{32}$$

Let

$$\alpha = \dot{a}_d - 2\lambda e_a + \lambda^2 e_v \tag{33}$$

and use (20), (30), and (32) to rewrite (31) in the form

$$k|S| \geq S[\hat{b}b^{-1}f - \hat{f} - \alpha(\hat{b}b^{-1} - 1)] + \eta\hat{b}b^{-1}|S|. \tag{34}$$

Taking into account that for $S = 0$ condition (31) is automatically satisfied, from (34) and (23), we obtain

$$k \geq \hat{b}b^{-1}(\varepsilon_f + \eta) + |\hat{b}b^{-1} - 1||\hat{f} - \alpha|. \tag{35}$$

Using (25) and introducing the gain margin

$$\gamma_b = \sqrt{\frac{b_{\max}}{b_{\min}}} \tag{36}$$

with $0 < b_{\min} \leq b \leq b_{\max}$, we find that

$$\hat{b}b^{-1} \geq \gamma_b. \tag{37}$$

Recall also (29) and with the notation (33), we find the final condition for k

$$k \geq \gamma_b(\varepsilon_f + \eta) + (\gamma_b - 1)\hat{b}\,|u_{\mathrm{eq}}|. \tag{38}$$

In conclusion, for k satisfying (38) we are assured that the sliding condition (31) is met.

As a final step, we replaced the *signum* function in (30) with an expression resulting from a fuzzy technique (Ha, 1997)

$$u = \hat{u}_{\mathrm{eq}} - \hat{B}^{-1}k\tan h\left(\frac{S}{\phi}\right), \tag{39}$$

where $\phi > 0$ is used for smoothing the control action between the levels $\hat{u}_{\mathrm{eq}} \pm (\hat{B}^{-1}k)$. Its value has to be adjusted to achieve an optimal balance between the position error, and the level of control chattering.

4. Experimental set-up

The testbed used is part of a 4-DOF manipulator shown in Fig. 2, having the mechanical structure and the functional capability of the manipulators used in various mining operations. Electro-hydraulic on–off valves traditionally used in mining have been replaced with proportional technology, and pressure and displacement sensors have been added to the cylinders. The axis on which the experiments were conducted consists of a double acting, single-ended hydraulic cylinder ($2.5'' \times 1.5''$), driven via a proportional directional control valve. Connecting them are two $\frac{3}{8}''$ hydraulic hoses, approximately 6.5 m long each. Their presence is one of the main characteristics of mobile machinery used in the mining and construction industries which puts additional burden on the controllers. Pressures at both ports are measured using typical transducers, while piston position is measured by an internal LVDT. All controllers were run at a rate of 50 Hz.

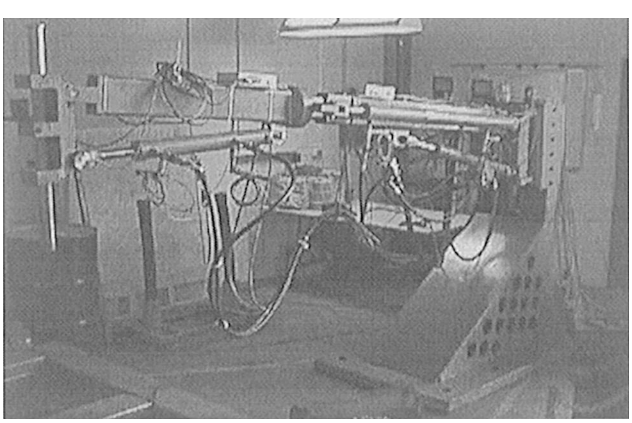

Fig. 2. Experimental mining manipulator.

