

# Research on 4-DOF Adaptive Control of Hydraulic Excavator<sup>\*</sup>

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**Abstract** - The research status of robotic excavator was introduced and the electro-hydraulic system schematic diagram of test excavator was analyzed. On the base of principle of load independent flow distribution system, the mathematical model of test excavator was derived. The reference model for adaptive control was given after logically simplifying electro-hydraulic model by combination of theory and experiment results. According to the local parameter optimum, an adaptive controller was developed. The test was conducted on the platform modified by SWE17E of SUNWARD. And the experiment results show that: the tracking error of the angle is 13.8% when adopting traditional PID controller, while it is within 6.8% by using adaptive controller developed in the paper, it demonstrates that the controller designed in the paper can improve control precision and obviously promote the steady state response speed, anticipated control requirements of stability and robustness can be accomplished.

**Index Terms** – excavator electro-hydraulic proportional system load, independent flow distribution (LUDV) system, trajectory tracing, adaptive control.

## I. INTRODUCTION

The hydraulic excavator has the characteristics of large amount of users, high energy consumption, poor emission. According to statistics, the world's various earthworks about 65% to 70% is completed by hydraulic excavator[1]. With the continuous improvement of the people's requirement to the construction operation, the operators' requirement to the work environment is also improved, which makes the highlighted problems of the excavator can not be ignored in each application field[2-3]. Therefore, researches on the hydraulic excavator about intelligent and robot become the focus of the field. The literature[4] presents a multilevel feed-forward neural network control method. According to the error back-propagation algorithm for training, the excavator working device can move with two degree of freedom, and make the excavator achieve the level of line mining. The control precision is within 110mm. The literature[5] converted a John Deere 4410 series tractors with backhoe, and the machine integrated with a variety of intelligent sensor. A logging bio-medical tactile enhanced function can operate various actions on the testing machine through the interface panel. With the help of computer, the system, on the basis of the feedback

data from manipulator end and the state detection sensor to solve real-time kinematic parameters, can achieve the function of automatic mining. The literature[6] did some related transformation according to the Japanese Komatsu's PC45. The testing data shows that the tracking precision based on model reference adaptive control algorithm is highest. When using this algorithm, the tracking accuracy of reproduction is 20mm in the bucket tooth tip speed of less than 120mm/s. The literature[7-8] proposes a robust control method based on the control theory of time delay, and does many tests and transformations to the modern type HX60W-2 excavator. The result shows that the tracking error is within 100mm under the condition of the linear mining.

The study takes the Sunward electronic control type SWE17E excavator as the test platform which adopts the Rexroth LUDV system[9]. Based on the research of the literature[10], this considers the rotary motion of the excavator and establishes the four freedom of the electric hydraulic system model excavator. According to the basis, it starts the bucket trajectory tracking research of digging, loading(including rotating).

## II. ELECTRO-HYDRAULIC PROPORTIONAL SYSTEM MODEL

In this study, the basic platform is Sunward SWE17E electronically controlled hydraulic excavator. After the robot transformation, the electro-hydraulic proportional schematic diagram is shown in Figure 1.

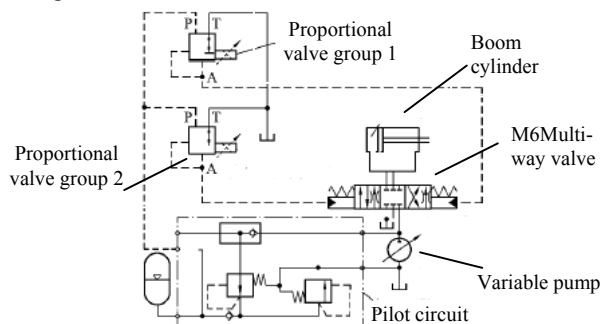


Fig. 1 Electro-hydraulic proportional system schematic diagram.

A. Dynamic characteristics of the electro-hydraulic proportional valve

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In the test, the electro-hydraulic system is mainly composed of both electro-proportional valve and hydraulic control multi-way valve. The relation of the electro-proportional valve output pressure and the multi-way valve spool displacement is linear, and its output pressure and control output current are linear relationship. Therefore, it can be assumed that the output current of electro-proportional valve and multi-way valve output are linear. The transfer function between the input current  $I$  of electro-hydraulic proportional valve and the spool displacement  $X_v$  is as follows :

$$X_v(s) / I(s) = K_I / (1 + bs) \quad (1)$$

$K_I$  is proportional valve current gain;  $b$  is the time constant in the inertia link.

