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Research Article

Research on Simulation of Giant Forging Hydraulic Press Decoupling Control for Synchronous Control System

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Abstract: A giant forging hydraulic press active synchronous control system is a mutually-coupled multi-input and multi-output system. To solve the elimination of the multi-channel interference, a multiple-input and multiple-output mathematical model center on active-beam is established; multi-channel synchronous decoupling control strategy is studied. The simulation results show that: the system eliminates the role of strong interference between multi-channel accesses to very good inhibitory effect of synchronization error, eliminating the system's external disturbance on the synchronization precision control impact.

Keywords: Decoupling, hydraulic press, synchronous control

INTRODUCTION

Giant forging hydraulic press plays a vital role in defense and aviation industries for a large forging production. As the hydraulic system of the non-linear, time-variant, multi-cylinder sports cross-coupling effects; eccentricity produced by forging non-symmetry of processing of parts, as well as the activities of beams and frame can not have absolute rigidity, power flow transmission distortion occurs and so on, leading to beam tilt during operation ,affect the study piece machining accuracy. The role of synchronous control system is to prevent the activities of beams to withstand eccentric torque occurs when the tilt, so that it remains at a high level degree of accuracy within the scope in order to ensure the required dimensional precision forging parts, but also conducive to improving the hydraulic rack force status to extend the service life of hydraulic press (Zhongwei, 2007).

In the past, giant forging press are used in the active beams under the four corners of the layout of a passive synchronous control systems, which will not only increase the cost of investment in equipment and its maintenance, but also some special hydraulic press due to space issues, can not be arranged passive synchronization control systems.

In this study, a five-cylinder-driven giant forging hydraulic press, proposed a program of active synchronous control system based on the multi-point-driven. By controlling flow and pressure of the five main cylinder's oil line to alter speed of the main cylinder which layout the four corners and balance torque generated by the active beam, the concrete gesture of the active beam depends on various moments of the mutual coupling. Deflection when the active beam did not occur, the five master cylinder's pressure and flow rate equal to, the master cylinder at the same rate, so at this time balance torque which the master cylinder affect on active beam is zero; when the active beam occur deflection, through adjusting the flow rate of compensation of Proportional flow valve, resulting in a reasonable balance torque to restore the level of the active beam (Xinliang, 2010) discussed. Thereby this study could provide some foundations to resolve similar problems.

THE MATHEMATICAL MODEL OF SYNCHRONOUS CONTROL SYSTEM

Active beam as the research object. As the driving force of the central cylinder has always been forced the center of the active beam, so it will not produce eccentric moment to active beam, we can ignore the influence of the central cylinder. The mechanical analysis shown in Fig. 1: XY plane is the level of the active beam plane, to XY plane geometry center horizontal X-axis and vertical Y-axis for the decomposition of the torque on it. Beam deflection is the event where the plane and the angle between the X-axis as θ_x and the angle between Y axis as θ_y, θ_1 and θ_2 is the angle between the two diagonal and the original level plane when the active beam deflection.

By geometric characteristics and dynamics theory can be:

$$A\ddot{\theta} + B\dot{\theta} + C\theta = M_T + M_l \tag{1}$$

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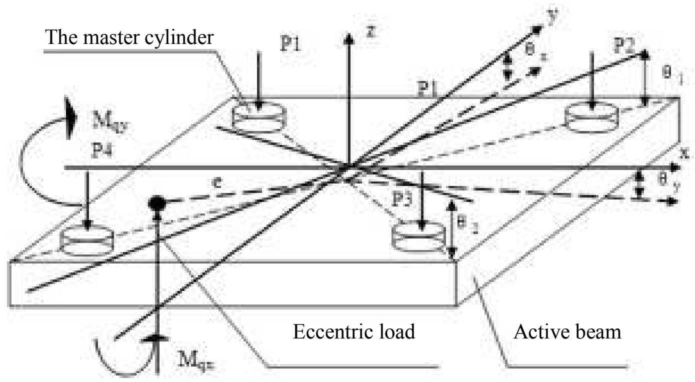

Fig. 1: Force model of the active beam

where,

A, B, C = The system coefficient matrix associated with Structural parameters

M_T = Balance torque generated by the main hydraulic cylinder

M_l = External disturbance torque, including eccentric torque M produced by the study and friction torque between the active beam and the ball and socket

By the characteristics of the electro-hydraulic proportional flow valve and the system can be:

$$M_T\,[M_{Tx}\; M_{Ty}]^T = DU \tag{2}$$

$$U = \left[-\int u_1dt \quad -\int u_2dt \quad -\int u_3dt \quad -\int u_4dt\right]^T = \left[U_1 \quad U_2 \quad U_3 \quad U_4\right]^T$$

D = The coefficient matrix associated with the parameters.