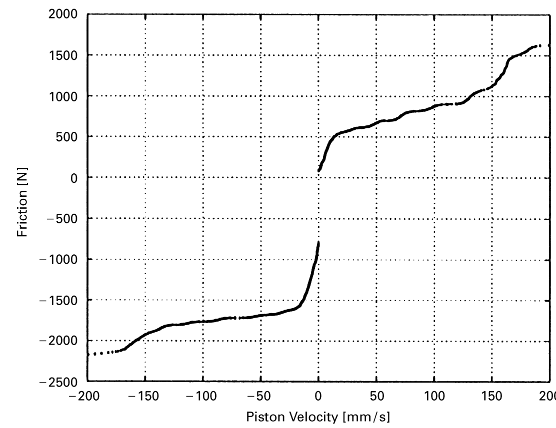

Fig. 3. Measured friction in the hydraulic system.

5. Results

Experimental data obtained for friction identification purposes is visualised in Fig. 3. Friction is plotted here as a function of velocity, and two distinct zones are noticed both in extension and retraction. A sharp increase in friction occurs in areas close to the origin, corresponding to small velocities, whereas for values of velocity above a certain threshold, the rate of increase is one order of magnitude smaller. The particular configuration of seals generates differences between the friction magnitude in extension and retraction. A pressure-based model has been shown to give a good fit (Bonchis et al., 1999)

$$F_{\mathrm{f}}^{+} = 87 + 0.0032(p_1 - p_2) + 0.0012p_2, \tag{40}$$

$$F_{\mathrm{f}}^{-} = -170 + 0.0041(p_1 - p_2) + 0.0014p_2, \tag{41}$$

where F_{f}^{+} and F_{f}^{-} represent the friction in extension and retraction motion, respectively. All variables in (40) and (41) are expressed in SI units. The corresponding root mean square errors between the experimental and predicted friction are 30 N in extension and 12 N in retraction.

The results presented for AFB involve the use of three friction observers. One is the observer based on the experimentally identified friction model (FRID), another one is the Tafazoli observer (TAFO), and the third one is the variable structure observer (VSO). The numerical values of the parameters in TAFO are $\mu = 1.5$, $k_v = 10$, $k_a = 100$, and for VSO are $\sigma_M = 200$, $\gamma_{ey} = 0.1$, $L_1 = 20$, and $L_2 = -100$.

Friction estimates obtained with the observers considered are compared in Figs. 4 and 5. The tuning of the AFB controller resulted in $K_{\mathrm{AFB}} = 0.001$, $\alpha_{\mathrm{AFB}} = 420$, and $\beta_{\mathrm{AFB}} = 90\,000$. Test results for nominal and changed operating conditions are shown in Fig. 6. The effect of changing mass and supply pressure is attenuated in the

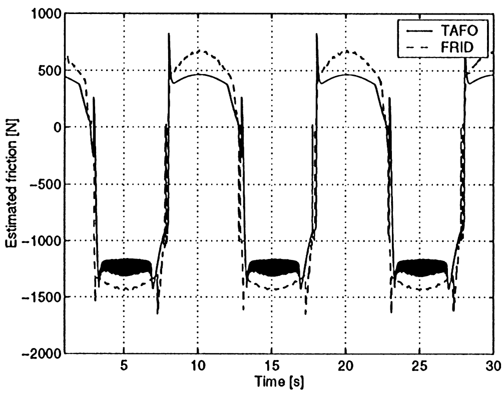

Fig. 4. Comparison between friction estimates of TAFO and FRID.

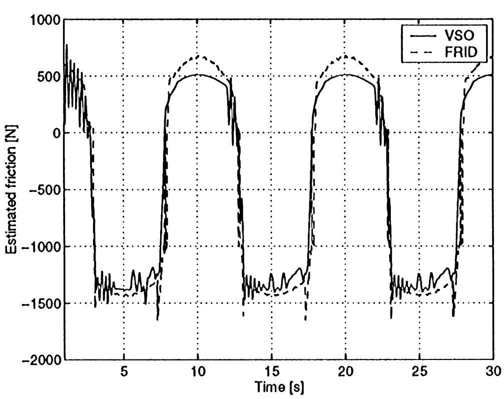

Fig. 5. Comparison between friction estimates of VSO and FRID.

VSO, due to the switching nature of the estimation and a proper choice of σ_M which is large enough to absorb the variations from the nominal values.