#### B. Flow characteristic equation of electro-hydraulic proportional valve

The test platform SWE17E uses REXROTH Company LUDV system. According to its working principle, the flow characteristic equation of the valve can be deduced[9].

$$Q_1 = C_d W X_v \sqrt{2\Delta p_1 / \rho} = \begin{cases} C_d W X_v \sqrt{2\Delta p / \rho} & (I(t) \geq 0) \\ -C_d W X_v \sqrt{2\Delta(p_1 - p_r) / \rho} & (I(t) < 0) \end{cases} \quad (2)$$

$$Q_2 = C_d W X_v \sqrt{2\Delta p_2 / \rho} = \begin{cases} -C_d W X_v \sqrt{2(p_2 - p_r) / \rho} & (I(t) \geq 0) \\ C_d W X_v \sqrt{2\Delta p / \rho} & (I(t) < 0) \end{cases} \quad (3)$$

$p_r$  is the system back to the oil pressure;  $\Delta p$  is the setting pressure of regulator spring of load sensing valve;  $C_d$  is the valve flow coefficient;  $W$  is the opening gradient of valve ;  $\rho$  is the fluid density;  $I(t)$  is the input control current of proportional valve;  $Q_1$  is the large cavity flow of the cylinder, and  $Q_2$  the small cavity flow ;  $p_1$  is the large cavity pressure, and  $p_2$  is the small cavity;  $\Delta p_1$  and  $\Delta p_2$  respectively is the valve port pressure of both ends; When the excavator is in normal working condition, the setting value of  $\Delta p$  is constant, and usually is  $2Mpa$ .

The characteristic of LUDV system is that the valve flow and the open area of valve are positive correlation, regardless of the working load. Therefore, “(2)” can be simplified to :

$$Q_1(s) = K_q K_I I(s) / (1 + bs) \quad (I(t) \geq 0) \quad (4)$$

$K_q$  is the flow gain coefficient of valve,

$K_q = C_d W K_I \sqrt{2\Delta p / \rho}$ ,  $\Delta p = 2Mpa$  in the moment.

#### C. Hydraulic cylinder continuity equation

To simplify the calculation, the inside and outside leakage of hydraulic system is ignored. When the hydraulic cylinder is in process of the rod chamber back to the oil and the rod-less cavity into the oil (the rod chamber back to the oil and the rod-less cavity into the oil referring to the literature[11]), the fluid continuity equation of the hydraulic cylinder can be inferred[11].

$$\begin{cases} Q_1 = A_1 \dot{y} + V_1 \dot{p}_1 / \beta_e \\ Q_2 = A_2 \dot{y} - V_2 \dot{p}_2 / \beta_e \end{cases} \quad (5)$$

$V_1$  is the oil volume from the cylinder into the oil chamber to the multi-way valve oil outlet, and  $V_2$  is the oil volume from the cylinder back to the oil to the multi-way inlet;  $A_1$  is the effective area of rod-less cylinder chamber, and  $A_2$  the effective area of piston rod chamber;  $y$  is the effective displacement of cylinder piston;  $\beta_e$  is the elastic modules of oil volume.

#### D. The force balance equation of the hydraulic cylinder

Ignoring the hydraulic cylinder in the quality of the oil can obtain this equation :

$$F = p_1 A_1 - p_2 A_2 = M \ddot{y} + \omega \dot{y} + F_1 + F_f \quad (6)$$

$F$  is hydraulic drive force of system to hydraulic cylinder;  $\omega$  is whole viscous friction coefficient of hydraulic cylinder;  $F_1$  is load opposite force and its direction of movement In contrast to the direction of piston rod.  $M$  is equivalent mass of the piston rod;  $F_f$  is the friction of the piston rod.