Figure 2 shows the analysis diagram of active beam external interference anti-torque, by characteristics of spherical hinge friction (the whole process are regarded as dynamic friction) and the mechanical analysis of the system can be drawn:

Eccentric Moment:

$$\begin{cases} M_x = Fe_x \\ M_y = Fe_y \end{cases} \tag{3}$$

Friction Moment:

$$\begin{cases} M_{M_x} = \left(F_1 + F_2 + F_3 + F_4\right) L_x \beta_x \\ M_{M_y} = \left(F_1 + F_2 + F_3 + F_4\right) L_y \beta_y \end{cases} \tag{4}$$

where,

L_x = The distance from the master cylinder to the x-axis

L_y = The distance from the master cylinder to the y-axis

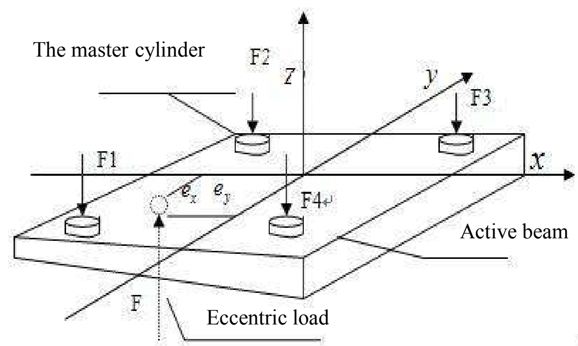

Fig. 2: The analysis diagram of active beam external interference anti-torque

β_x = The friction coefficient when active beam connecting the main cylinder spherical joint rotation around the x-axis

β_y = The friction coefficient when active beam connecting the main cylinder spherical joint rotation around the y-axis

Joint formula (1), (2) received the final mathematical model of the system as follows:

$$A\ddot{q} + B\dot{q} + Cq = DU + M_l \tag{5}$$

From this system is a four input $[U_1\ U_2\ U_3\ U_4]^T$, two output $[q_1\ q_2]^T$ system and the system between input and output coupling, so the system has a strong coupling, higher order and so on.

- **Decoupling control of synchronous control system:** In the actual study process in order to reduce the coupling between the master cylinder and the difficulty of controller design, each time correcting the process, according to the scope and positive and negative of the

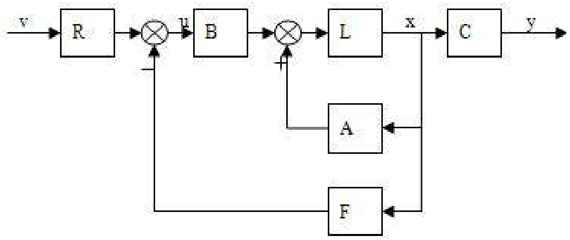

Fig. 3: Diagram of decoupling control equation of state

deviation, selection the two master cylinder of the four master cylinder to the closed-loop, while the proportion of flow valve of the other two master cylinder is not adjusted to maintain the original pressure and speed. So, around the layout of the four master cylinder is divided into two groups, each of the corner a group. Therefore, mathematical model of the active control systems can be simplified a two input two output system.

State-feedback decoupling control: State-space of the closed-loop system can be described as follows:

$$\sum(A-BF \quad BR \quad C): \begin{cases} \dot{x} = (A-BF)x + BRv \\ y = Cx \end{cases} \tag{6}$$

Input transformation matrix: A: n×n, B: n×n, C: m×n, R: m×m
Feedback gain matrix: F: m ×n
Reference input: v: m×1

The organizational structure of the state feedback system of input transform shown in Fig. 3.

Decoupling control system can control the existence of a matrix of the closed-loop system transfer function matrix is diagonal.

Control system can be decoupled is the existence of matrix to {R, F} so that the transfer function matrix of closed-loop system $G_L(s) = C\ (sI– A + BF)^{-1}\ BR$ is diagonal matrix.