We switch our attention to the VSC case, where tuning led to the following set of values for the controller parameters: $\lambda = 3900$, $\eta = 30\,000$, and $\phi = 0.1$. For comparison purposes, a PD controller was also implemented, with the control law

$$u(t) = K_{\mathrm{P}} e_y(t) + K_{\mathrm{D}} \frac{\mathrm{d}e_y(t)}{\mathrm{d}t}, \tag{42}$$

The controller gains found in the tuning phase were $K_{\mathrm{D}} = 4.06$ Vs/m and $K_{\mathrm{P}} = 840$ V/m. The VSC output in the case of nominal plant parameters is presented in Fig. 7 while the resulting position errors are indicated in Fig. 8. The errors obtained with PD control are also shown for comparison. In dealing with VSC one has to remember that error minimisation can be achieved at the cost of considerable control chattering. With the method presented in Section 3 chattering can be significantly reduced. The inherent robustness of VSC was confirmed by the results obtained for step-wise load variation and

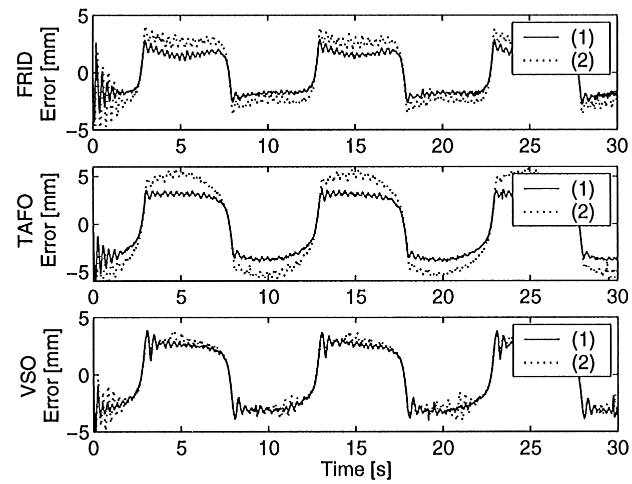

Fig. 6. Robustness test results for acceleration feedback controller with different friction observers: (1) nominal plant and (2) step-wise load variation, reduced supply pressure.

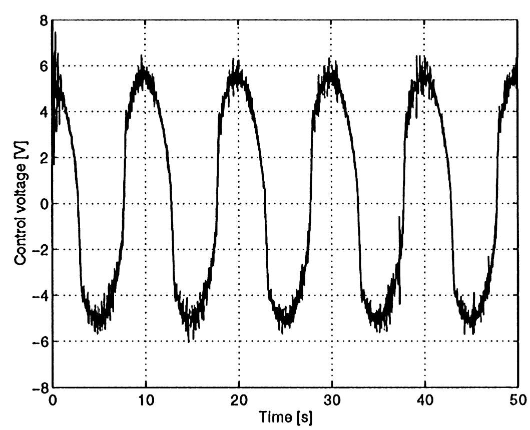

Fig. 7. VSC output for nominal plant parameters.

changed supply pressure, shown in Fig. 9. The level of noise in the system was further increased by injecting a white Gaussian signal at the plant input, with a variance of 0.25 V, which resulted in the control output and the position error shown in Fig. 10. The piston motion is much smoother in extension than in retraction, given the differences in the hydraulic capacitance of the two piston chambers.

6. Conclusions

In the work reported here, we investigated the suitability of variable structure methods for the position control of hydraulic servo systems. Friction is a major sources of nonlinearity, and several techniques have been analysed to address this issue. Model-based friction compensation is not as straightforward as in electrical servo systems.

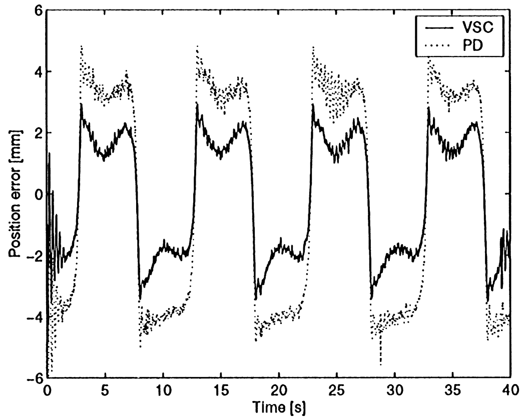

Fig. 8. Position errors obtained with VSC and PD control for nominal plant parameters.

