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The controlling the hydraulic electro drive system on the basis of the adaptive sliding mode control method

Tran Duc Chuyen[1], Nguyen Thanh Tien[2], Vu Duy Hung[1], Pham Ngoc Sam[1], Tran Hai Duong[1]
[1]Faculty of Electrical Engineering, University of Economics, Technology for Industries, Hanoi, Vietnam
[2]Military Institute of Mechanical Engineering, Hanoi, Vietnam

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ABSTRACT

In this paper, presents the construction of an automatic control system for mobile lifting machines applied in industry and traffic. The control algorithm is built on the basis of calculations from the actual system taking into account the estimation of the nonlinear factors of frictional moment uncertainty, on the basis of the adaptive sliding mode control method. Using programming and simulation tools on the basis of MATLAB/Simulink to demonstrate the research results. The results of the simulation study are the basis for the design and calculation of the hydraulic electro drive system, which shows the practical effectiveness of the study of electromechanical traction electric transmission systems such as mobile lifting machines. The motion uses hydraulic valve systems to transmit power control (lifting and lowering as required).

Corresponding Author:

Tran Duc Chuyen
Faculty of Electrical Engineering, Universityof Economics, Technology for Industries
Hanoi, Vietnam
Email: tdchuyen@uneti.edu.vn

1. INTRODUCTION

The hydraulic transmission system is responsible for transmitting force and torque, creating linear or rotational movements according to a given trajectory or according to the specific requirements of each mobile lifting machine [1]–[5]. These tasks can be accomplished by other forms of motion, such as mechanical or electric drives. However, the hydraulic electro drive system has more advantages, so it is widely used, especially on mobile machines and equipment. The general structure of the electro-hydraulic drive system is shown in Figure 1, it consists of three main parts as follows. The first is the power supply: here the mechanical power from the electric motor, diesel engine or turbine is converted into hydraulic power through the pump (flow Q and pressure p) the drive motor and the pump drive are the two main elements of the power supply; in addition, there are other auxiliary components such as accumulator, oil tank, cooling system, and oil heating. The second is the transmission and control part: responsible for transmitting and controlling hydraulic energy through hard steel oil pipes or rubber hoses. The third is the energy consumption part: hydraulic power through the cylinder and engine will be converted into mechanical energy to drive the working parts of the machinery [6]–[8]. Some lifting machinery of mobile lifting machines use hydraulic valve systems to transmit power control (lifting and lowering as required), such structures as: forklift goods at warehouses, excavators, agricultural machines, and forestry machines, have a structural diagram as shown in Figure 2 [3], [6].

The from the past, we have worked on traditional controllers such as portional integral-derivative (PID), fuzzy proportional derivative (PD), PD+I, and proportional integral (PI), many other classical controllers that have not yet met the control process for the system this system [4]–[10]. The biggest problem with these methods is not yet available for this electro-hydraulic drive system. Therefore, the problem is that we do not use the manual adjustment method to improve the quality of the system, but need to use the study of the control algorithm to suit the system to bring about the best results high quality. Because when doing so, the system will cause many errors, the system operates incorrectly, and the quality of the controller is not high. The working system is not optimal, not good. Many times the optimal value is not achieved at all [11]–[15]. The biggest benefit of the adaptive sliding control algorithm is that a suitable controller is that the model self-tunes to achieve the optimal values, and gives the correct operation of the hydroelectric powertrain force of the lifting machine in the lifting machine [4], [6], [8], [10], [16]–[22]. In some of the previous studies as shown in the document [3], [13], [14], with only the use of switch-type relay control systems, and the use of a simple algorithm, the quality criteria were not met quantity of the system [23]–[26]. In this paper, we present an adaptive sliding control algorithm in the control cases of hydraulic transmission systems of transport lifts taking into account the nonlinear uncertainties such as the frictional moment of the engine lifting structure. The adaptive sliding control law is implemented to solve the problem to improve the control quality for this particular hydraulic drive system.

