A Comparative Study of Variable Structure and Model Reference Adaptive Control for Hydraulic Servo Systems

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Abstract

The framework of this study is computer-assisted control of hydraulic actuated mobile machinery, such as mining manipulators and earth moving equipment. The design of both methods is treated and experimental results are presented. Several criteria for comparison are detailed to assist in a rational decision making process. In this paper, two different approaches have been taken in an attempt to address the issue from a deterministic standpoint. One of them is a variable structure controller with sliding modes, and the other one is a model reference adaptive control.

1 Introduction

Feedback systems are intrinsically insensitive to modeling errors and disturbances. While this is a fundamental property, in practice, the degree with which they can cope with uncertainty varies significantly. Factors affecting their performance include, but are not limited to, the nature of the plant dynamics, the type of the control scheme, the characteristics of various disturbances affecting the system. Ultimately, the only real freedom a control engineer has is the choice of the control algorithm.

The work presented here was centred around the problem of controlling heavy-duty hydraulic servo systems, typically found in mining manipulators and earth moving equipment. Fuelled by theoretical developments, the last decade has seen significant results in the control of such systems. Two major methods have made a big impact: model reference adaptive control (MRAC) and variable structure control (VSC) with sliding modes. Applications of model reference adaptive control to electro-hydraulic position control systems, such are those reported in [2], [8], and [4], have been soon followed by applications of the variable structure control [5], [7], [3]. The later ones were however dealing with hydraulic motors, although in practice asymmetric hydraulic cylinders are in widespread use. Previously [1], we reported on a robust sliding mode controller for a hydraulic system with such cylinders. Although the results obtained were notable, to our knowledge there is no direct comparison between MRAC and VSC. With this paper, we try to share the experience we had in applying them both to the same testbed. There is a direct benefit to the person(s) involved in hydraulic control, as we assess their suitability against a number of criteria.

The following section describes the configuration of the hydraulic test rig used. In Section 3, we present the algorithm for the model reference adaptive controller. The system under test is SISO, with the strictly positive real condition failing to be met. As such we followed the approach set in [6], where the stability of the parameter adjustment mechanism is guaranteed by the introduction of the augmented error concept. The major points in the development of the variable structure controller are presented in Section 4, a detailed account of our work in this area can be found in [1]. The results in Section 5 open the way for the enumeration of several assessment criteria and a discussion on the suitability of each of the controllers for practical purposes. We then end by drawing several conclusions in Section 7.

2 System configuration

The testbed used is part of a 4–DOF generic machine having the mechanical structure and the functional capability of the existing mining manipulators used in rock breaking and roof bolting operations. Standard off the shelf commercial components have been used in order

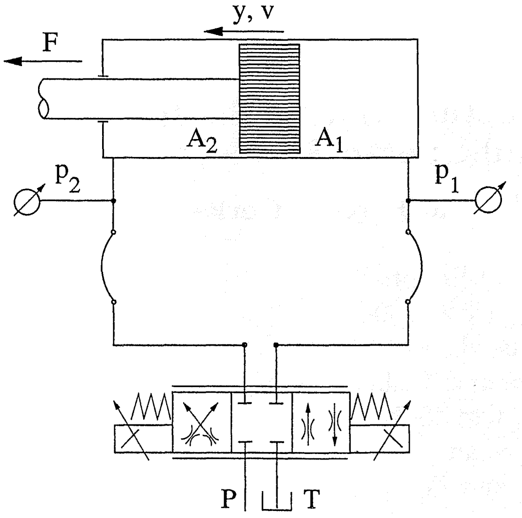


Figure 1: Experimental setup.

to preserve the resemblance with the machines currently operating underground. The only difference is the replacement of the electro-hydraulic on-off valves with proportional technology, and the addition of pressure and displacement sensors. The rough environment in which these machines are required to operate makes the proportional valves more suitable than servo valves, in spite of the enhanced performance offered by the later. Demands for high level of filtration and the cleanliness required when performing maintenance on servo valves are practically impossible to achieve in a mining environment, on a reasonable economic basis.

