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Position Control based ASMC for Hydraulic Cylinder Actuator

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***Abstract*—Conventionally, a method utilized for controlling the position of hydraulic cylinder actuator is a highly difficulty. A friction force is occurred while the movement of the hydraulic cylinder actuator. This paper proposes a position control based adaptive sliding mode control (ASMC). The majority of this work is a mathematical model of a hydraulic cylinder actuator that includes a linear and non-linear function. A method of ASMC design is then presented and developed in order to compensate the uncertainty of linear and non-linear function. The experimental results are compared to the actual positions and the trajectory positions. The different positions were less than 100 micrometers which is acceptable practically.**

***Keywords—** Adaptive control; Friction force; Non-linear function; Sliding mode control; Hydraulic cylinder actuator*

I. INTRODUCTION

Nowadays hydraulic cylinder actuator is widely used in application of modern industry. In general, the application works perfectly for 2 stages, there are forwards and backwards moving. After that it's developed to be able a position and velocity control. The concept of position and velocity control are applied to gain more flexibility. Nevertheless, a problem of operation is friction force occurred at the beginning of the movement. Friction force makes the position and velocity control of the hydraulic cylinder actuator to difficult complex. To address the issue, [1-4] proposed the control of the hydraulic cylinder actuator. However, the authors have ignored the friction force, giving rise to controlling errors.

This paper presents the adaptive PD controller designed based on sliding mode [5] for hydraulic cylinder actuator. The proper mathematical model of the hydraulic cylinder actuator composed of linear and non-linear functions. Moreover, the design of control system can perfectly work to be able to compensate the linear function using the PD controller. Additionally, the proposed controller can compensate the non-linear function of the friction force using adaptive controller. The proposed controller responses the position and velocity of the hydraulic cylinder actuator to be most convergent towards the referent signal.

The friction force has to be controlled its position and velocity to the hydraulic cylinder actuator drive system. The position and velocity controller of hydraulic cylinder actuator is designed to send the control signal to the hydraulic servo valve and to the hydraulic cylinder actuator. For the position feedback signal, it uses a linear scale encoder. The hydraulic cylinder actuator structure composes of friction force. Such a friction force was a non-linear function and unknown parameter [6]. Consequently, the linear control may be not sufficient for the operational control.

II. MATHEMATICAL MODEL

A. Physic Model

The hydraulic cylinder actuator drive system consists of hydraulic cylinder actuator, hydraulic servo valve, counter balance valve, fine filter, coarse filter, and hydraulic pump, as shown in fig. 1. The movement of hydraulic cylinder actuators has 2 parts, forward and backward, thus determination of the mathematical model must apply with friction force to be for the integrated analysis.

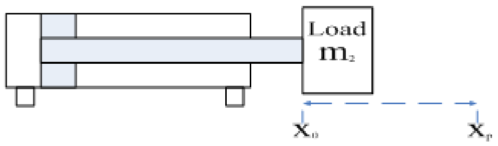

Figure. 1 A diagram of hydraulic cylinder actuator

Hence, the equivalent of the mathematical model shown as (1) [7-9],

$$(m_1+m_2)\ddot{x}+d\dot{x}+f = u(t) \qquad (1)$$

where

m_1 is mass of piston rod.
m_2 is mass of loaded.
f is friction force.
$\ddot{x}$ is acceleration of piston rod.
$\dot{x}$ is velocity of piston rod.
x is position of piston rod.
d is damping coefficient.
$u(t)$ is control signal.