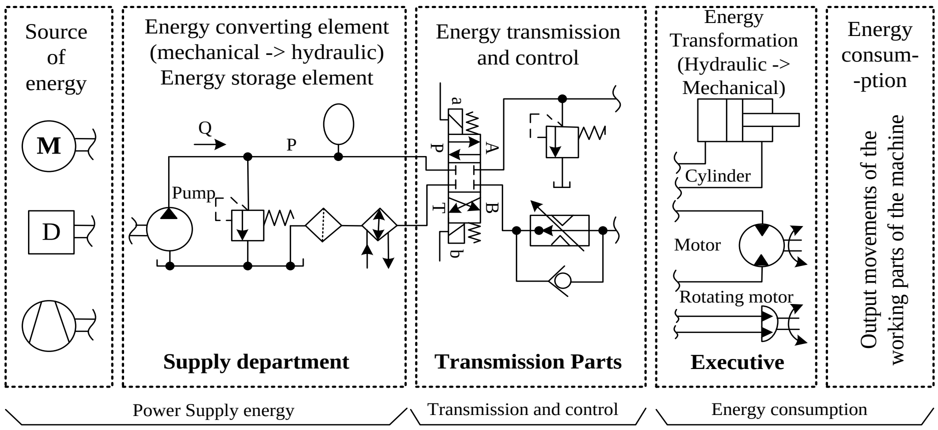


Figure 1. Basic structure of an electro-hydraulic drive system

2. CONTROL KINEMATICS MODEL FOR ELECTRO HYDRAULIC DRIVE SYSTEM

To study the control kinematics model for the closed-circuit hydraulic transmission system for the lifting machine using the drive motor. We consider a hydraulic control system of a four-wheel mobile machine with a load lifting mechanism as shown in Figure 2.

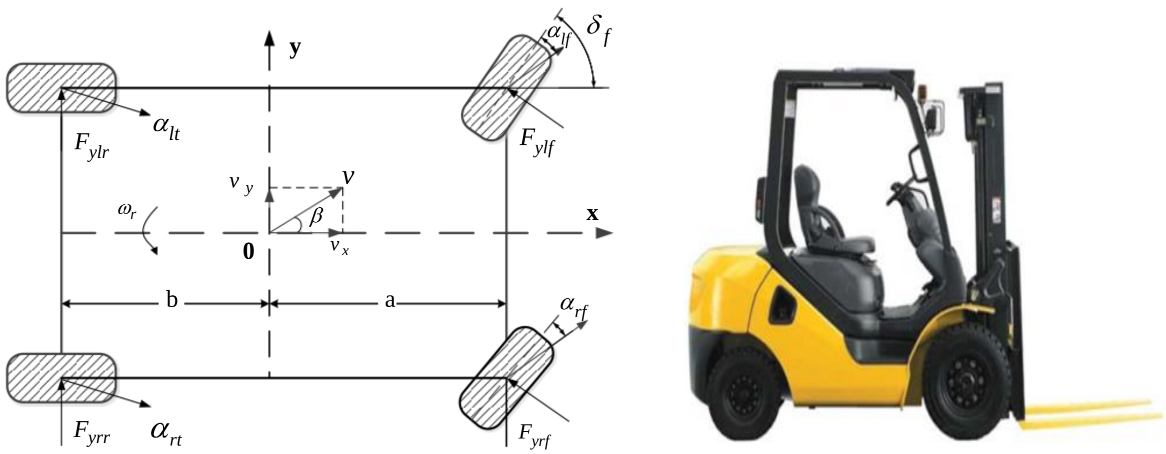


Figure 2. The kinetic model of forklift using electro-hydraulic drive system

In which the system structure is shown on the coordinate system oy, ox, v_y, v_x , M is the total mass of the vehicle, M_s is the vehicle's center of gravity, the distance from the center of gravity to the wheels is a, b, left front wheel F_{ylf}, right front wheel F_{yrf}, left rear wheel F_{ylr}, right rear wheel F_{yrr}, in addition, there are corresponding slip angles of tires α_{lf}, α_{rf}, α_{lt}, α_{rt}, angle β is the sliding angle of the vehicle [3], [7], [12], [13].