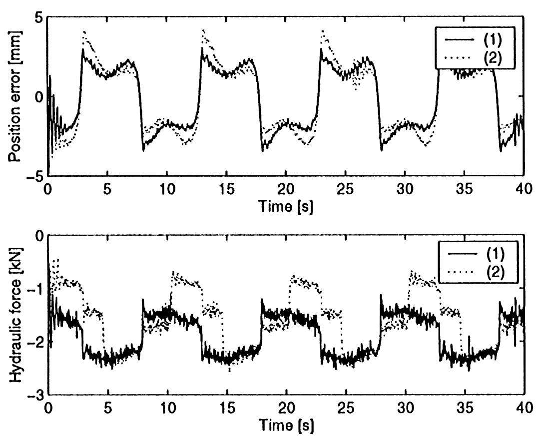

Fig. 9. Tracking results for VSC: (1) nominal plant and (2) step-wise load variation, reduced supply pressure.

Acceleration feedback coupled with a friction estimator provides a solution to the problem. Three ways for providing quantitative information on friction were used. One of them is based on an experimentally identified friction model, a second one was previously reported by Tafazoli et al. (1995), and used successfully in a related application. The third one was a variable structure friction observer reported by Ha et al. (2000). As an alternative to solve the position control problem, we presented a novel implementation of a variable structure control with sliding mode. For comparison purposes a PD control was also tested on the equipment.

Experimental results have shown that VSC produces the best results despite the presence of the unmodeled friction. In model-based friction compensation, the VS observer proved more robust to system parameter changes than the Tafazoli observer. Although the experimentally identified friction model works well in a model-based compensator, there are several associated drawbacks. Firstly, due to variation in manufacturing tolerances, material properties, etc., identification has to

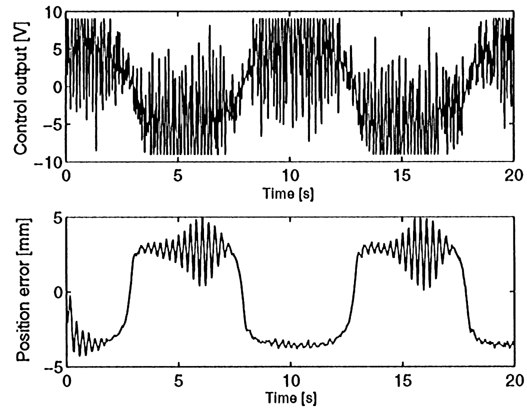

Fig. 10. Tracking with VSC in the presence of additional noise at the input.

be done for each actuator separately, which amplifies the time requirements. Secondly, wear, the aging of materials, temperature fluctuations, result in a variation of model parameters.

Compared with a PD control, all the friction compensation methods investigated resulted in a reduction of the tracking error with at least a factor of 1.6 in extension and 2.2 in retraction. We conclude by advocating for the use of variable structure methods presented here in the control of hydraulic servo systems.

References

Armstrong-Hèlouvry, B., Dupont, P., & Canudas de Wit, C. (1994). A survey of models, analysis tools and compensation methods for the control of machines with friction. *Automatica*, *30*(7), 1083–1138.

Bonchis, A., Corke, P. I., & Rye, D. C. (1999). A pressure based — velocity independent friction model for asymmetric hydraulic cylinders. *Proceedings of the 1999 International Conference on Robotics and Automation*, vol. 3. (pp. 1746–1751). Piscataway, NJ, USA: IEEE Inc.

Friedland, B., & Mentzelopoulou, S. (1992). On adaptive friction compensation without velocity measurement. *Proceedings of the IEEE Conference on Control Applications*, vol. 2. Dayton, OH (pp. 1076–1081).