From (1), a simple way used for design of the position controller, the objective of the controlling is to control the position and the initial velocity movement its smoothness with total stable operational mass of no change, $(m_1+m_2) = m_t$ practical new equation shown as follows;

$$m_t\ddot{x}+d\dot{x}+f = u(t) \qquad (2)$$

B. Friction Force Characteristics

In the hydraulic cylinder actuator drive system, the hydraulic cylinder actuator presents friction force between the surfaces of the objects. From the analysis, it shows that the proposed friction force [6,10] would be composed of static friction, coulomb friction, and viscous friction. Therefore, presented frictions force is non-linear function, as shown in fig. 2. Such the static friction will be operated in a case of no movement. However, when it is moved, the static friction will become null. Afterwards, the coulomb and viscous frictions are shown while the coulomb friction presented from the reaction of the tangential surfaces of the objects, and the viscous friction presented while the work piece or the loaded object presents its movement. Such friction force varies directly to velocity. The presented friction force would effect to velocity. The equation of friction force that used to the position controlling shown as the following equation;

$$f = f_{s-}(1-\mu) + f_{s+}.\mu + f_{c-}(1-n) + f_{c+}.n + f_v\dot{x} \qquad (3)$$

The co-efficiencies of the friction occurrence

$$\mu = 1; \dot{x} = 0_{+} \quad \mu = 0; \dot{x} = 0$$
$$\eta = 1;\ \dot{x} < 0 \quad \eta = 0;\ \dot{x} > 0$$

where

f_{s-} is static friction of negative directions.
f_{s+} is static friction of positive directions.
f_{c-} is coulomb friction of negative directions.
f_{c+} is coulomb friction of positive directions.
f_v is viscous friction.

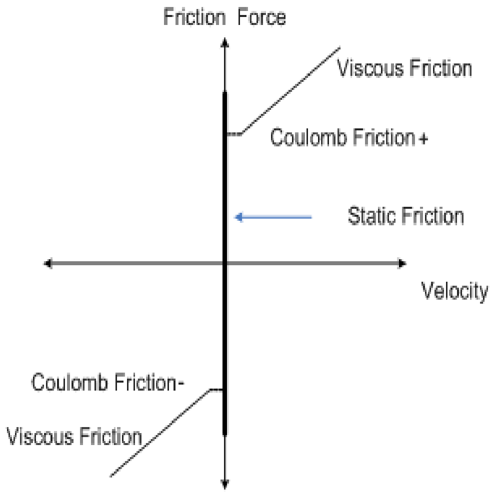

Figure. 2 A Characteristic of total friction

A convenient way is used for analysis of the friction force according to the property of the coulomb friction, the remained value of coulomb friction shown as the following (4), and could be replaced with $f = f_c\,\mathrm{sgn}(\dot{x})$. The viscous friction directly varying to its velocity will be added to the viscous friction as $d_t = d + f_v$. The static one is occurred while the velocity becomes null, and when static one will be disappear, consequently, the force will be ignored. Substituting the frictions into (2), resulting in (4);

$$m_t\ddot{x} + d_t\dot{x} + f_c\,\mathrm{sgn}(\dot{x}) = u(t) \qquad (4)$$

III. CONTROLLER DESIGN

The position control design will be done as adaptive controller [11,12], by adaptation law using Lyapunov [12] function which its derivatives follow its practical condition to produce its stable system. The design of the controller will be done by defining the controlling condition that could feed input signal for the hydraulic cylinder actuator drive system as follows;

$$u(t) = \hat{m}_t x_r + \hat{d}_t x_r + \hat{f}_{c+}.\eta + \hat{f}_{c-}.(1-\eta) - K_D E \qquad (5)$$

where

$\hat{m}_t$, $\hat{d}_t$, $\hat{f}_{c+}$ and $\hat{f}_{c-}$ are the estimate parameters of the controller.

$K_D E$ is the gain of the linear controller.

While $E = \dot{\tilde{x}} + \lambda\tilde{x}$ and define $\tilde{x} = x - x_d$, where x_d is the trajectory position. $K_D E$ is equal the gain of the PD controller, while x_r is the reference position shown as the following equation;

$$x_r = \dot{x}_d - \lambda\tilde{x} \qquad (6)$$

Then, the equation of the closed system will be defined replacing (6) into (5), becomes as follows;

$$m_t E + (K_{D1} + \lambda)E = Y\tilde{p} \qquad (7)$$

where

$$Y = [\dot{x}_r \ \ x_r \ -\eta \ -(1-\eta)]$$
$$\tilde{p} = [\tilde{m} \ \ \tilde{d} \ \ \tilde{f}_{c+} \ \ \tilde{f}_{c-}]^T$$
$$\tilde{m} = \hat{m}_t - m_t$$
$$\tilde{d} = \hat{d}_t - d_t$$
$$\tilde{f}_{c+} = \hat{f}_{c+} - f_{c+}$$
$$\tilde{f}_{c-} = \hat{f}_{c-} - f_{c-}$$