To establish the dynamical model of the dual robot system, the system is considered as two separated robots with external forces at tips of the robots. The external forces are the interactive forces between the robot arms and the object as shown in Figure 2. The dynamical equations for the dual robot system are described as (1):

$$\begin{cases} I_x\dot{p} - I_{xz}\dot{\omega}_r = M_s g h_l \psi - K_\psi \psi - C_\psi p + M_s h_l v_x(\beta + \psi_r) \\ M v_x(\dot{\beta} + \dot{\omega}_r) - M_s h_l \dot{p} = F_{ylf} + F_{yrf} + F_{ylr} + F_{yrr} \\ I_z\dot{\omega}_r - I_{xz}\dot{p} = -b(F_{ylf} + F_{yrf}) + a(F_{ylr} + F_{yrr}) \\ F_y = k\alpha \end{cases} \quad (1)$$

Then we can calculate the approximate values as: $\beta \approx tan\beta = \frac{v_y}{v_x}$,$\alpha_f = \beta + \frac{a}{v_x}\omega_r - \delta_f - R_f\psi$, $\alpha_r = \beta - \frac{b}{v_x}\omega_r - R_f\psi$, in there δ_f is the component of the forward steering angle of the vehicle, and R_f; R_r are the steering coefficients of the front and rear wheels respectively:

$$\begin{cases} \dot{x}(t) = Ax(t) + Bu(t) \\ y(t) = Cx(t) \end{cases} \quad (2)$$

in there,

$$A = M_1^{-1}M_2 B = M_1^{-1}M_3 C = \begin{bmatrix} 1 & 0 & 0 & 0 \\ 0 & 1 & 0 & 0 \\ 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 1 \end{bmatrix}; M_1 = \begin{bmatrix} 0 & Mv_x & 0 & -M_s h_l \\ I_z & 0 & 0 & -I_{xz} \\ -I_z & -M_s h_l v_x & 0 & I_x \\ 0 & 0 & 1 & 0 \end{bmatrix}; M_3 = \begin{bmatrix} k_1 \\ k_1\alpha \\ 0 \\ 0 \end{bmatrix}$$

$$M_2 = \begin{bmatrix} \frac{-k_1 a + k_2 b}{v_x} - Mv_x & -k_1 - k_2 & k_1 R_f + k_2 R_2 & 0 \\ -\frac{k_1 a^2 + k_2 b^2}{v_x} & -ak_1 + bk_2 & k_1 a R_f - k_2 b R_r & 0 \\ M_s h_l v_x & 0 & -K\psi + M_s g h_l & -C_\psi \\ 0 & 0 & 0 & 0 \end{bmatrix}$$

The process of moving on the surface when loaded, unloaded and on the road of transport forklifts for industrial and traffic applications is always a matter of concern. For the kinematics of this system, the power-to-mass ratio (kW/kg) is always met. So with the same transmission power, the structure of the hydraulic motor is much more compact than the other types other engine.

3. RESEARCH AND APPLICATION OF ADAPTIVE SLIDNG ALGORITHM FOR HYDRAULIC ELECTRO DRIVE SYSTEM

From the kinematic model of the electro-hydraulic drive system of the forklift truck above, when taking into account the nonlinear uncertainties at the output of the powertrain along with the variation of the load carried by the truck, carrier lifting machine. From there, we have the transformation equation of the system rewritten as (3):

$$\begin{cases} \dot{x}(t) = Ax(t) + Bu(t) \\ y(t) = Cx(t) + Dd(t) + F_f \end{cases} \quad (3)$$

