SUBOPTIMAL PID CONTROL METHOD FOR AN ELECTRO-HYDRAULIC SYSTEM

PHƯƠNG PHÁP ĐIỀU KHIỂN PID CẬN TỐI ƯU CHO MỘT HỆ THỐNG ĐIỆN THỦY LỰC

Dang Xuan Ba[1], Kyoung Kwan Ahn[2]
[1]Faculty of Electrical and Electronics Engineering, HCMUTE
[2] Department of Mechanical Engineering, University of Ulsan

Received 26/10/2016, Peer reviewed 2/11/2016, Accepted for publication 15/12/2016

ABSTRACT

This research proposes an advanced proportional-integral-derivative (PID) controller for position-tracking control problem of a pump-controlled Electro-Hydraulic System (EHS). Although conventional PID controller is a practically robust and easy-to-use method, it is difficult to maintain good control performance of such system which contains both high nonlinearities and unknown terms. The key idea of the proposed approach is thus developed from a combination of the conventional PID signal, a robust nonlinear compensator which is properly derived from the system model, and a suboptimal regulator. Effectiveness and feasibility of the designed controller have been successfully evaluated through theoretical proof and experimental results of fast response (0.3s) and excellent steady-state error (0mm).

Keywords: *Electro-hydraulic system; PID control; Advanced PID control;Optimal control; Tracking control.*

TÓM TẮT

Trong nghiên cứu này, tác giả đề xuất một bộ điều khiển nâng cao được mở rộng từ cấu trúc bộ điều khiển vi-tich-phân-tỉ-lệ (PID) để giải quyết bài toán điều khiển bám vị trí áp dụng cho một hệ thống bơm điện - thủy lực. Mặc dù, bộ điều khiển PID truyền thống là một phương pháp dễ sử dụng và được áp dụng rộng rãi trong công nghiêp, nhưng nó khó đảm bảo được chất lượng điều khiển tốt cho những hệ thống có độ tuyến tính cao và chứa đựng nhiều yếu tố chưa biết. Vì vậy, ý tưởng thiết kế chính của bộ điều khiển được đề xuất ở đây là dựa trên sự kết hợp ưu điểm của bộ điều khiển PID truyền thống, với một bộ bù phi tuyến và một bộ chỉnh định tối ưu. Sự hiệu quả và tính khả thi của phương pháp thiết kế này đã được kiểm chứng thành công dựa vào chứng minh lý thuyết và các kết quả thực nghiệm với đáp ứng nhanh (0.3s) và sai số xác lập tốt (0mm).

Từ khóa: *Hệ thống điện thủy lực; Điều khiển PID; Điều khiển PID nâng cao; Điều khiển tối ưu; Điểu khiển bám.*

1. INTRODUCTION

Nowadays, Electro-Hydraulic actuators (EHA) have been very widely used in modern industries, especially heavy industry, thank to ability of high force generation. Some typical applications of the actuators have been commercialized as press machines [1], excavators [2] or track cranes [3], etc. There are two common kinds of electro-hydraulic systems: Valve-controlled EHS and pump-controlled EHS. The configuration of valve-control EHS usually consists of a high-pressure fluid supplier and controlled valve(s). The supply pressure is maintained at a fixed value while the system output is decided by operation of the valve(s). Although the system can provide a fast response, a lot of system energy may be lost at the control valve(s) due to throttle phenomenon. In order to cover this problem, the pump-controlled systems have been proposed and already commercialized in recent years. The operation of the new systems is controlled directly from the speed of controlled pumps. Hence, the lost energy is reduced significantly. Low bandwidth and complicated design are drawbacks of this kind of systems.

Many techniques have been developed for the position or force control of the EHA system. First of all, it must be mentioned about the proportional-integral-derivative (PID) and advanced PID approaches [4-6]. The PID gains are employed under many methods such as genetic algorithm, or incorporating with a fuzzy controller or analyzing the detailed modeling of control system, etc. The obtained performances are improved clearly. However, the closed-loop stability has not yet been proven. Another category of adaptive nonlinear controllers has been successively proposed to control the system with the full mathematical nonlinear model. The adaptive sliding mode [7] and back-stepping [8-11] controllers have been adopted to cover the system uncertain nonlinearities by adaptation law and robust nonlinear design. Though the remarkable results have been displayed, the adaptation only applied to last stage of the mathematical model. In practice, the uncertainties exit in all stages of the system. As a sequence, some comprehensive control approaches have been designed based on the integrated adaptive nonlinear techniques [12-13]. Both transient and steady-state control performances are further improved. Nonetheless, employing so many control gains may make difficulties in reproducing the controllers in different applications.