From (7), the closed system equation of the driving set of the hydraulic cylinder actuator, moreover, consideration of the stability of the system by using Lyapunov function and rearranging the equation as follows;

$$V = \frac{1}{2}m_t E^2 + \frac{1}{2}\tilde{p}^{\,T}\Gamma\tilde{p} \qquad (8)$$

where $V > 0$, and Γ is the *positive symmetric matrix*, the derivative of (8) would be executed when defining $\dot{V} \le 0$ will be result in the rule of the adaptation gain as follows;

$$\dot{\tilde{p}} = -\Gamma^{-1} Y^T E \qquad (9)$$

define

$$\Gamma^{-1} = diag[\gamma_1 \ \gamma_2 \ \gamma_3 \ \gamma_4]$$

where γ_n are the adaptation gain and positive definite ($n = 1..4$), from (8), it describes the adaptation equation as following;

$$\dot{\tilde{m}} = -\gamma_1 \dot{x}_r E$$
$$\dot{\tilde{d}} = -\gamma_2 x_r E$$
$$\dot{\tilde{f}}_{c+} = \gamma_3 E \eta$$
$$\dot{\tilde{f}}_{c-} = \gamma_4 E(1-\eta)$$

IV. EXPERIMENTAL RESULTS

A. Hardware Setup

On the experiment of the position controlling of the hydraulic cylinder, the Bosch rexroth servo valve is applied at the experimental velocity of 100 mm per minute at the displacements of 100, 300, and 500 millimeters, and the position feedback signal used the linear scale encoder of Heidenhain model Rod 426, to find the efficiency of the position controlling of such tested hydraulic cylinder actuator, as shown in the fig. 3.

B. Trajectory Generator

This paper shows the experimental result for confirmation of the effectiveness of ASMC for hydraulic cylinder actuator. In experiments, the sampling time is 100 milliseconds, the initial position is 0, the designed positions are 300 and 500 millimeters, the initial velocity is 0 millimeter per second, the designed velocity is 1 millimeter per second and the t_f is 0.58 second.

The argument function $x_d(t)$ is defined as

$$x_d(t) = \begin{cases} x_0 + x_1 t + x_2 t^2 + x_3 t^3 & if \ 0 < t < t_f \\ 0 & if \ t \geq t_f \end{cases} \qquad (10)$$