The axis on which experiments were conducted consists of a double acting, single-ended hydraulic cylinder (2.5" x 1.5"), driven via a proportional directional control valve., as shown in Figure 1. Connecting them are two 3/8" hydraulic hoses, approximately 6.5m long each. This 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 50Hz.

3 The model reference adaptive controller

For the system mentioned above the reference model chosen is $y_m = M(s) \cdot r$ with

$$M(s) = \frac{10000}{s^3 + 80s^2 + 1700s + 10000} \qquad (1)$$

and r the reference signal. This corresponds to a bandwidth of only 3Hz, and for comparison we should mention that the manufacturer indicates for the proportional valve a bandwidth of 5Hz. Let the plant be described by a transfer function of the form $P(s) = K_p \cdot N(s)/D(s)$. We are in the safe area as far as the general requirements for MRAC are concerned: the relative degree n^* of the plant must be known and $M(s)$ has the same relative degree; the order of the plant is known and the plant characteristic polynomial is monic; the plant and model are completely observable and controllable; plant and model high frequency gains are of the same sign; the plant is minimum phase. There could be a doubt related to the last condition, usually hydraulic systems are non-minimum phase, partly due to the delay in the transmission lines (and cylinder for long strokes), and partly because sampling could make things worse by introducing unstable zeros. However, methods for dealing with such cases are available, see [8] and [4].

The following I/O filters are used

$$\dot{\omega}_1 = \Lambda\, \omega_1 + \mathbf{h}\, u; \quad \dot{\omega}_2 = \Lambda\, \omega_2 + \mathbf{h}\, y \qquad (2)$$

with $\omega_{1,2} \in \mathbb{R}^2$, $\mathbf{h} \in \mathbb{R}^2$, $\Lambda \in \mathbb{R}^2$ such that $(s - \Lambda)$ is a Hurwitz polynomial of 2nd order. With this choice, the desired zero of the reference model can be imposed. For the vector of adaptive parameters

$$\boldsymbol{\theta}^T(t) = [\ \theta_1(t) \quad \boldsymbol{\theta}_2(t) \quad \boldsymbol{\theta}_3(t) \quad \theta_4(t)\ ] \qquad (3)$$

and the corresponding regressor vector

$$\omega^T(t) = [\ r(t) \quad \omega_1(t) \quad \omega_2(t) \quad y(t)\ ] \qquad (4)$$

the control u is defined as

$$u = \boldsymbol{\theta}^T(t)\omega(t) \qquad (5)$$

The auxiliary error

$$\eta(t) = \boldsymbol{\theta}^T(t)M(s)[\omega] - M(s)[\boldsymbol{\theta}^T(t)\omega(t)] \qquad (6)$$

together with the output error $e(t) = y - y_m$ defines the augmented error

$$\epsilon(t) = e(t) + \alpha(t)\eta(t) \qquad (7)$$

in which $\alpha(t)$ will be determined by adaptation. Using the gradient method with normalisation, the controller parameters $\boldsymbol{\theta}(t)$ and $\alpha(t)$ will be updated according to

$$\dot{\boldsymbol{\theta}}^T = -\frac{\gamma\epsilon\omega_f}{1 + \omega_f^T(t)\omega_f(t)} \qquad (8)$$

$$\dot{\alpha}(t) = -\frac{\gamma\epsilon\eta}{1 + \omega_f^T(t)\omega_f(t)} \qquad (9)$$

where $\omega_f = M(s)[\omega]$.

