A Study and Development of an Electro-Hydraulic Pressure Control System for a Riveting Machine

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Abstract — An electro-hydraulic servo pressure control system is proposed to meet the requirements of fast response, zero overstep and high control precision by analyzing the techniques of riveting. The appropriate control parameters which are obtained by the results of the PID algorithm computations in PLC succeeded to achieve the precise control of system pressure. In practice the device has good application value.

Keywords - Servo control; Riveting machine; Control precision; PID arithmetic

I. INTRODUCTION

The steering ball stud which is the key to the integrity of car is the essential element and safety part of car. It should have enough strength, rigidity and service life when it was processing. The appearance of product and range of torque determine whether it’s a good product or not. The traditional way contenting the need of processing technique is the use of riveting directly once because of the down force of riveting which is general 350KN-630KN is so big that the request of deformation is achieved. This equipment use the technique of riveting of rolling whose the force of riveting-the production is step by step- is one tenth of the force of riveting directly in order to achieve the request of quality of whole product. However, the improper control of pressure leads to the shaking moment can’t meet the production requirement in the process. So the pressure of system should be accurately controlled in order to content the indexes such as fast response, zero overstep and good stability in the condition of low pressure [1, 2]. The System of Riveting Machine synthesizing hydraulic pressure, advanced manufacture, detection and control, computer and Internet and other related technique and subjects is an electro-hydraulic system of high property of high technical equipment. An important guarantee is that both advanced design of overall structure and the optimized configuration about parameter assure the working system in the practice about the use of Riveting Machine [3-6].

II. THE SYSTEM STRUCTURE

The riveting machine of ball stud consists of hydraulic system, mechanism, Industrial PC(IPC), PLC, sensors of detection and so on[7,8]. It is illustrated in Figure 1. The part of drive is hydraulic system which drives mechanism to accomplish the riveting of rolling; the control part is PLC which control hydraulic system and sensor of detection to analyze and calculate sampling value; Industrial personal computer and the PLC programme the serial communication protocol by themselves [9, 10, and 11].

![Figure 1. System Structure](image)

III. MATHEMATICAL MODEL OF HYDRAULIC SYSTEM

To meet the requirements of equipment with fast response, zero overstep, good stability and so on, a project which establish the electro-hydraulic pressure control system was presented. As shown in the Figure 2, this system consists of amplifier of servo valve, servo pressure valve, hydraulic cylinder, and pressure transducer[12,13,14].

![Figure 2. The electro-hydraulic servo pressure control system](image)

According to figure 2, use the following equation to calculate the bias voltage signal.

\[ U_c = U_r - U_f \]  

(1)

Where,
- $U_r$ is the command voltage signal.
- $U_f$ is the feedback voltage signal.

The equation for the pressure sensors:

$$ U_f = K_{sp} F_g $$ (2)

where,

- $K_{sp}$ is the pressure sensor gain.
- $F_g$ is the hydraulic cylinder output force.

The dynamic of servo valve amplifier can be ignored, the output current is given by

$$ \Delta I = K_a U_r $$ (3)

Where, $K_a$ is the gain of servo valve amplifier.

So, the transfer function of servo valve can be given by

$$ \frac{X_v}{\Delta I} = K_v G_{sv}(s) $$ (4)

Where,

- $X_v$ is the servo valve spool displacement.
- $K_v$ is the servo valve gain.
- $G_{sv}(s)$ is the transfer function, when $K_{sv} = 1$.

Assumed load is the quality, flexibility and damping, the hydraulic cylinder valve dynamic can be described by the following three available equations.

$$
\begin{align*}
Q_L &= K_q X_v - K_p p_L \\
Q_L &= A_p s X_p + C_q p_L + \frac{V}{4\beta_p} p_L \\
F_g &= A_p p_L = m_r s^2 X_p + B_p s X_p + K Q_p
\end{align*}
$$ (5)

Where,

- $m_r$ is the load quality.
- $B_p$ is the load damping coefficient.
- $K$ is the load spring stiffness.
- $C_{pL}$ is the total leakage coefficient of hydraulic cylinders.

According to the formula (5), the block diagram of the control system can be drawn, as shown in Figure 3. Where, $K_{ce} = K_x + C_{pL}$.

![Figure 3. The block diagram of the control system](image)

According to formula (5), the intermediate variables $Q_L$ and $X_p$ can be eliminated, we can get the transfer function between the spool displacement $X_v$ and the hydraulic cylinder output force $F_g$.

$$
\frac{F_g}{X_v} = \frac{K_q A_p (m_r s^2 + 1)}{K_{ce}} \frac{A_p m}{K_{ce}} + \frac{A_p^2}{K_{ce} K_h} s^2 \left( \frac{\beta_p A_p}{K_{ce} K_h} \right) s + 1
$$ (6)

Usually, the damping coefficient of load $B_p$ is very small, $B_p$ can be neglected. So, the formula (6) can be simplified as:

$$
\frac{F_g}{X_v} = \frac{K_q A_p (m_r s^2 + 1)}{K_{ce}} \frac{A_p m}{K_{ce} K_h} s^2 \left( \frac{\beta_p A_p}{K_{ce} K_h} \right) s + 1
$$ (7)

where $K_h$ is Hydraulic spring stiffness, $K_h = \frac{4\beta_p A_p^2}{V_r}$.