#### E. Continuity equation of rotary motor

Ignoring the leakage of rotary motor can obtain the continuous fluid equation of oil return from multi-way valve to rotary motor inlet and motor exported to multi-way valve:

$$\begin{cases} Q_{in} - \omega D_m = (V_3 / \beta_e) \dot{p}_{in} \\ Q_{out} + (V_4 / \beta_e) \dot{p}_{out} = \omega D_m \end{cases} \quad (7)$$

$Q_{in}$  and  $Q_{out}$  are the entrance flow and export flow of rotary motor respectively;  $\omega$  is angular velocity of the variable motor;  $D_m$  is the displacement of hydraulic motor;  $V_3$  and  $V_4$  are volume of multi-way valve to hydraulic motor entrance and volume of multi-way valve to hydraulic motor export volume respectively;  $p_{in}$  and  $p_{out}$  are the entrance pressure and export pressure of hydraulic motor respectively.

#### F. Torque balance equation of rotary motor

$$D_m(p_{in} - p_{out})\eta_m = J \frac{d\omega}{dt} + b_m\omega + T_1 \quad (8)$$

$\eta_m$  is the mechanical efficiency of rotary motor;  $J$  is rotary platform equivalent to the inertia of the motor;  $b_m$  is viscous damping of rotating recover motor;  $T_1$  is outside load torque of rotary motor.

*G. Simplified model of electro-hydraulic proportional system*

Transforming “(4)-(6)” using the Laplace Transform and simplifying them can obtain the transfer function of electro-hydraulic proportional valve and hydraulic cylinder system:

$$Y(s) = [b_l X_v(s) + b_f s F_1(s)] / s(a_0 s^2 + a_1 s + a_2) \quad (9)$$

$$b_l = \beta_e K_q (A_1 V_2 + V_1 A_2^2 / A_1); b_f = V_1 V_2; a_0 = V_1 V_2 M; a_1 = \omega V_1 V_2; a_2 = \beta_e (V_2 A_1^2 + V_1 A_2^2).$$

Transforming “(4)”, “(7)” and “(8)”, using the Laplace Transform can infer the transfer function of electro-hydraulic proportional valve and rotary motor system:

$$\theta(s) = [c_l x_v(s) - c_t s T_1(s)] / s(d_0 s^2 + d_1 s + d_2) \quad (10)$$

$$d_0 = V_3 V_4 J; d_1 = 16\beta_e K_q J / D_m + b_m V_3 V_4; c_l = V_3 V_4; c_t = 4K_q \beta_e D_m; d_2 = 4\beta_e D_m^2.$$

### III. DESIGN OF ADAPTIVE CONTROLLER

The design of the adaptive controller is on the basis of the method of optimal local parameter. This method is suitable for low order system, especially for the first order stabilization system. And it are with the characteristics of good control effect and high the adaptive adjustment speed. Therefore, setting up first the stable first-order model is necessary before designing adaptive controller[12]. This study simplify further electro-hydraulic ratio model. During test, step current signal is inputted to electric proportional valve, and then obtain the flow-time test curve of tester hydraulic cylinder(See the reference [3]). The electro-hydraulic system can be dealt with as a first-order linear system and system with small effect in stable work condition. Simplifying “(9)” and substituting into “(1)”, can obtain:

$$Y(s) = K_q K_I I(s) / [s A_1 (1 + bs)] \quad (11)$$

According to “(11)”, the speed of hydraulic cylinder piston rod and transfer function input current of electro-hydraulic proportional valve can be obtained:

$$v(s) = K_q K_I I(s) / [A_1 (1 + bs)] \quad (12)$$

Similarly, the transfer function of rotary motor rotating angular velocity and input current can be gotten:

$$\omega(s) = K_q K_I I(s) / [D_m (1 + bs)] \quad (13)$$

In order to achieve control convenient, the following model as a controller reference model can be chosen:

$$v_r(s) = K_{qr} K_{lr} I(s) / [A_1 (1 + bs)] \quad (14)$$

Another important content of the adaptive controller design is to look for  $K$  to make following performance metrics to be minimum, basing on the optimal local parameter and using the optimal method of local parameter.

$$J_0 = \int_{t_0}^t e^2(\tau) d\tau = J_{\min} \quad (15)$$

Looking for partial derivative of this equation:

$$\frac{\partial J}{\partial K} = \int_{t_0}^t 2e(\tau) \frac{\partial e}{\partial K} d\tau \quad (16)$$

Takeing the initial step  $\alpha$  ( $\alpha$  is an arbitrary constant), reference along the gradient descent method:

$$\Delta K = -\alpha \frac{\partial J}{\partial K} = -\alpha \int_{t_0}^t 2e(\tau) \frac{\partial e}{\partial K} d\tau \quad (17)$$