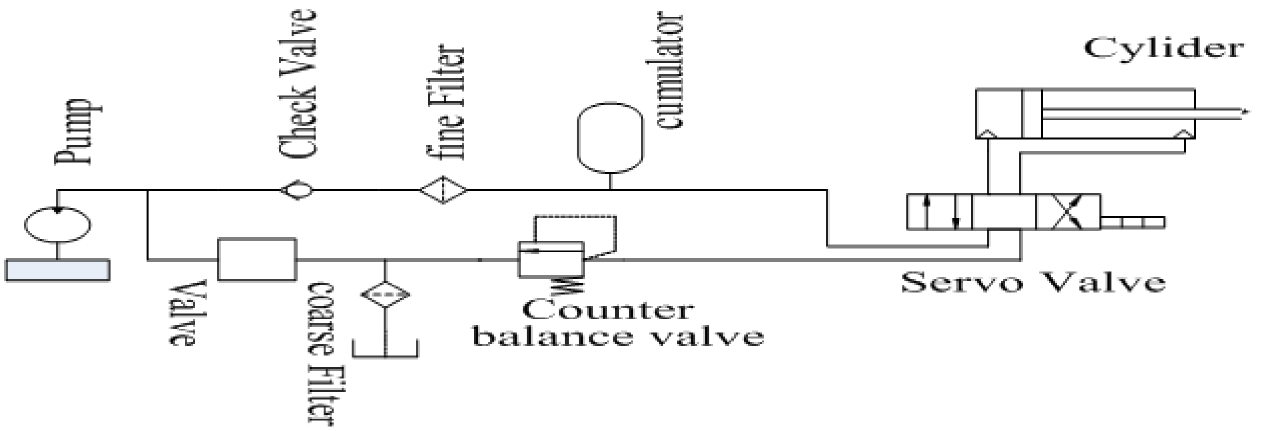

Figure. 3 A block diagram on the experimental

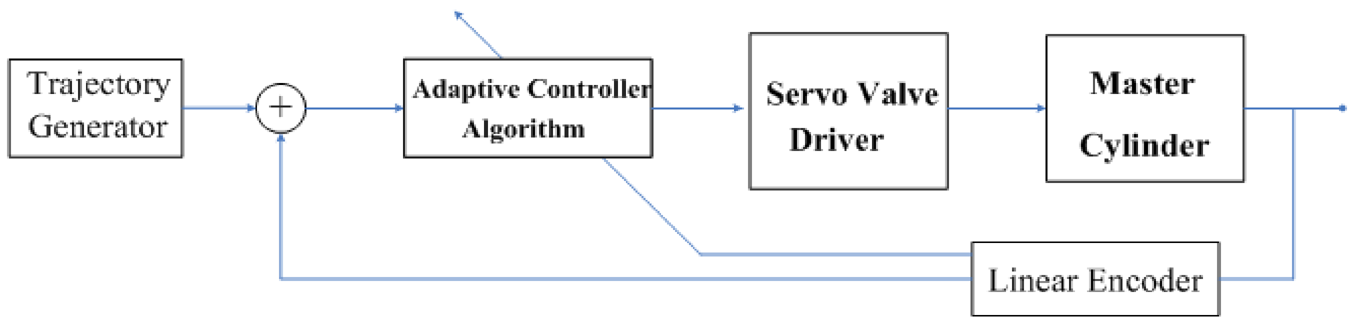

Figure. 4 A block diagram of ASMC on the experimental

C. Experimental Results

The position control of the hydraulic cylinder actuator shown the result in graph to compare the position,and difference position. The results show that the position as show in the fig. 5-7, the result of the position and velocity controlling experiment. The results of the differences of the position show in the fig. 8-10. The experiment result shows the good response which is acceptable practically. The acceptable differences position is less than 100 micrometers while the applied parameters of the controller were equal K_D = 3.78, λ = 0.871, the expansion rate of the adaptation gains are shown in table I.

Table I. Adaptations gain

Adaptation gain	Values
γ_1	0.034
γ_2	0.0068
γ_3	0.0032
γ_4	0.0038

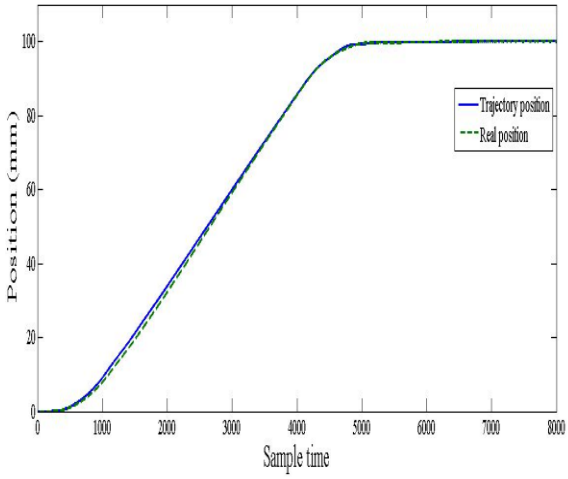

Figure. 5 Comparison between the trajectory position and the actual position at 100 millimeters.