Suppose when a sensor error occurs, the model can be described as (4):

$$\begin{cases} \dot{x}(t) = Ax(t) + Bu(t) \\ y_1(t) = C_1 x(t) + D_1 d(t) + F_1 f_1 \\ \vdots \\ y_m(t) = C_m x(t) + D_m d(t) + F_m f_m \end{cases} \quad (4)$$

Where the m component is the number of sensors that have failed with the system. Then to ensure the process of correcting the errors caused by the sensor, we perform the transformations to bring (4) to the standard form as (5) [3], [26]:

$$\begin{cases}\dot{x}(t) = Ax(t) + Bu(t)\\ y_i(t) = C_i x(t) + D_i d(t) + F_i f_i\end{cases} \tag{5}$$

Assuming the state variable z is defined, which is the first-order low-pass filter output of y(t) according to document [3], then this error is determined as (6):

$$\dot{z} = -A_{si} z_i + A_{si} y_i \tag{6}$$

Then we have:

$$\begin{cases}\begin{bmatrix}\dot{x}\\ \dot{z}_i\end{bmatrix} = \begin{bmatrix}A & 0\\ A_{si}C_i & -A_{si}\end{bmatrix}\begin{bmatrix}x\\ z_i\end{bmatrix} + \begin{bmatrix}B\\ A_{si}G_i\end{bmatrix}u + \begin{bmatrix}0\\ A_{si}D_i\end{bmatrix}\theta(t) + \begin{bmatrix}0\\ A_{si}F_i\end{bmatrix}f_i\\ z_i = \begin{bmatrix}0 & I_1\end{bmatrix}\begin{bmatrix}x\\ z_i\end{bmatrix}\end{cases} \tag{7}$$

A new state variable and the corresponding matrix can be represented as (8):

$$\begin{aligned}\bar{x}_i = [x \quad z_i]^T, \bar{y}_i = z_i, \bar{A}_i = \begin{bmatrix}A & 0\\ A_{si}C_i & -A_{si}\end{bmatrix}, \ \bar{B}_i = \begin{bmatrix}B\\ A_{si}G_i\end{bmatrix},\\ \bar{C}_i = [0 \quad I], \bar{H}_i = \begin{bmatrix}0\\ A_{si}D_i\end{bmatrix}, \bar{F}_i \begin{bmatrix}0\\ A_{si}F_i\end{bmatrix}\end{aligned} \tag{8}$$

When taking into account the nonlinear factor of uncertainty in a certain state, then we express as (9):

$$\begin{cases}\dot{\bar{x}}_i(t) = \bar{A}_i\bar{x}_i + \bar{B}_i u(t) + \bar{H}_i d(t) + \bar{F}_i f_i\\ \bar{y}_i(t) = \bar{C}_i\bar{x}_i(t)\end{cases} \tag{9}$$

Conduct research with electro-hydraulic drive control system with structure as (9). That is, the system's nonlinear nonlinear parameters lie on the left side of the complex plane, then for all numbers in the calculated coordinate system, the value R(s)≥0, as (10):

$$rank\begin{bmatrix}sI - \bar{A}_i & 0 & 0\\ A_{si}\bar{C}_i & sI + A_{si} & -A_{si}\\ 0 & I_1 & 0\end{bmatrix} = rank[sI - \bar{A}_i] + 2 \tag{10}$$

Then on the basis of sliding control we have:

$$\begin{cases}\dot{\hat{\bar{x}}}_i(t) = \bar{A}_i\hat{\bar{x}}_i + \bar{B}_i u(t) + L_i(\bar{y}_i - \hat{\bar{y}}_i) + \bar{F}_i v_i\\ \hat{\bar{y}}_i(t) = \bar{C}_i\hat{\bar{x}}_i(t)\end{cases} \tag{11}$$

where the value $\hat{\bar{x}}_i(t)$ is the state observation vector component of $\bar{x}_i(t)$, L_i is the gain matrix of the designed estimate in the sliding mode, $e_{yi} = y_i - \hat{\bar{y}}_i$ is the system output estimation error, v_i is the input vector sliding mode control is used to evaluate the error, then v_i is written as:

$$v_i = \begin{cases}\rho_i(t)\frac{D_i\bar{e}_{yi}}{||D_i e_{yi}||} & e_{yi} \neq 0\\ 0 & e_{yi} \neq 0\end{cases} \tag{12}$$