In this research, an advanced PID controller is studied on a typical model of the pump-controlled EHS. First of all, to realize control objective of the system, a conventional PID controller is utilized. According to the reviewed studies [4-6], the nonlinearities of such system are difficult to be compensated well by a linear method. A nonlinear control signal is thus derived to improve the control performance from the mathematical model. Selecting proper control gains to satisfy both the transient and steady-state responses is facilitated by designing an additional optimal control signal. Stability of the closed-loop system is then theoretically proven by Lyapunov constrains. Finally, effectiveness of the proposed method is verified on a real-time application via various experimental conditions.

The paper is organized as followings: Section 2 presents the mathematical nonlinear modeling of the studied system, Section 3 derives the proposed controller based on a combination of the PID controller, the nonlinear compensator and the suboptimal regulator, Section 4 briefly shows the experimental setup and results, and Section 5 contains conclusions of the study.

2. SYSTEM MODELING

Working principle of the studied system is presented in Fig. 1. The system motion is generated by a double-acting single-rod cylinder (DAJON TECH, D140H-SD50B-

N300), which is driven by a gear pump (GALTECH, SM-G-4-R-SAEA-13GGA-VT) through a control hydraulic circuit. The pump is actuated by an AC servo motor (HIGEN FMACN10-AB00). In order to exactly control speed of the motor, a motor driver (HIGEN FDA7010) is used with a stable controller already integrated inside.

Position dynamics of the system can be presented by applying Newton's second law

$$m\ddot{x} = P_L A_1 - F_l. \tag{1}$$

wherein, m is total mass including the piston mass and load; $P_L = P_1 - P_2(A_2/A_1)$ is load pressure in which P_1 and P_2 are pressures in chambers of cylinder. A_1, A_2 are bore and rod-side areas, respectively; F_l is lumped unknown force which consists of the external force *d*, viscous friction, static friction, Coulomb friction and hard-to-model terms [7, 14].

Pressure dynamics inside the chambers can be written as in Ref. [14-16]

$$\begin{cases} \dot{P}_1 = \dfrac{\beta_e}{V_{t1}}\left(Q_1 - A_1\dot{x} - Q_{L1}\right) \\ \dot{P}_2 = \dfrac{\beta_e}{V_{t2}}\left(Q_2 + A_2\dot{x} + Q_{L2}\right). \end{cases} \tag{2}$$

wherein, β_e is the effective bulk modulus of used hydraulic fluid, $V_{t1} = V_{10} + A_1\dot{x}$ and $V_{t2} = V_{20} - A_2\dot{x}$ are active volumes, V_{10} and V_{20} are original total volumes in the chambers, respectively. The original volume consists of volume of pipelines and initial cylinder chamber at the same side; Q_1, Q_2 denote supply flows to the appropriate chambers. Q_{L1}, Q_{L2} are the flumped leakage flows in the sides of the chamber 1 and chamber 2, respectively.

The concerned flows can be calculated as follows:

$$\begin{cases} Q_{L1} = C_L P_L + \delta_{L1} \\ Q_{L2} = Q_{L1} + \delta_{L2} \\ Q_1 = Q_{P1} + Q_{d1} = Dw \\ \quad + \dot{x}\dfrac{\alpha A_2}{V_{t2}}(A_1 - A_2)st(-\dot{x}) + \delta_{d1} \\ Q_2 = Q_{P2} + Q_{d2} = -Dw \\ \quad + \dot{x}\dfrac{\alpha A_1}{V_{t1}}(A_1 - A_2)st(\dot{x}) + \delta_{d2} \end{cases} \tag{3}$$

where $\delta_{d1}, \delta_{d2}, \delta_{L1}$, and δ_{L2} are modeling errors Q_{d1}, Q_{d2}, Q_{L1}, and Q_{L2}, respectively.