That is:

$$G_L(s) = diag\left[g_{11}(s) \quad g_{22}(s) \quad \ldots \quad g_{mm}(s)\right] \tag{7}$$

And $g_{ii}(s) = \dfrac{y_i(s)}{v_i(s)}, i = 1, 2, \cdots, m$

d_i is defined to meet the inequality: $C_i A^l B \neq 0$, $l = 0, 1, \ldots, m-1$

C_i is the i row vector of the output matrix C, so that the subscript of d_i is the number of rows.

Define the following matrix according to d_i:

$$D = \begin{bmatrix} C_1 A^{d_1} \\ C_2 A^{d_2} \\ \vdots \\ C_m A^{d_m} \end{bmatrix}, \quad E = DB = \begin{bmatrix} C_1 A^{d_1} B \\ C_2 A^{d_2} B \\ \vdots \\ C_m A^{d_m} B \end{bmatrix}, \quad L = DA = \begin{bmatrix} C_1 A^{d_1+1} \\ C_2 A^{d_2+1} \\ \vdots \\ C_m A^{d_m+1} \end{bmatrix}$$

Necessary and sufficient conditions which controlled system can be decoupled is matrix E is nonsingular. State feedback matrix $F- E^{-1} L$ and input transformation matrix $R = E^{-1}$ can be drawn.

The original system transfer function is:

$$G_0(s) = C(sI-A)^{-1} B = diag\left[\frac{1}{s^{d_1+1}} \quad \frac{1}{s^{d_i+1}} \quad \cdots \frac{1}{s^{d_m+1}}\right]\left[E + L(sI-A)^{-1} B\right] \tag{8}$$

The decoupling system transfer function matrix is:

$$G_L(s) = G_0(s)\left[I + F(sI-A)^{-1} B\right]^{-1} R \tag{9}$$

Therefore, the formula (8) into formula (9) can be drawn:

$$G_L(s) = diag\left[\frac{1}{s^{d_1+1}} \quad \frac{1}{s^{d_i+1}} \quad \cdots \frac{1}{s^{d_m+1}}\right]\left[E + L(sI-A)^{-1} B\right]\left[I + F(sI-A)^{-1} B\right]^{-1} E^{-1}$$

$$= diag\left[\frac{1}{s^{d_1+1}} \quad \frac{1}{s^{d_i+1}} \quad \cdots \frac{1}{s^{d_m+1}}\right] \tag{10}$$

Therefore, the transfer function matrix of the closed-loop system available is:

$$G_L(s) = C(sI - A + BF)^{-1} BR = \begin{bmatrix} \frac{1}{s^{(d_1+1)}} & & \\ & \ddots & \\ & & \frac{1}{s^{(d_m+1)}} \end{bmatrix} \tag{11}$$

Formula (11) shows that the original system has been achieved decoupling to (F, R).

Pole placement of decoupling controller: As the poles of the integrator decoupled system are at the origin, the dynamic performance is not satisfactory, thus need to re-configure the system's poles, re-do of a state feedback based on integrator decoupled system to achieve the required configuration poles (Zhong, 2007).

Decoupling has been implementation of the integrator system do following $\hat{A}$ $\hat{B}$ $\hat{C}$ transformation:

$$\hat{A} = \begin{bmatrix} A_1 & & & \\ & A_2 & & \\ & & \ddots & \\ & & & A_p \end{bmatrix}, \quad \hat{B} = \begin{bmatrix} b_1 & & & \\ & b_2 & & \\ & & \ddots & \\ & & & b_p \end{bmatrix}, \quad \hat{C} = \begin{bmatrix} C_1 & & & \\ & C_2 & & \\ & & \ddots & \\ & & & C_p \end{bmatrix} \tag{12}$$

where $A_i = \begin{bmatrix} 0 & 1 & & \\ & 0 & \ddots & \\ & & \ddots & 1 \\ & & & 0 \end{bmatrix}$, $b_i = \begin{bmatrix} 0 \\ \vdots \\ 0 \\ 1 \end{bmatrix}$, $C_i = [1\ 0\ \dots\ 0]$