We have:

$$\begin{aligned}\frac{d\rho_i(t)}{dt} = \eta_i\left\|D_i\bar{e}_{yi}\right\| sgn(\left\|D_i\bar{e}_{yi}\right\| - \lambda_i), i = 1, \dots m\ \rho_i(0) > 0, \eta_i\\ < 2, 0 < \lambda_i < 1, i = 1, \dots\dots m\end{aligned} \tag{13}$$

The state estimation error value is determined as (14):

$$\bar{e}_i = \bar{x}_i - \hat{\bar{x}}_i \tag{14}$$

From expression (10) and expression (9), we have:

$$\dot{\bar{e}}_i = (\bar{A}_i - L_i\bar{C}_i)\bar{e}_i + \bar{H}_i d(t) + \bar{F}_i(f_i - v_i) \tag{15}$$

Then the Lyapunov function is defined and calculated as (16):

$$V(\bar{e}_i) = \bar{e}^T{}_i P\bar{e}_i + \frac{1}{2}(\rho_i - \rho_i^*)^2 \tag{16}$$

Now we take the derivative of (16) we get the following expression (17):

$$\begin{aligned}\dot{V}(\bar{e}_i) &= \bar{e}_i^T[(\bar{A}_i - L_i\bar{C}_i)^T P_i(\bar{A}_i - L_i\bar{C}_i)]\bar{e}_i + 2\bar{e}_i^T P_i\bar{F}(f_i - v_i) + \dot{\rho}_i(\rho_i - \rho_i^*) + 2\bar{e}_i^T P_i\bar{H}_i d(t)\\ &= -||e_i||[\lambda_{min}(\varrho_i)||e_i|| - 2||P_i||||\bar{H}_i||\omega] + 2\bar{e}_i^T P_i\bar{F}(f_i - v_i)\\ &\quad + n_i(\rho_i - \rho_i^*)||D_i\bar{e}_{yi}||sgn\big(||D_i\bar{e}_{yi}|| - \lambda_i\big)\\ &\leq -||\bar{e}_i||[\lambda_{min}(\varrho_i)||\bar{e}_i|| - 2||P_i||||\bar{H}_i||\omega] + (2\gamma_i - 2\rho_i) + n_i(\rho_i - \rho_i^*)||D_i\bar{e}_{yi}||\\ &= -||\bar{e}_i||[\lambda_{min}(\varrho_i)||\bar{e}_i|| - 2||P_i||||\bar{H}_i||\omega] + (2\gamma_i - 2\rho_i + 2\rho_i^* + n_i(\rho_i - \rho_i^*))||D_i\bar{e}_{yi}||\\ &= -||\bar{e}_i||[\lambda_{min}(\varrho_i)||\bar{e}_i|| - 2||P_i||||\bar{H}_i||\omega] + 2\big((\gamma_i - \rho_i^*)\big) - |\rho_i - \rho_i^*|(n_i - 2)||D_i\bar{e}_{yi}||\end{aligned} \tag{17}$$

Combined with the above conditions and calculation expressions, then $\dot{V}(\bar{e}_i) < 0$ and component $\|\bar{e}_i\| < 0$ converge in the following range of values:

$$U(\bar{e}_i) = \left\{\|\bar{e}_i\| \leq 2\|P_i\|\|\bar{H}_i\|\omega/\lambda_{\min}(Q_i)\right\} \tag{18}$$

where $\phi > 0$, $\gamma > 0$, and q, p are positive odd numbers.