Looking for partial derivative of this equation:

$$\dot{K} = -2\alpha e \frac{\partial e}{\partial K} \quad (18)$$

According to the control block diagram:

$$e(s) / I_r(s) = (K_{qr} K_{lr} - K K_{qK_I}) / [A_1 (1 + bs)] \quad (19)$$

Therefore,

$$e(s) = [\frac{K_{qr} K_{lr}}{A_1 (1 + bs)} - \frac{K K_{qK_I}}{A_1 (1 + bs)}] I_r(s) \quad (20)$$

Looking for partial derivative of this equation:

$$\partial e(s) / \partial K = -K_{qK_I} I_r(s) / [A_1 (1 + bs)] \quad (21)$$

Therefore,

$$A_1 (1 + bs) \partial e(s) / \partial K = -K_{qK_I} I_r(s) \quad (22)$$

The equation of the reference model is:

$$A_1 (1 + bs) v_r(s) = K_{qr} K_{lr} I_r(s) \quad (23)$$

With “(13)” divided by “(14)”:

$$\partial e(s) / \partial K = -K_{qK_I} I_r(s) / (K_{qr} K_{lr}) \quad (24)$$

Substitute “(15)” to “(20)” can obtain the adaptive control law based on the parameter  $K$  :

$$\dot{K} = 2\alpha e(s)K_q K_I I_r(s) / (K_{qr} K_{Ir}) \quad (25)$$

So far, design of theoretical model adaptive controller is complete. Combining with the system Parameter identification method of related research[13], obtaining :

$$K_q K_I = 2.663 \times 10^{-4} m / (sA^{-1}) \quad (26)$$

time constant is  $b = 0.2316s$  and  $\alpha = 0.01$ .

#### IV. EXPERIMENTAL RESEARCH AND COMPARATIVE ANALYSIS OF EXCAVATOR WORKING DEVICE

In order to verify simplified design rationality and effectiveness of the algorithm of the model adaptive controller model, the test results of the controller were compared with the conventional PID controller (the hardware, programs and the detailed results of PID control can be found in reference [3]). In order to more directly reflect the effect of the controller, the test collected the changing angel of boom, arm, bucket and swing in the working process. The zero calibration reference point of tilt sensor in the test is selected as: boom tilt sensor relative to the ground, the arm tilt sensor relative to the boom, bucket tilt sensor relative to the arm; the rotary encoder zero angle is perpendicular to the excavator front, through the rotation center line and perpendicular to the ground plane and all counterclockwise angles are positive, clockwise is negative. Here are the test results of the testing machine at light loads and loading conditions are typical.

In the test, the adaptive controller gets the tilt test value of the boom, arm, bucket and swing, which is basically the same as the controller input target value, but there are some fluctuations. From Figures 2 and 3, it can be assumed that:

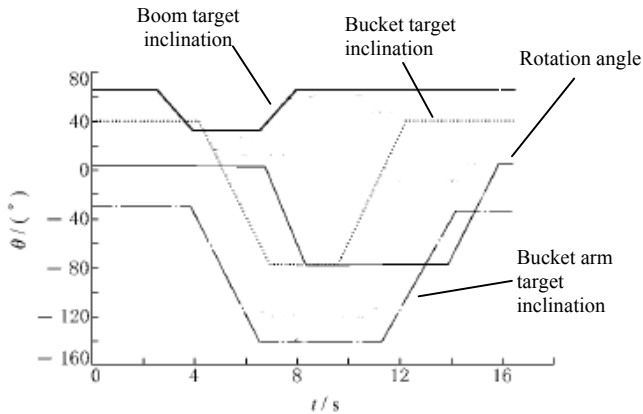


Fig. 2 Controller input target values for each tilt sensor.

The boom, arm, bucket and swing maximum amplitude fluctuation rates are 5.4%, 5.3%, 6.8%, 1.3%. From the comparison of Figures 2 and 4, it can be learned: in the same condition, when using PID controller, the boom, arm, bucket and swing maximum amplitude fluctuation rates are 10.4%, 11.3%, 13.8%, 9.0%. Comparing Figures 2 to 4, it also shows

that the convergence rate of adaptive controller relative to conventional PID controller is fast. Mainly in the input value kept unchanged, the tilt angle changes of the adaptive controller output control can quickly enter the stable state.