And satisfies: $d_1 + d_2 + \dots + d_p + p = n$
Claimed that the system is decoupled standard,.
Subsystem ($A_i\ b_i\ C_i$) using state feedback:

$$V_i = \hat{K}_i \hat{x}_i + \omega_i, i = 1, 2, \cdots, p \tag{13}$$

The system can achieve pole placement, $\hat{K}$ has the following form:

$$\hat{K} = \begin{bmatrix} k_{10} \cdots k_{1d_1} & & & \\ & k_{20} \cdots k_{2d_2} & & \\ & & \ddots & \\ & & & k_{p0} \cdots k_{pd_p} \end{bmatrix} \tag{14}$$

Therefore, the closed-loop function can be drawn by decoupling the system and after pole configuration:

$$\hat{G}_L = \hat{C}\left[sI - \hat{A} - \hat{B}\hat{K}\right]^{-1}\hat{B} = \begin{bmatrix} \frac{1}{g_1(s)} & & & \\ & \frac{1}{g_2(s)} & & \\ & & \ddots & \\ & & & \frac{1}{g_p(s)} \end{bmatrix} \tag{15}$$

where,
$g_i(s) = s^{d_i+1} + k_{id_i}s^{d_i} + k_{id_{i-1}}s^{d_i-1} + \cdots + k_{i1}s + k_{i0}$

The original system after decoupling control and pole placement can be simplified as a separate sub-circuit, which greatly simplifies the complexity of the system, it can follow the single-loop design of some basic ways to deal with the whole system. Such as PID, fuzzy control(Yu, 2005).

DECOUPLING CONTROL SIMULATION

For this system, calculated as described above:

$$E = \begin{bmatrix} -16.6002 & -1.7302 \\ -1.7302 & -16.6002 \end{bmatrix}$$

$$F = \begin{bmatrix} -10.9171 & -5.6631 & 5.5289 & 2.2548 \\ -5.5289 & -2.2548 & 10.9171 & 5.6631 \end{bmatrix}$$

Therefore,

$$\bar{A} = \begin{bmatrix} 0 & 1 & 0 & 0 \\ 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 1 \\ 0 & 0 & 0 & 0 \end{bmatrix} \bar{B} = \begin{bmatrix} 0 & 0 \\ 1 & 0 \\ 0 & 0 \\ 0 & -1 \end{bmatrix} \bar{C} = \begin{bmatrix} 1 & 0 & 0 & 0 \\ 0 & 0 & -1 & 0 \end{bmatrix}.$$

$$L = E^{-1} = \begin{bmatrix} -0.0609 & 0.0063 \\ 0.0063 & -0.0609 \end{bmatrix}$$

$$K = -E^{-1}F = \begin{bmatrix} -0.6298 & -0.3306 & 0.2674 & 0.1014 \\ -0.2674 & -0.1014 & 0.6298 & 0.3306 \end{bmatrix}$$

Decoupled system will be transformed into decoupled Standard:

$$\bar{A} = \begin{bmatrix} 0 & 1 & 0 & 0 \\ 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 1 \\ 0 & 0 & 0 & 0 \end{bmatrix} = \begin{bmatrix} A_1 & 0 \\ 0 & A_2 \end{bmatrix} \bar{B} = \begin{bmatrix} 0 & 0 \\ 1 & 0 \\ 0 & 0 \\ 0 & -1 \end{bmatrix} = \begin{bmatrix} B_1 & 0 \\ 0 & B_2 \end{bmatrix}$$

$$\bar{C} = \begin{bmatrix} 1 & 0 & 0 & 0 \\ 0 & 0 & -1 & 0 \end{bmatrix} = \begin{bmatrix} C_1 & 0 \\ 0 & C_2 \end{bmatrix}$$

After the decoupling of two independent subsystems $\sum_1 = (A_1\ B_1\ C_1)$ and $\sum_2 = (A_2\ B_2\ C_2)$ for the state feedback:

For $\sum_1 = (A_1\ B_1\ C_1)$ has $v_1 = [k_1 k_2] \begin{bmatrix} x_1 \\ x_2 \end{bmatrix} + u_1$,
For $\sum_2 = (A_2\ B_2\ C_2)$ $\sum_2 = (A_2\ \ B_2\ \ C_2)$ has $v_2 = [k_3 k_4] \begin{bmatrix} x_3 \\ x_4 \end{bmatrix} + u_2$