4. SIMULATION AND EXPERIMENTAL RESULT

Based on the control algorithm for the system studied above. Going to study hydraulic systems with closed control structures using proportional valves with position feedback as shown in Figure 3. The diagram of the control system of the hydraulic transmission system of the closed-circuit transport lifting machine with the use of a permanent magnet excitation synchronous electric motor.

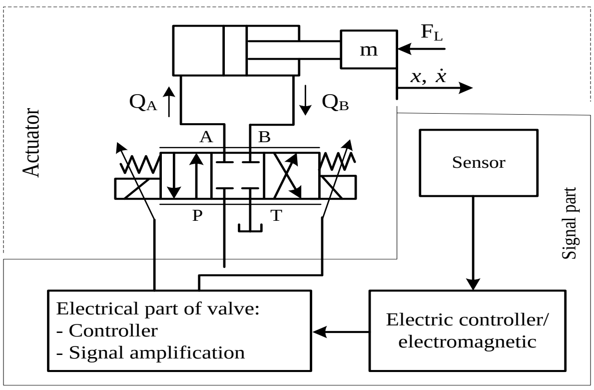


Figure 3. Block diagram of hydraulic system control using proportional valve with position feedback

As shown in Figure 4, M is a permanent magnet synchronous motor (PMSM) with P=0.75 kW, rated speed is 3,000 rpm, elements 4, 5, 6 are pressure regulating valves for the system and system. The pump system is rigidly connected to the electric motor, throttle valve, in addition, there is a pressure gauge, and protective switchgear for the system. In Figure 5 is the simulation result of the friction torque function of the piston and the cylinder of the hydraulic electro drive system according to the speed.

When the system is working at a given angle, the input is a step function with a value of 0.1 rad, when the external load torque is constant M=5 Nm. The simulation results as shown in Figures 6(a) and (b). The angle applied to the baby is 0.05 rad, the system works in the motion mode of the control system, we consider the influence of the frictional moment on the motor shaft, and the frictional moment on the load side; Simulation results as shown in Figures 7(a) and (b).