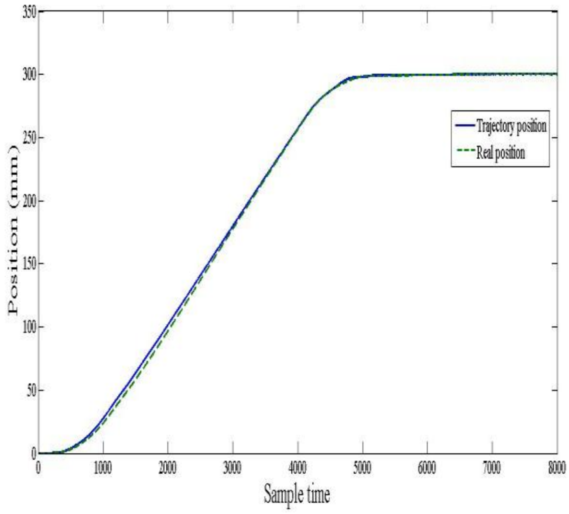

Figure. 6 Comparison between the trajectory position and the actual position at 300 millimeters.

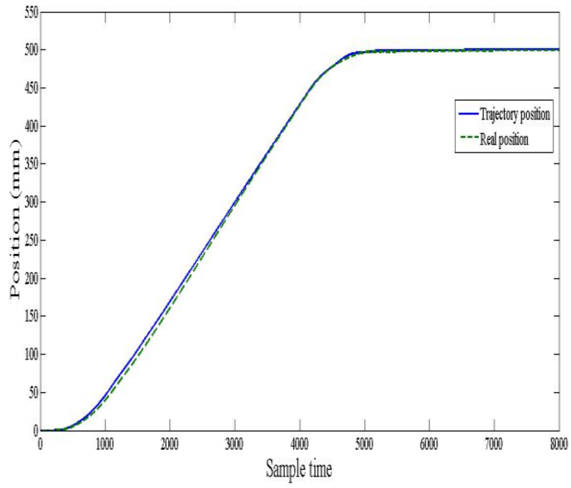

Figure. 7 Comparison between the trajectory position and the actual position at 500 millimeters.

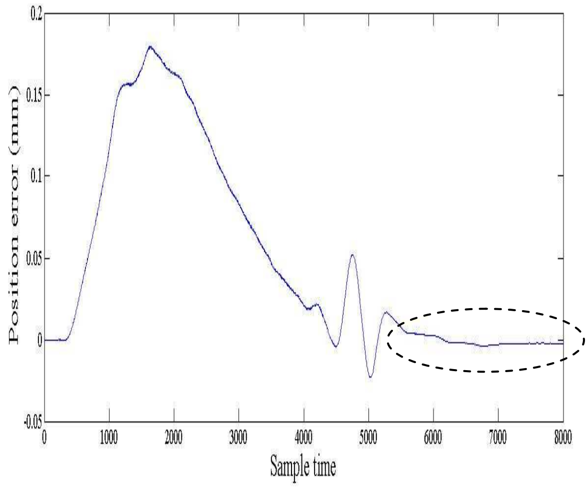

Figure. 8 The difference between the trajectory position and the actual position at 100 millimeters.

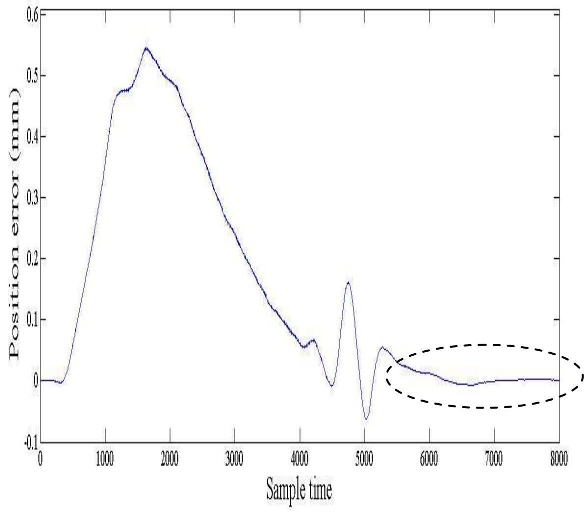

Figure. 9 The difference between the trajectory position and the actual position at 300 millimeters.