4 The variable structure controller

In the process of minimising the output error, the model reference adaptive control is actually "forcing" the hydraulic system to behave in a linear manner, although significant nonlinearities are present. If their existence is taken into account at the onset of the controller design, some other control algorithms would have to be considered. This is where variable structure control comes under the spotlight, as one of its major advantages is robustness to unmodeled dynamics. Based on a priori knowledge about the influence of various sources of nonlinearities, a 3rd order model is proposed, which will be expressed directly in control canonical form. This is considerably simpler than the 6th order model which has been devised previously for the system under investigation [1]. The development of the VS controller will be exemplified for the case when the piston extends. The treatment of the piston retracting case is similar.

$$\dot{a}_y = \frac{1}{M_p} \cdot (\dot{p}_1 A_1 - \dot{p}_2 A_2) \tag{10a}$$

$$\dot{p}_1 = \frac{B}{V_{L1} + y \cdot A_1} \cdot (K_q \sqrt{p_s - p_1} u - v_y A_1) \tag{10b}$$

$$\dot{p}_2 = \frac{B}{V_{L2} + (S - y) \cdot A_2} \cdot (-K_q \sqrt{p_2 - p_t} u - v_y A_2) \tag{10c}$$

where:

A_i – piston area in both chambers, i=1,2
B – oil bulk modulus
M_p– inertial mass of piston, rod, and load
p_i – pressure inside actuator chambers, i=1,2
p_s – supply pressure
p_t – tank pressure
u – the control signal
S – piston stroke
V_{Li}– ineffective volume, i=1,2
v_y – piston velocity
a_y – piston acceleration
y – piston displacement

with K_q representing a constant flow coefficient, which can be approximated using data supplied by the valve manufacturer.

When the piston is in extension, i.e. $v_y \geq 0$, substituting (10b) and (10c) in (10a), we get in compact form:

$$\dot{a}_y = f(y, v_y) + b(y, v_y)\, u \tag{11}$$

where

$$f = -\frac{1}{M_p} \cdot [\beta_1(y) A_1^2 + \beta_2(y) A_2^2] \cdot v_y \tag{12}$$

$$b = \frac{K_q}{M_p} \cdot [\beta_1(y) A_1 \sqrt{p_s - p_1} + \beta_2(y) A_2 \sqrt{p2 - pt}] \cdot v_y \tag{13}$$

$$\beta_1 = \frac{B}{V_{L1} + A_1 y}\ ,\ \beta_2 = \frac{B}{V_{L2} + A_2(S - y)} \tag{14}$$

The system dynamics f cannot be exactly known but estimated by $\hat{f}$. We hope to find a bound for the estimation error

$$| f - \hat{f} | \leq \epsilon_f \tag{15}$$

where the bound ϵ_f could be state dependent, i.e. $\epsilon_f = \epsilon_f(y, v_y)$. Assuming that the piston areas $A_{1,2}$ and the piston mass M_p are known with sufficient accuracy, an expression for ϵ_f is found of the form

$$\epsilon_f = -\frac{B}{M_p} \cdot (\frac{A_1^2}{V_{L1}} + \frac{A_2^2}{V_{L2}}) \cdot max(| \dot{y} - \dot{y}_d |) \tag{16}$$

the index d denoting reference values. The control input gain is also unknown exactly, but can be estimated. Due to the multiplicative effect, the estimate $\hat{b}$ is chosen as the geometric mean of the bounds b_{min} and b_{max}

$$\hat{b} = \sqrt{b_{min} \cdot b_{max}} \tag{17}$$

By manipulating equation (13), followed by a back substitution, we get

$$\hat{b} = \frac{K_q B}{M_p} \sqrt{\frac{1}{2} \sqrt{p_{vmin} p_{vmax}}} \sqrt{max(g) min(g)} \tag{18}$$

where

$$g(y) = \frac{A_1}{V_{L1} + A_1 y} + \frac{A_2}{V_{L2} + A_2(S - y)} \tag{19}$$

and p_{vmin}, p_{vmax} represent the minimum and maximum valve pressure drops.