Therefore, the characteristic polynomial of the closed-loop system is:

$$\det\left[\lambda I - \left(\bar{A} + \bar{B}\hat{K}\right)\right] = \det\begin{bmatrix} \lambda & -1 & 0 & 0 \\ -k_1 & \lambda - k_2 & 0 & 0 \\ 0 & 0 & \lambda & -1 \\ 0 & 0 & k_3 & \lambda + k_4 \end{bmatrix} =$$
$$\lambda^4 + (k_4 - k_2)\lambda^3 + (k_3 - k_2k_4 - k_1)\lambda^2 - (k_2k_3 + k_1k_4)\lambda - k_1k_3$$

According to the actual performance of the system, expect the poles of the closed-loop system is $-1 \pm i$, $-1 \pm i$. Achieve the desired characteristic polynomial is $\lambda^4 + 4\lambda^3 + 8\lambda^2 + 8\lambda + 4$.

Therefore,

$$\begin{cases} k_4 - k_2 = 4 \\ k_3 - k_2k_4 - k_1 = 8 \\ k_2k_3 + k_1k_4 = -8 \\ k_1k_3 = -4 \end{cases} \quad \text{Obtained} \begin{cases} k_1 = -2 \\ k_2 = -2 \\ k_3 = 2 \\ k_4 = 2 \end{cases}$$

That is $\hat{k} = [-2\ -2\ 2\ 2]$
So the system transfer function matrix is:

$$\hat{G}_L = \begin{bmatrix} \frac{1}{s^2 + 2s + 2} & \\ & \frac{1}{s^2 + 2s + 2} \end{bmatrix} \tag{16}$$