By defining state variables as: $x_1 = x, x_2 = \dot{x}, x_3 = A_1 P_L / m$, and combining Eqs. (1 - 3), the system dynamics can be rewritten in a state-space form as

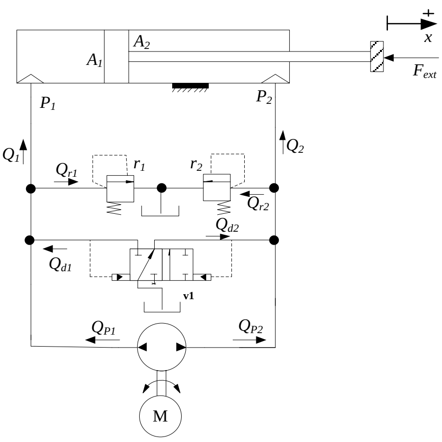

Fig. 1 *Configuration of the studied system*

$$\begin{cases} \dot{x}_2 = x_3 - \dfrac{F_l}{m} \\ \dot{x}_3 = \dfrac{\beta}{mV_{t1}V_{t2}}\Big(-\left(A_1^2 V_{t20} + A_2^2 V_{t1}\right)x_2 \\ \qquad - C_{le}P_L\left(A_1 V_{t20} + A_2 V_{t1}\right) \\ \qquad + \left(A_1 V_{t20} + A_2 V_{t1}\right)Dw \\ \qquad + x_2 \alpha A_2 A_1 (A_1 - A_2) sign(-x_2) \\ \qquad - x_2 \alpha A_2 A_1 \left(A_1 - A_2\right) sign\left(x_2\right) + \delta_3\Big) \end{cases} \tag{4}$$

Where in, δ_3 is lumped modeling error.

It can be seen that the studied system is a highly nonlinear system with presence of unknown terms of the external force and un-modeled elements. In next section, a robust controller will be designed to improve the control performance based on the derived system.

3. SUBOPTIMAL ROBUST CONTROL DESIGN

In this section, a nonlinear controller is developed to control the system output x track to the desired input x_d as closely as possible and cover all nonlinearities of the system.

Assumption:

1. The desired input x_d is bounded in the working range of the system.
2. The state variables x_1, x_2 and x_3 are measurable.
3. All modeling errors are bounded.

A conventional PID controller can be expressed as follows:

$$w_{PID} = K_P e + K_I \int_0^t e(\tau)d\tau + K_D \dot{e} \tag{5}$$

wherein, w_{PID} is control input, $e = x_d - x$ is control objective, and $K_P, K_I,$ and K_D are positive control parameters.

Proper selection of the control parameter can yield a good control performance for a specific set point. However, the performance is difficult to maintain for position-tracking control. A compensator is thus proposed as the following procedure. First, define a new variable as

$$x_{3n} = \dot{x}_2 + k_1 x_1 + k_2 x_2 \tag{6}$$

The system dynamics (4) can be re-written as follows

$$\begin{cases} \dot{x}_1 = x_2 \\ \dot{x}_2 = x_{3n} - k_1 x_1 - k_2 x_2 \\ \dot{x}_{3n} = V_\Pi \left(-C_L \left(x_{3n} - k_1 x_1 - k_2 x_2\right) V_\Sigma \right) \\ -\left(A_1^2 V_{t2} + A_2^2 V_{t1}\right) V_\Pi x_2 + V_\Pi V_\Sigma D w \\ -\alpha A_2 A_1 (A_1 - A_2) sign(x_2) V_\Pi x_2 \\ -k_1 k_2 x_1 + (k_1 - k_2^2) x_2 + k_2 x_{3n} - \delta_n. \end{cases} \tag{7}$$

where $\delta_n \triangleq -V_\Pi(\delta_3 - \dot{F}_l)$ is a composited modeling error, and

$$\begin{cases} V_\Sigma \triangleq A_1 V_{t20} + A_2 V_{t1} \\ V_\Pi \triangleq \dfrac{\beta}{m V_{t1} V_{t2}} \\ V_S \triangleq V_\Pi V_\Sigma. \end{cases}$$

The use of x_{3n} is to replace the pressure signal x_3 which commonly contains heavy noise due to employing the analog sensor.