This system meets the condition:

$$
\begin{pmatrix}
K_{ce} \sqrt{K m_r} \\
A_p^2 (1 + K/K_h)
\end{pmatrix}^2 < 1
$$

Therefore, so we can write:

$$
\frac{F_g}{X_v} = \frac{K_q A_p (m_r s^2 + 1)}{K_{ce}} \frac{A_p m}{\omega_m^2 + 1} \left( \frac{\omega_m^2}{\omega_0^2} + \frac{2\omega_m^2}{\omega_0^2} s + 1 \right)
$$ (8)

Where,
$\omega_m$ is the load natural frequency, $\omega_m = \sqrt{\frac{K}{m_l}}$.

$\omega_r$ is the ratio of the spring stiffness and damping coefficient, when the hydraulic spring and load spring in serial connection, $\omega_r = \frac{K_p}{A_p} / \left( \frac{1}{K_h} + \frac{1}{K} \right)$.

$\omega_h$ is the natural frequency, which is formed because of the spring stiffness and load quality, when hydraulic spring and load spring in parallel connection, $\omega_h = \omega_n \sqrt{1 + \frac{K_h}{K}}$.

$\zeta_0$ is damping ratio, $\zeta_0 = \frac{\beta h}{2\omega_0 V_r (1 + K/K_s)}$.

Therefore, the figure 4 can be simplified into the figure 5, and the transfer function of this system become formula (9).

$$G(s)H(s) = \frac{K_0 G_f(s)}{(s / \omega_r + 1)(s / \omega_h + 1)}$$

Where $K_0$ is the open-loop gain of system, $K_0 = K_f K_s \frac{K_h}{K_p} A_p K_{ph}$.

IV. ANALYSIS BY SIMULATION

Establish the simulation model of hydraulic system by the Matlab to get the closed-loop step response of system.

In Figure 5, the rise time is at most 0.1s according to the figure, the transit time is at most 0.6s, but the overshoot of system is more than 35%, the peak value of oscillation amplitude is oversize. The shaking moment of part beyond the requested range because of high pressure.

As the overshoot in Figure 5 is so big, the increase of $K_d$ decreases overshoot and the decrease of transit time has little effect on rise time and static error to add the correction of PD. $K_d$ usually is chosen as the 8 times of reciprocal of oscillation period of system approximately.

In Figure 6, the response curve of system is still a style of attenuation of oscillation according to the figure, but the number of oscillation observably decreases and overshoot reduces largely and the problem now is that static error is still not to be zero. Because the increase of $K_i$ can eliminate the static error in order to add correction of PI, as shown in Figure 7. With considering the $K_i$ effect which largely decrease $K_p$, the value of $K_i$ can be taken big and decrease it step by step. The steady-state value of system is 1 after adding the PI correction, in other words, the static error is 0 so that output of system can indistinguishably track the change of set value eventually. However, with doing this result in another problem that observable increase of transit time so that rapidity is decrease.
Finally, it’s available to consider adding PID correction which uses the stable boundary method to get the parameter of Ki, Kg and Kd, then adjusting with drawing figure step by step. As shown in Figure 8. To apply the PID is very effective to general system because of similarity between response curve of system and step function. But it’s just an idealized condition of control so the live debugging is a need to revise the control parameter [15-18].

V. FIELD DEBUGGING OF PLC PARAMETERS

To establish a PID algorithm of a process of closed-loop control by calling created subroutine in the main program with PID Wizard provided in PLC, it could finish the task of PID control. It’s available to modify parameter of PID online in the monitoring pattern without halting to configure again with copying the address of control parameter into state table. As shown in Figure 9. The parameters wrote into block data or transmitted into corresponding data area by programming after the proper parameter are debugged. After that, the surveillance of operation of PID loop is achieved by graphics mode through panel of modulating control with debugging control parameter, the actual controlling curve basic meet the needs [19, 20]. As shown in Figure 10.

VI. CONCLUSIONS

According to the requirements of processing of equipment, the PID arithmetic is used in the Electro-hydraulic servo pressure control system. All the control parameters got by simulating are modified in the live debugging by PLC. In practice pressure curve can meet the reserved requests. The quality of product and the level of automation have been improved with riveting machine of ball stud used a device of electro-hydraulic servo system.

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