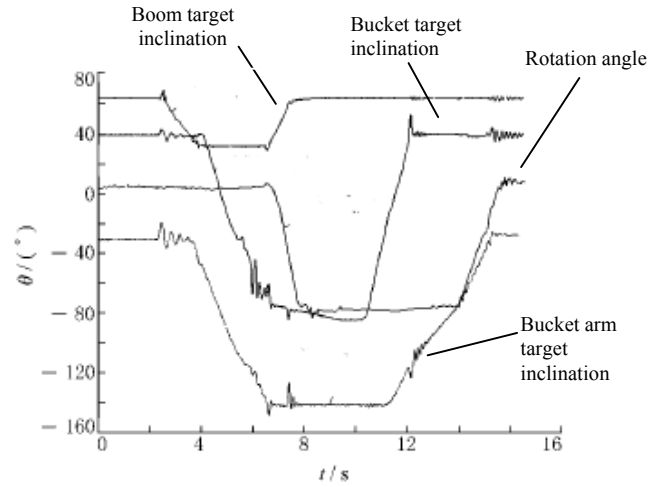


Fig. 3 The measured values of adaptive controller tilt sensor.

Figure 5 shows: Trends of the control law  $K$  values and controller design analysis is basically the same, and there are a small amplitude oscillations. Indicating that the actuator has a slight fluctuation in the work process, also that the regulation of control laws are more sensitive, the system has the good dynamics response performance.

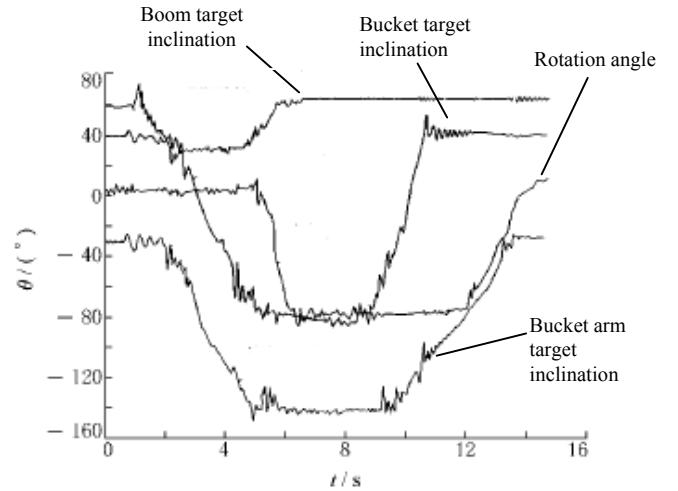


Fig. 4 PID controller tilt sensor measured values.

#### V. CONCLUSION

A. The article Analyses the research status of domestic and foreign mining robot. On the basis of two or three degrees of freedom robotic excavator automatic control research which is existing achievements, it considers the necessary conditions when the excavator is working-rotary and loading and does some transformations aiming at the test digging robot. According to the hydraulic system which the test machine used, the mathematical model in the four degrees of freedom

of electro-hydraulic proportional system is derived, which has a certain application value in engineering practice.

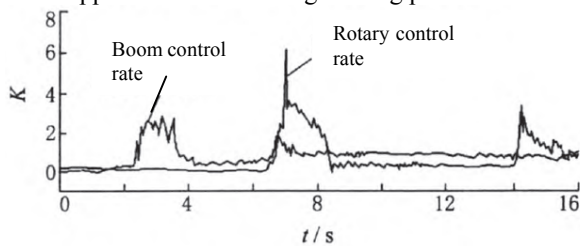


Fig. 5 The curve of control law values of the boom rotation.

B. On the basis of mathematical model of experimental electro-hydraulic system, the model of the whole system is further simplified by the method of combining with experimental research and theoretical analysis. The general model of adaptive controller is designed and completed, and mathematical model of adjustable parameter adaptive control law  $K$  can be obtained by local parameters optimal law.

C. Based on Sunward SWE17E excavators, a test platform of a mining robot is built. Tracking test results are compared between the controller designed and the conventional PID controller. The results showed that: the inclination of the tracking error obtained by conventional PID controller is up to 13.8%, but this study lets the inclination tracking error of adaptive controller keep within 6.8% and makes its accuracy greatly improve. In addition, the research of hundreds of hours reliability test is carried out, which proves that the system has good stability and robustness.

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