For the state error vector

$$\mathbf{e} = [\ e_y \quad e_v \quad e_a\ ]^T \tag{20}$$

with the components $e_y = y - y_d$, $e_v = v_y - v_{yd}$, and $e_a = a_y - a_{yd}$, we define a scalar time-varying surface $S(\mathbf{e}, t) = 0$, with S being

$$S(\mathbf{e}, t) = \left(\frac{d}{dt} + \lambda\right)^2 \cdot e_y, \quad \lambda > 0 \tag{21}$$

which results in

$$S(\mathbf{e}, t) = e_a + 2\lambda e_v + \lambda^2 e_y \tag{22}$$

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

$$u_{eq} = \hat{b}^{-1}[-\hat{f} + \dot{a}_{yd} - 2\lambda e_a + \lambda^2 e_v] \tag{23}$$

To accommodate the estimation errors, a discontinuous term is added to (23)

$$u = u_{eq} - \hat{b}^{-1}k \cdot sgn(S) \tag{24}$$

The stability analysis is used to determine the bounds on k, see [1] for a detailed discussion.

5 Experimental results

We assessed experimentally the performance of the controllers in point-to-point positioning tasks. A reference signal was designed using a trajectory generator based on a quintic polynomial. Different kinds of demands were generated, some of them resulting in small displacements close to the limits of the piston stroke, while others involved moving the piston over a distance close to full stroke. All these allow for the inclusion of the nonlinearities associated with the volume variation of the two oil columns trapped in the cylinder chambers.

For the adaptive controller, the tracking results for constant load and given supply pressure are shown in Figure 2(a), while Figure 2(b) shows the controller parameters variation.

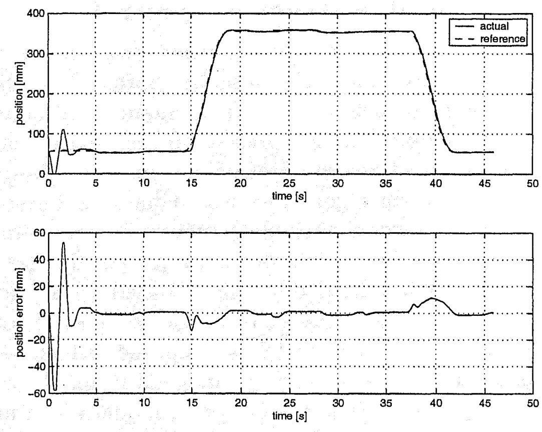
The same experimental conditions, when using the variable structure controller, produced the results in Figure 3, with the path following and position errors in Figure 3(a), while the corresponding control input is shown in Figure 3(b).

To test the sensitivity to parameter variation, we conducted experiments in which the previous load was applied step–wise, and with the supply pressure decreased by 25%. Corresponding results for the two different controllers are presented in Figures 4 and 5. Looking at the force variation in both plots, a jump of approximatively 1000N can be noticed at moments when the load changes.

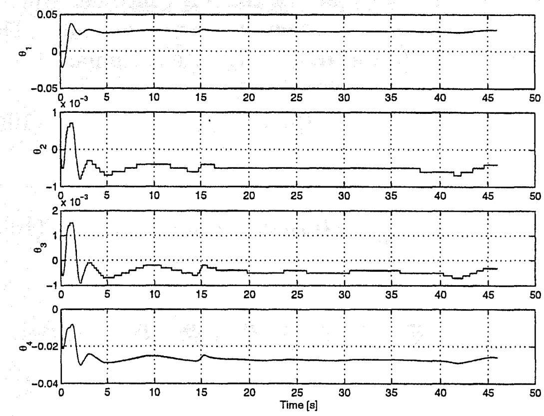
A model reference adaptive controller based on a 2nd order model was also tried. The model used was

$$M(s) = \frac{1000}{s^2 + 70s + 1000} \tag{25}$$

The convergence of the parameters in this case displayed significant oscillations at motion start compared to the 3rd order model, as shown in Figure 6. However, later on during the motion, when high velocity demands were generated, position errors were significantly reduced (compare with Figure 2(a)).