Here, the nonlinear control input can be easily obtained by combination of the new system and the desired input x_d:

$$\begin{aligned} w_c = \frac{1}{V_\Sigma D}\Bigg(& \frac{1}{V_\Pi}\left(\dot{x}_{3d} + k_1 k_2 x_{1d} - (k_1 - k_2^2) x_{2d}\right) \\ & + C_L \left(x_{3d} - k_1 x_1 - k_2 x_2\right) V \\ & - k_2 x_{3d} + \left(A_1^2 V_{t2} + A_2^2 V_{t1}\right) x_2 \\ & + \left(\alpha A_2 A_1 (A_1 - A_2)\left(sign(x_2) - sign(-x_2)\right)\right) x_2 \Bigg) \end{aligned} \tag{8}$$

wherein $x_{2d} = \dot{x}_{1d}$ and $x_{3d} = k_1 x_{1d} + k_2 x_{2d} + \dot{x}_{2d}$.

To verify theoretical effectiveness of the designed method, the following theorem is studied:

Theorem 1:

Given a nonlinear system as Eq. (4), if employing a PID controller (5) incorporated with a nonlinear compensator (8) and properly selecting the control parameters k_1, k_2, then the control objective e will be asymptotical stable at infinite time.

Proof:

Consider a synthesized error vector as

$$E \triangleq [e_0; e_1; e_2; e_3]^T = [\int_0^t e(\tau)d\tau; e; \dot{e}; (\ddot{e} - k_1 e - k_2 \dot{e})]^T \quad (9)$$

The error dynamics is presented from Eqs. (7) and (8)

$$\dot{E} = \begin{bmatrix} 0 & 1 & 0 & 0 \\ 0 & 0 & 1 & 0 \\ 0 & -k_1 & -k_2 & 1 \\ 0 & -k_1 k_2 & (k_1 - k_2^2) & -C_L V_S + k_2 \end{bmatrix} E + \left(V_S D\left(w_c - w\right) + \delta_n\right)\begin{bmatrix} 0 \\ 0 \\ 0 \\ 1 \end{bmatrix} \quad (10)$$

After applying both the PID and nonlinear control inputs to the system, the closed-loop dynamics becomes

$$\dot{E} = \begin{bmatrix} 0 & 1 & 0 & 0 \\ 0 & 0 & 1 & 0 \\ 0 & -k_1 & -k_2 & 1 \\ K_{e1} & K_{e2} & K_{e3} & K_{e4} \end{bmatrix} E + \delta_n \begin{bmatrix} 0 \\ 0 \\ 0 \\ 1 \end{bmatrix} = AE + B\delta_n \quad (11)$$

$$K_{e1} = -V_S D K_I; K_{e2} = -V_S D K_P - k_1 k_2;$$
$$K_{e3} = -V_S D K_D + (k_1 - k_2^2); K_{e4} = -C_L V_S + k_2$$

Now, validate a Lyapunov function as follows

$$V = E^T P E \quad (12)$$

If the control parameters k_1, k_2 are chosen such that the matrix A is positive definite, then according to Lyapunov theory, there exists a pair of positive matrices (P, S) for:

$$\dot{V} = -E^T S E + 2B^T P E \delta_n \le -\frac{\lambda_{\min}(S)}{\lambda_{\min}(P)} V + 2B^T P E \delta_n \quad (13)$$

Employing Barbalat's Lemma leads to the proof of Theorem 1.

Remark: With a given set of PID parameters, there always exists a proper nonlinear signal to lead the system converge to a desired bound. Value of k_2 should be less than that of $\min(C_L V_S)$. However, selecting the proper control parameters to maintain the transient performance is not a trivial task. As a result, to deal with this work, an additional control signal is proposed from the optimal control technique.

Now consider an index functions as follows

$$J = \frac{1}{2}\int_{t_0}^{t_f} \left(E^T Q E + r w_o^2\right) d\tau \quad (14)$$

where w_o is the additional control signal, Q and r are positive definite matrix and constant, respectively.