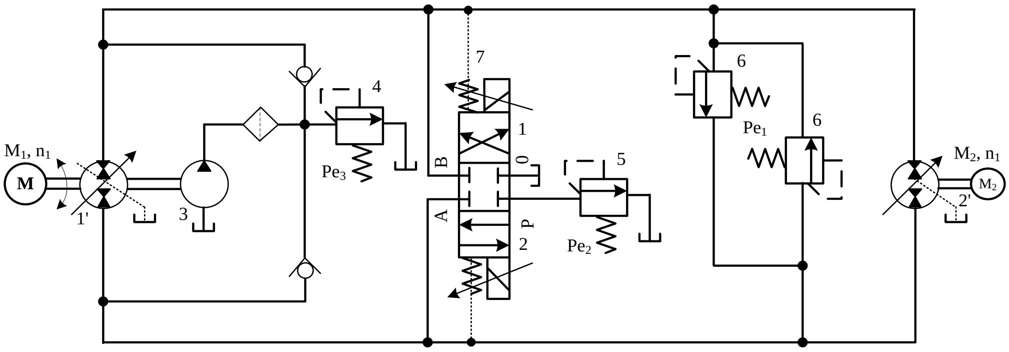


Figure 4. Closed-circuit drive system with electric motor drive PMSM

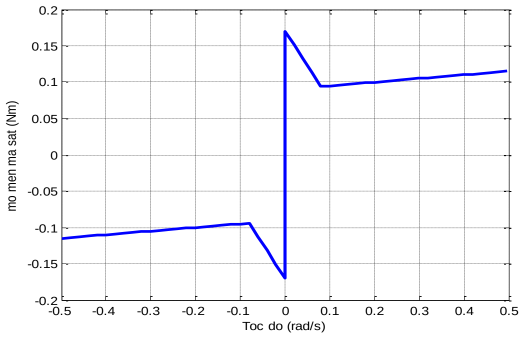


Figure 5. Friction torque characteristic of speed hydraulic control system

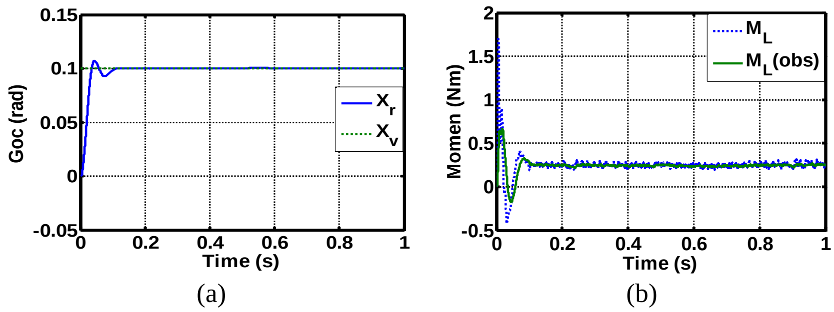


Figure 6. Controller input/output response (a) angular I/O response and (b) load torque estimator M_L; $\widehat{M}_L$

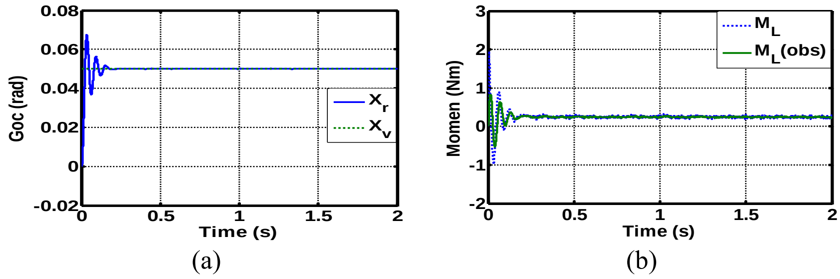
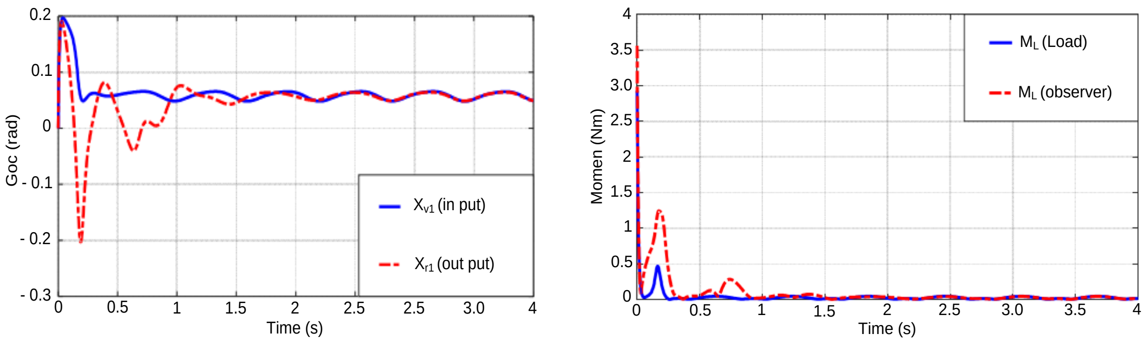


Figure 7. Position controller I/O response (a) angular I/O response and (b) load torque observer M_L; $\widehat{M}_L$

Simulation with the position controller when there is a variable heavy load is lifted with a large load that changes all the time as shown in Figure 8. Then the electro-hydraulic transmission system works well, the output output closely follows the input volume in the balancing process, as shown in Figure 8(a). Loading and unloading are also taken into account with the impact of noise and friction of the drive system over a certain period of time, which together with the variation of the load is transported by the forklift truck bring. The control torque response of the system always meets the working process, as shown in Figure 8(b). From the simulation research results in the environment of MATLAB/Simulink as Figures 6-8, we can see that: the output of the system changes during the transition, when the frictional moment on the motor shaft and the frictional moment on the load side change, with system oscillations with 2 times the number of oscillations; transient time is 0.16 s; the output still follows the input in the equilibrium process. The controller ensures stable working for the system to work accurately and reliably.