Fung, R. F., Wang, Y. C., Yang, R. T., & Huang, H. H. (1997). A variable structure control with proportional and integral compensation for electro-hydraulic position servo control system. *Mechatronics*, *7*(1), 67–81.

Ha, Q. P. (1997). Sliding performance enhancing with fuzzy tuning. *IEE Electronics Letters*, *33*(16), 1421–1523.

Ha, Q. P., Bonchis, A., Rye, D. C., & Durrant-Whyte, H. F. (2000). Variable structure systems approach to friction estimation and compensation. *Proceedings of the 2000 International Conference on Robotics and Automation* (pp. 3549–3555). Piscataway, NJ, USA: IEEE Inc.

Hwang, C. L., & Lan, C. H. (1994). The position control of electrohydraulic servomechanism via a novel variable structure control. *Mechatronics*, *4*(4), 369–391.

Slotine, J.-J. E., & Sastry, S. S. (1983). Tracking control of nonlinear systems using sliding surfaces with application to robot manipulators. *International Journal of Control*, *39*(2), 465–492.

Tafazoli, S., de Silva, C., & Lawrence, P. (1995). Friction estimation in a planar electrohydraulic manipulator. *Proceedings of the 1995 American Control Conference,* (pp. 3294–3298) Seattle, Washington.

Tafazoli, S., de Silva, C. W., & Lawrence, P. D. (1998). Tracking control of an electrohydraulic manipulator in the presence of friction. *IEEE Transactions on Control Systems Technology*, *6*(3), 401–411.

Adrian Bonchis received a Bachelor degree in Mechanical Engineering in 1989 from the Politechnica University of Timisoara, Romania. He is completing his Ph.D. in Control Systems at the University of Sydney, Australia, with a thesis on modelling and control of hydraulic servo systems. Since 1998 he has conducted his research at CSIRO Australia, working with the Automation group at the Brisbane laboratory. He is mainly interested in the application of control theory to robotic systems.

Peter Corke completed Bachelor and Masters degrees in Electrical Engineering and a Ph.D. in Mechanical Engineering, all at the University of Melbourne. He is currently a Senior Principal Research Scientist with CSIRO Manufacturing Science and Technology, where he leads the Automation group based at the Brisbane laboratory. His research interests span computer vision, robotics, system modelling and control and distributed real-time computer architectures. Current applications focus is on robotic systems for mining and related industries and includes projects such as autonomous underground vehicles, large excavation machines, and autonomous helicopters. He has previously held visiting positions with the Coordinated Science Laboratory at the University of Illinois at Urbana-Champaign and the GRASP Laboratory at University of Pennsylvania.

D. C. Rye received the B.E. degree (1st class honours) from the University of Adelaide, Australia, and the Ph.D. degree, from the University of Sydney, Australia, all in Mechanical Engineering, in 1980 and 1986, respectively. From 1986 to December 1987, he served as a lecturer in mechanical engineering at the Department of Mechanical Engineering, the University of Newcastle, Australia. Since 1988 he has been with the Department of Mechanical and Mechatronic Engineering, the University of Sydney, Australia, as a lecturer and then senior lecturer in mechanical engineering. Dr. Rye is a Deputy Director of the Australian Centre for Field Robotics. His research interests include mechanics of tracked vehicles, autonomous excavation, crane dynamics and control, mechatronics and automation, nonlinear control, variable structure control, and high-precision positioning control.

Q. P. Ha received the B.E. degree in Electrical Engineering from Ho Chi Minh City Polytechnic University, Vietnam, the Ph.D. degree in Engineering Science from Moscow Power Institute, Russia, and the Ph.D. degree in Electrical Engineering from the University of Tasmania, Australia, in 1983, 1992, and 1997, respectively. From 1997 to 2000, he was a senior research associate at the Centre for Field Robotics, the University of Sydney. He is currently a lecturer at the University of Technology, Sydney, Australia. His research interests include nonlinear control, variable structure systems, robotics, and applications of artificial intelligence in engineering.