To force the control errors to be as small as possible in the shortest time with minimum control effort, the control signal is designed as follows

$$w_c = V_\Sigma D r^{-1} T_s B^T \left(I + T_s A\right)^{-1} Q E \quad (15)$$

where T_s is the sampling time and I is the proper-size identify matrix.

4. EXPERIMENTAL VALIDATION

In this section, experimental results of the designed controller for a real-time system are presented and discussed. The experimental system consisted of the studied model, which has been introduced in Section 2, and a control-data acquisition (CDAQ) system. The CDAQ system includes an Advantech Industrial Computer (Core i3-2100 3.1GHz), a data acquisition (DAQ) card (PCI-6014), a encoder reader

(PCI-QUAD 04) and a sensor system. A linear encoder (WTB5-500MM) was added to the system to measure position. The designed controller was run in the computer within Real-time Window Target Toolbox of Matlab under a sampling time of 2ms. System specifications obtained from the manufactures are summarized on Table 1, which the system apparatus is setup as Fig. 2.

The system parameters were set as follows:

The total mass $m = 5kg$;

The effective bulk modulus $\beta_e = 700MPa$;

Coefficients of the leakages $C_L = 10^{-15}(m^3 s / Pa)$;

Original total control volumes $V_{10} = 3.2\times10^{-4}m^3$ and $V_{20} = 2.15\times10^{-4}m^3$;

Control input range: $u \in [-10;10](V)$;

Two kinds of the desired input signals (sinusoidal and smooth multistep) were used to verify the proposed controller. To clearly show advantages of the proposed idea, the method was employed in two cases under same working conditions: a conventional PID controller and the PID controller combined with the nonlinear compensator and regulator. The PID gains were determined by trial-and-error method as: $K_P = 65; K_I = 1.2; K_D = 5.$ Meanwhile, the control parameters of the compensator and regulator were chosen to be $k_1 = 0.5; k_2 = 0.0001$ and $Q = eye([0.1 \quad 12 \quad 1.5 \quad 0.002])$ and $r = 10^{-3}$, respectively. Moreover, a Fuzzy PID controller was also applied to the same system to confirm the effectiveness of the designed method more certainly. Structure of the Fuzzy PID method was developed as shown in [5].

Table 1.*Specification of the studied system*

Device	Specification
Hydraulic cylinder	Type: DAJON D140H-SD50B-N300 Stroke: 300(mm) Tube diameter: 50(mm) Rod diameter: 30(mm) Max. pressure: 140(bar)
Hydraulic pump	Type: GALTECH 2SM-G-4-R-SAEA-13GGA-VT Displacement: 4 (cc/rev) Max. speed: 4000(rpm)
AC Servo motor	Type: HIGEN FMACN10-AB00 Power: 1(kW) Max. speed: 3000(rpm)

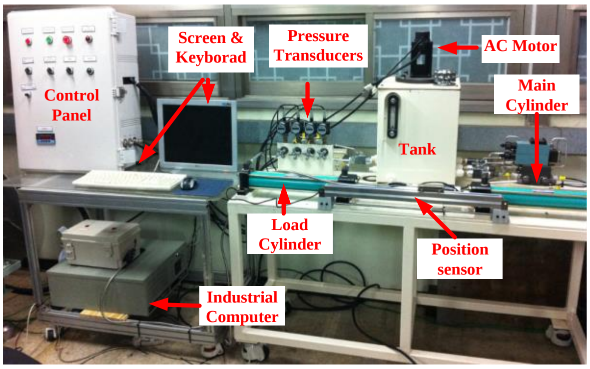

Fig. 2 *Photograph of the experimental apparatus*

The first case of simulation process was carried out with the sinusoidal desired inputof amplitude 100 mm and a frequency of 0.05 Hz. Figures 3 and 4 show the control results of the proposed controller (APID), the conventional PID controller (CPID) and the Fuzzy PID controller (FPID). As seen in Fig. 3, trajectory of the proposed system was almost same as those of the conventional and Fuzzy PID systems. However, thanks to the use of the improvement terms (Eqs. (8) and (15)), output of the compensated system was more closed to the reference input than that of the conventional PID controller. Efficiency of the improvement can be clearly seen from the control errors in Fig. 4. The obtained error of the advanced system was only ±0.2 mm (~ 0.2%) while those of the PID system tuned by the trial-and-error method without compensator and the intelligent PID system were in the ranges of [-1.2; 1.7] (mm) (~ 1.2%) and [-0.4; 0.6](mm) (~ 0.4%), respectively. Here, the effectiveness of the proposed controller is confirmed.