(a) (b)

Figure 8. Simulation with position controller I/O response (a) angular I/O response and (b) load torque observer M_L; $\widehat{M}_L$ when the load changes

5. CONCLUSION

In this paper presents the construction of an automatic control system for mobile lifting machines applied in industry and traffic such as electro-hydraulic systems, dynamic systems using hydraulic control valves. These new researches are completely applicable: self propelled robots, industrial robots, medical robots, electromechanical traction drive systems such as mobile transport lifts that use hydraulic valve systems to convey transmissions energy control (lifting and lowering as required) in industrial plants in Vietnam as well as in the world. The research results use programming and simulation tools on the basis of MATLAB/Simulink to demonstrate the results, moreover, it is also the basis for calculation and design for hydraulic electro drive system in the industrial sector.

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BIOGRAPHIES OF AUTHORS

Tran Duc Chuyen received the Ph.D. degree in Industrial Automation from Le Qui Don Technical University (MTA), Hanoi, Vietnam in 2016. Now, works at Faculty of Electrical Engineering, University of Economics-Technology for Industries. He is currently the President of Council the Science of Faculty of Electrical Engineering. Dr Tran Duc Chuyen′s main researches: electric machine, drive system, control theory, power electronics and application, adaptive control, neural network control, automatic robot control, motion control, IoT, and artificial intelligence. He can be contacted at email: tdchuyen@uneti.edu.vn.

Nguyen Thanh Tien was born in 1973 in Thai Binh; received a university degree in Electrical and Electronic Engineering from the Military Technical Academy in 1995; received a master's degree in Automation from the Academy of Science and Technology in 2004, received a doctorate in Automation from the Institute of Control Problems, Academy of Sciences of the Russian Federation 2008; currently deputy director of Military Institute of Mechanical Engineering/Ministry of National Defense; 42 Dong Quan, Cau Giay District, Hanoi. Research fields: nonlinear control, electric drive, and intelligent control. He can be contacted at email: ngttienktd@gmail.com.

Vu Duy Hung received the M.E (2009) in Electrical Engineering from Hanoi Universityof Science and Technology, Hanoi, Vietnam (HUST). He is a lecturer, Faculty of Electrical Engineering, University of Economic and Technical for Industries, Hanoi, Vietnam. His research interests include electric machine, drive system, automatic robot control, electrical machine control, and industrial automation. He can be contacted at email: vdhung@uneti.edu.vn.

Pham Ngoc Sam received the degree of Automation Engineer from Ho Chi Minh City University of Technology and Education, Vietnam, in 1999, Master's degree in Automation from Vietnam Academy of Agriculture, 2012. He is currently a lecturer. Member of the Department of Industrial Electrical, Faculty of Electrical Engineering, University of Economics-Technology for Industries. His research areas include motion control, control systems and its applications, electrical machines, electrical machine control, and industrial automation. He can be contacted at email: pnsam@uneti.edu.vn.

Tran Hai Duong received an engineering degree in 2018 in control engineering and automation, University of Economic and Technical for Industries, Vietnam (UNETI). Received a master's degree in control engineering and automation in Control Engineering and Automation at Hanoi University of Science and Technology (HUST). He is a lecturer at the Faculty of Electrical Engineering; University of Economic and Technical for Industries, Hanoi, Vietnam. His research areas include electrical machines, automatic robot control, electrical machine control, and industrial automation. He can be contacted at email: thduong@uneti.edu.vn.