(a) Path following and error

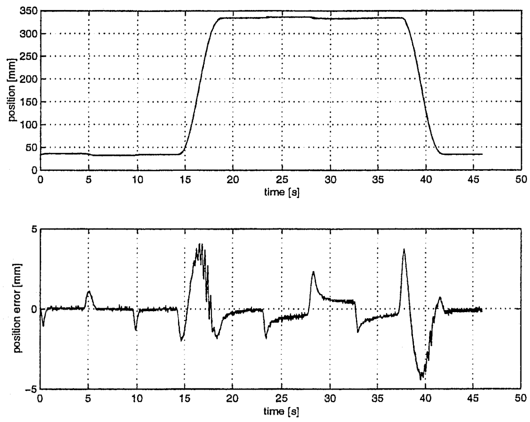


(b) Parameter variation

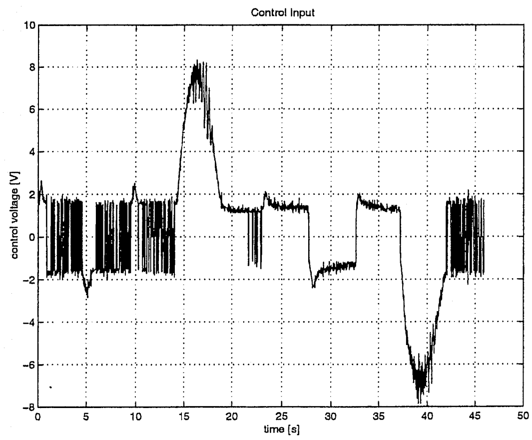
Figure 2: Tracking with constant load using MRAC.

6 Assessment of controllers

There are several criteria against which the suitability of the controllers can be assessed, and it is the responsibility of the control engineer to arrange them according to his/her priorities and/or constraints. For the position control problem, the controller accuracy would be naturally described by some metric involving the position error $e(t)$. We have chosen the root mean squared position error RMS(e), and the values obtained were 2.2 mm for VSC and 12 mm for MRAC. Ignoring however the initial transients in MRAC, caused by the lack of information on the initial values of controller parameters, the RMS decreases to 2 mm. Hydraulic servo systems are highly nonlinear and VSC seems to fit well in terms of tracking performance. It uses a nonlinear description of the system, and inherently, it is robust to unmodeled dynamics. Intuitively, we would expect MRAC to present us with problems, as it tries to "force" a plant, which includes significant nonlinearities, to behave in a linear manner.



(a) Path following and error



(b) Computed control

Figure 3: Tracking with constant load using VSC.

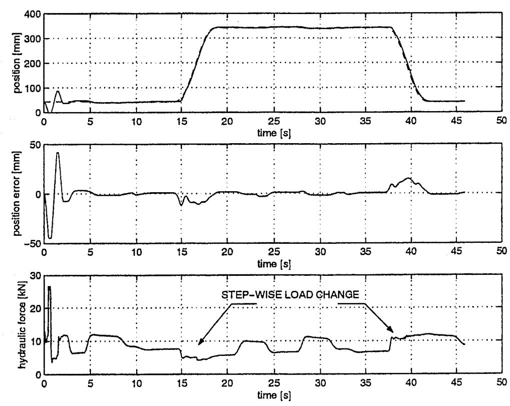


Figure 4: Tracking with varying and changed supply pressure load using MRAC.

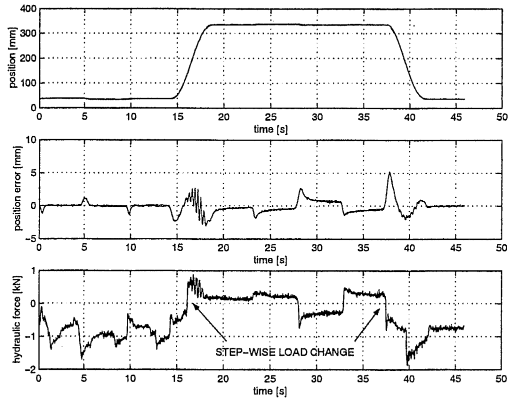


Figure 5: Tracking with varying and changed supply pressure load using VSC.