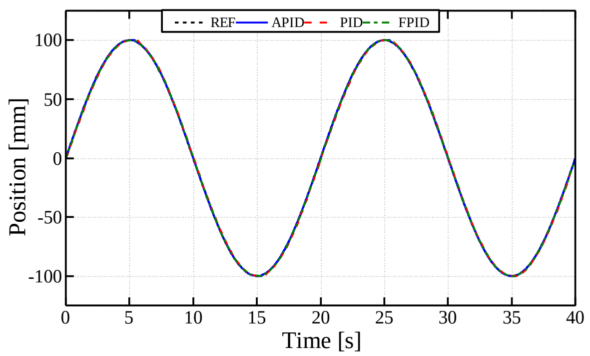

Fig. 3*Comparative responses with respect to case 1*

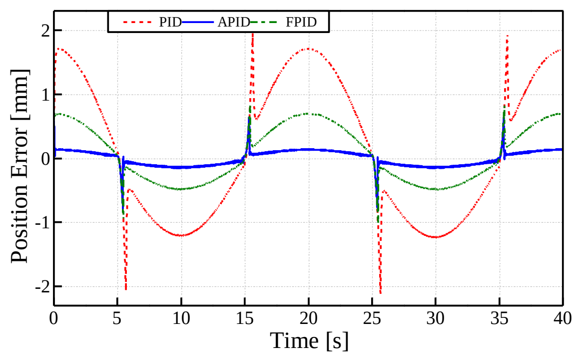

Fig. 4 *The obtained control errors with respect to case 1*

For further investigating the controller, the second experiment was performed for the smooth multi-step trajectory. After applying the same controllers to the system, comparative responses are depicted in Fig. 5 and 6. Although steady-state errors of three controllers are the same, transient behaviors are clearly different. For example, as presented in Fig.6, to control the system output from the -60mm set-point to the 0mm set-point, the maximum error and the settling time of the conventional and Fuzzy PID controllers are respectively about (4mm and 3s) and (2mm and 2.5s) while those of the advanced controller are only 2mm and 0.3s. Hence, for the faster response, the smaller overshoot and settling time are remarkable advantages of the proposed method.

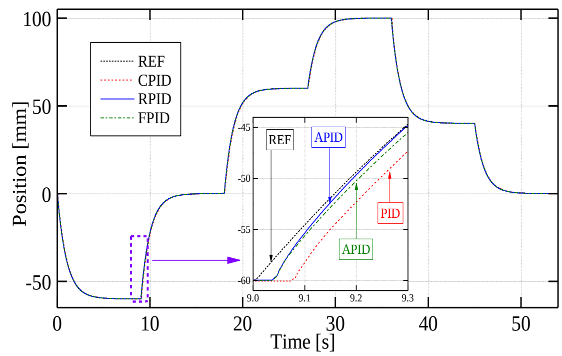

Fig. 5 *Comparative responses with respect to case 2*

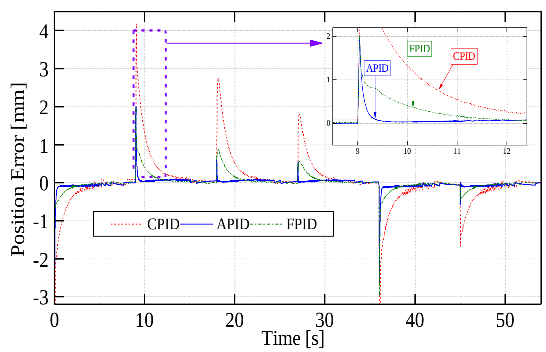

Fig. 6 *The obtained control errors with respect to case 2*

5. CONCLUSION

In this paper, an advanced PID controller has been introduced for position control of a pump-controlled hydraulic system. Both mathematical model of the studied system and the proposed algorithm are fully derived. Then, the designed controller is successfully verified via comparing with the conventional and Fuzzy PID controllers in real-time experiments. The effectiveness and feasibility of the advanced method are confirmed by both theoretical proof and experimental results. However, the control error can be further improved by employing adaptive laws to deal with the modeling error. Hence, developing an adaptive robust PID controller is planned as a future work of this study.