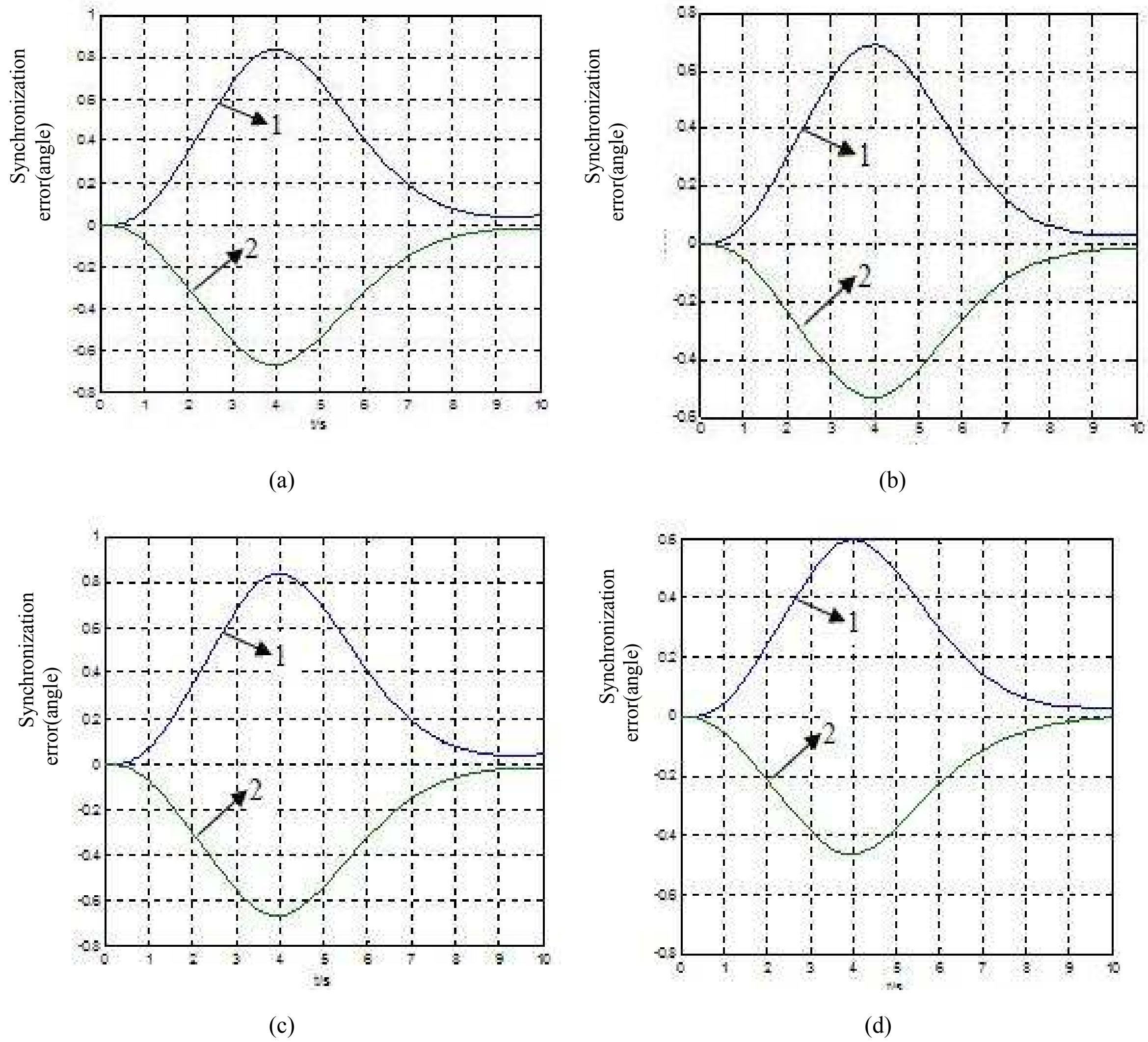

Fig. 4: Simulation results; (a) Decoupling control system synchronization error; (b) 20% increase in eccentric loading synchronization error; (c) 50% increase in eccentric loading synchronization erro; (d) Eccentric load reduced by 20% synchronization error 1-Angle θ_1 2-Angle θ_2

From formula (16) showed that the original strong coupling system to achieve dynamic decoupling. The following will be controlled separately of the two subsystems decoupled, in their control loop to set a PID controller to achieve decoupling control for the multi-cylinder hydraulic driven of synchronous.

By the simulation results (Fig. 4), giant forging hydraulic press active synchronization control system decoupled by state feedback, in their use of the classic PID loop control strategy, strong coupling system obtain a very good dynamic and static performance. The simulation results are based on eccentric loading on the activities of the diagonal beams, eccentric loading time is 4 seconds, the normal eccentric load is 6250N (Fig. 4a), eccentricity is 30cm. Figure 4a shows the synchronization error increases with the eccentric load increasing, mainly due to the lag of the characteristics of multi-cylinder driven hydraulic system synchronous control system and inertia of active beam. The maximum dynamic error occurs in about 4 seconds, the value of θ_1 is 0.64 degree, θ_2 is 0.48 degree, Just eccentric loading has finished loading ,the corner of the synchronization error is completely designed to meet the system requirements on the dynamic error (less than 1 degree), the system complete correction of about 7 seconds, two angles of the synchronous steady-state error is maintained within 0.5 degree. It meet the system steady-state error requirements.

Form Fig. 4b and c can be seen that the eccentric load when the system increased 20 and 50%, the system is the maximum dynamic error occurred in 4 seconds, the maximum were 0.7 degrees and 0.82 degrees. However, when the system load was reduced by 20%, the error shown in Fig. 4c shows, the system's maximum