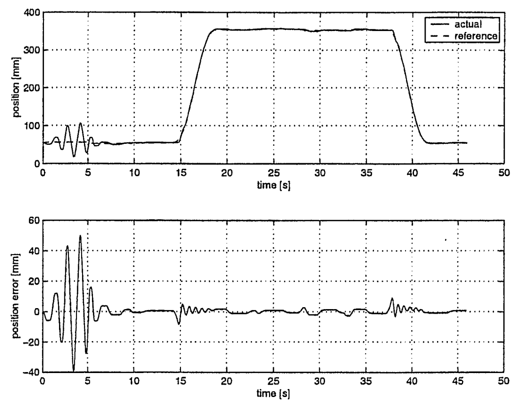


Figure 6: Tracking with MRAC based on a 2nd order model.

The robustness of the controllers can be assessed by looking at the relative variation of RMS(e) between the experimental results corresponding to the nominal and changed plant parameters. In the VSC case, RMS(e) varied with 12% while for MRAC the change was around 26% (excluding initial transients). Note that the experiments were dealing with fast changes. However, things can be different when variations are slow, and in time certain parameters could change such that the uncertainty boundaries used by VSC are no longer valid. On the other hand, MRAC would be able to cope with such changes.

Another important aspect issue is the development effort. The modeling and identification effort takes its toll on the time required for VSC design and implementation. Parameters in the model need to be specified in terms of nominal values and their associated error bounds. In addition, VSC requires complete state information. From this perspective, MRAC is less demanding, requiring the structure of the model only. The choice of reference model is however critical for the controller performance. The level of complexity in both algorithms is similar, the computation times per cycle being on average 8 ms on a Pentium 200 MMX, but in terms of hardware requirements, VSC requires two pressure sensors and one force sensor per each DOF, in addition to the cylinder extension sensor.

A point that often raises concerns in VSC applications is the chattering affecting the control signal, which occurs inevitably as we aim for smaller positioning errors. Control chattering has to be avoided because in general it could excite unwanted frequencies and in particular here because of the valve solenoids overheating. In turn, this could alter the solenoid force characteristic, with a loss of the proportionality between force and current. However, the two opposed design criteria were balanced within acceptable limits. The control signal computed and sent to the servo–amplifier is a voltage, which is subsequently transformed into a command current for the solenoids. As a result, the circuit inductance acts as a factor limiting the chattering effect.

Before ending this discussion, remarks have to be made about the theoretical requirements for the application of MRAC. Of concern here is the minimum phase condition imposed on the plant in the case of the standard algorithms. With the inclusion of transmission lines, and for long stroke cylinders, hydraulic servo systems are affected by delay. Additionally, unstable discrete–time zeros can be introduced by the sampling process. Choosing the sampling interval large enough to produce stable sampling zeros might interfere with the adaptive controller, whose performance will degrade. There are however methods to circumvent this problem [8, 4], which have been applied to electrohydraulic servo systems.

7 Conclusions

There are many possible avenues to pursue when faced with the task of implementing controllers for hydraulic servo systems. Model reference adaptive control and variable structure control with sliding mode have definitely made their mark in practice. However, a number of questions arise with respect to using one or the other in a given situation. This paper tries to provide answers for them, as we present the experiences we had with implementing both controllers on a hydraulic servo system, characteristic of mining manipulators and earth moving equipment. Strengths and weaknesses of each of them are presented in parallel, and as far as we know, such a direct comparison was not available before. The benefit lies directly with the person responsible for implementing the controllers, as he or she is armed with all the knowledge needed in the selection process.

Acknowledgements

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