REFERENCES

[1] Press Machine package, *Kawasaki Heavy Industries, Ltd*. (2014). [Online]. Available: http://www.khi.co.jp/kpm/pdf/data/Industrial_J_130403a.pdf

[2] Volvo EC380E Excavator, *Volvo Contruction Equipment*, Volvo Group. (2014). [Online]. Available:http://www.volvoce.com/SiteCollectionDocuments/VCE/Documents%20North%20America/crawler%20excavators/Volvo_EC380EProductBrochure%203-14-14.pdf

[3] Truck Cranes: QY100 Package, *Sany Group Co., Ltd*. (2014). [Online]. Available: http://www.sanygroup.com/products/en-us/hoisting/QY100.htm

[4] A. Ayman and Aly, PID parameters optimization using genetic algorithm technique for electrohydraulic servo control system, *Int. Cont. and Automation*, Vol. 2, No. 2, pp. 69-76, 2011.

[5] D. Q. Truong and K. K. Ahn, Force control for press machines using an online smart tuning fuzzy PID based on a robust extended Kalman filter, *Expert Systems with Applications*, Vol. 38, No. 5, pp. 5879-5894, 2011.

[6] Z. Jiangbo, W. Junzheng and W. Shoukun, Fractional order control to the electro-hydraulic system in insulator fatigue test device, *Mechatronics*, Vol. 23, No. 7, pp. 828-839, 2013.

[7] C. Guan and S. Pan, Adaptive sliding mode control of electro-hydraulic system with nonlinear unknown parameters, *Cont. Eng. Practice*, Vol. 16, No. 11, pp. 1275-1284, 2008.

[8] K.K. Ahn, D.N.C. Nam and M. Jin, Adaptive backstepping control of an electrohydraulic actuator, *IEEE/ASME Trans. On Mechatronics*, Vol. 19, No. 3, pp. 987-995, 2014.

[9] V. Milic, Z. Situm and M. Essert, Robust H-infinity position control synthesis of an electro-hydraulic servo system, *ISA Trans*., Vol. 49, No. 4, pp. 535-542, 2010.

[10] A. Mohanty and B. Yao, Intergrated direct/indirect adaptive robust control of hydraulic manipulators with valve deadband, *IEEE/ASME Trans. on Mechatronics*, Vol. 16, No. 4, pp. 707-715, 2011.

[11] A. Mohanty and B. Yao, Indirect adaptive robust control of hydraulic manipulators with accurate parameter estimates, *IEEE Trans. on Cont. Sys. Tech*., Vol. 19, No. 3, pp. 567-575, 2011.

[12] K. Guo, J. Wei, J. Fang, R. Feng, and X. Wang, Position tracking control of electro-hydraulic single-rod actuator based on an extended disturbance observer, *Mechatronics*, Vol. 24, pp. 47-56, 2015.

[13] D. X. Ba, K. K. Ahn, D. Q. Truong, and H. G. Park, Integrated model-based backstepping control for an electro-hydraulic system, *Int. J. Pres. Eng. Manufacturing*, Vol. 17, No. 6, pp. 1-13, 2016.

[14] M. Jelali and A. Kroll, Hydraulic Servo-systems modeling, identification and control, *Springer*, London, 2004.

[15] H. E. Merritt, Hydraulic Control Systems, *Wiley*, New York, 1967.

[16] N. D. Manring, Hydraulic Control Systems, *Wiley*, New York, 2005.

Corresponding author:

Dang Xuan Ba

Ho Chi Minh City University of Technology and Education

Email: dang.xuanba@gmail.com