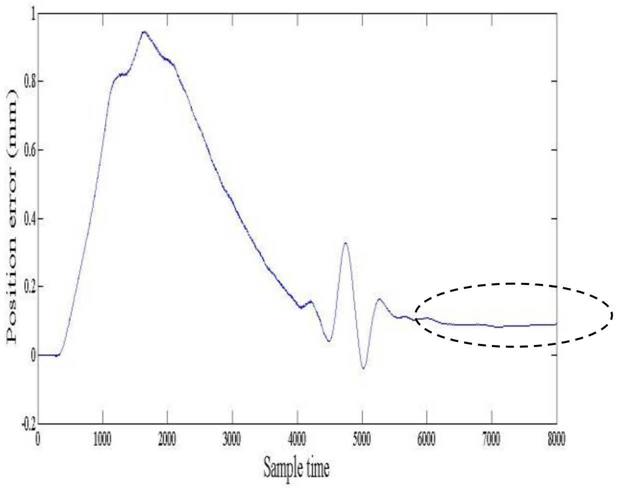

Figure. 10 The difference between the trajectory position and the actual position at 500 millimeters.

V. CONCLUSION

The presentation of the study has been done on the design of the position controlling of the adjustable hydraulic cylinder actuator by having the design of the controller that compensate the linear and non-linear functions using the Lyapunov method. The experimental results have been presented the comparisons of the responsive positions, and displacement differences of the hydraulic cylinder actuator and its target values. It makes the movement of the hydraulic cylinder actuator provide the acceptable displacement values. Moreover, the controller can control the initial smoothly movement.

REFERENCES

[1] Shih, M.C., Tsai, C.P. Servo hydraulic cylinder position control using a neuro-fuzzy controller. Mechatronics, (1995).vol. 5, no. 5, p. 497- 512.

[2] Edvard Deticek, Uros Zuperl. An Intelligent Electro-Hydraulic Servo Drive Positioning, Journal of Mechanical Engineering 57 (2011) 5, 394-404

[3] Ho Triet Hung, and Ahn Kyoung Kwan. A study on the position control of hydraulic cylinder driven by hydraulic transformer using disturbance observer. Control, Automation and Systems, 2008. ICCAS 2008. International Conference on, 14-17 Oct. 2008 p.2634 – 2639

[4] Lei Si, Zhongbin Wang, Xinhua Liu, and Lin Zhang, A Novel Compound Control Method for Hydraulically Driven Shearer Drum Lifting. Hindawi Publishing Corporation Journal of Control Science and Engineering Volume 2014, Article ID 691787, 12 pages

[5] G. Song, R.W.Longman, and R.Mukherjee, Integrated sliding mode adaptive–robust control, IEE Proc.-Control Theory Appl.,Vol.146, No.4 July 1999, pp. 341-347.

[6] Brian Armstrany-Helouvry,Control of Machine with Friction, Kluwer Academic Publishers, 1991

[7] Lennart Ljung, System Identification : Theory for the User, Prentice Hall, inc. Englewood Cliffs, New Jersey 07632

[8] K.J.Astrom, and B.Wittenmark, Computer Controlled Systems, Prentice Hall, 1984

[9] Mark W.Spong, M. Vidyasagar, Robot Dynamic and Control, Jonh Wiley & Sons, 1989

[10] Slotine J.E,and Li W., Applied Non linear Control, Prentice-Hall International, Inc 1991.

[11] Weiping Li, and Xu Cheng, Adaptive High-Precision Control of Positioning Tables – Theory and Experiment ,IEEE Transaction on Control System Technology, Vol. 2 No.3, September 1994

[12] Bacciotti A, and Rosier L, Lyapunov functions and stability in control theory, Springer-Verlag ,London,2001