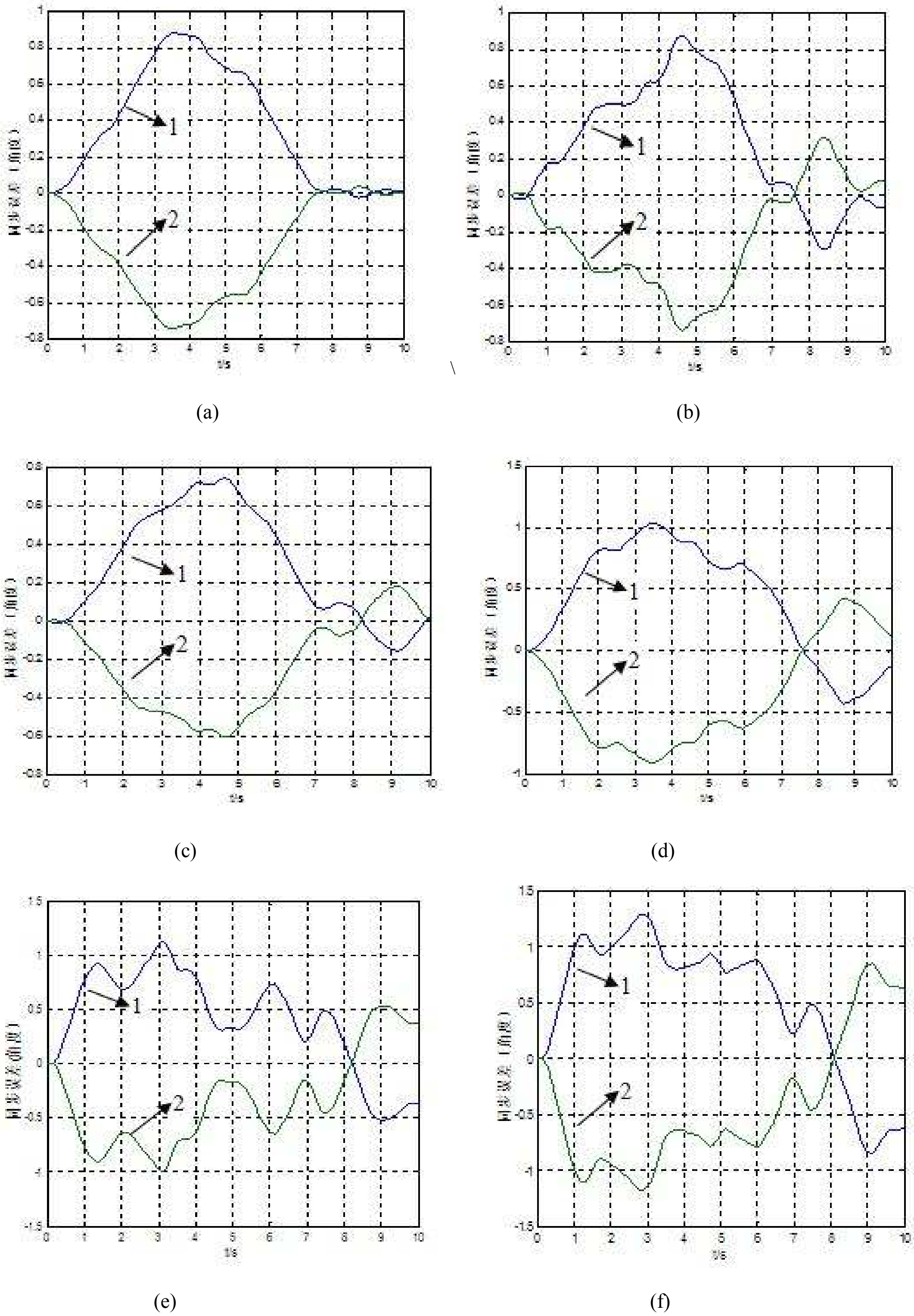

Fig. 5: Simulation results for parameter fluctuations; (a) Friction coefficient increased by 20%; (b) Friction coefficient increased by 50%; (c) Damping coefficient increased by 20%; (d) Damping coefficient increased by 50%; (e) Friction coefficient and damping are increased by 20%; (f) Friction coefficient and damping are increased by 50% 1- Angle θ_1 2-Angle θ_2

dynamic error θ_1 is 0.6 degrees. Eccentric load in the process of change, only affects the system dynamic error, while not affect the basic stability of the system, the time system to stabilize about 7 seconds. So the giant forging hydraulic press synchronous control systems should minimize or avoid eccentric loading, The same eccentric load, should be installed in the forging of the mold as far as possible the middle of the active beam to reduce the eccentricity.

Closer to the real system, to verify the decoupling controller adjustment performance, analysis of system parameters fluctuations. Figure 5 is the friction coefficient and damping coefficient increased 20 and 50% in the performance of the synchronization error simulation results.

From the above simulation results show that, when the friction coefficient increased by 20 and 50%, the system of synchronization error occur fluctuations, the system also increased the maximum dynamic error of 30%, mainly because the friction coefficient increased equivalents to increase the active beam's anti-bias external interference torque and the system had a strong all-channel interference, making the synchronization error is changed. However, giant forging hydraulic press owns to the characteristics of large inertia, the synchronization error of change more slowly. Similarly there are from Fig. 5c and d, We find that when the system damping is also increased by 20% and 50%, the system also had a wave synchronization error, but when the system damping coefficient increased by 50%, the system more than the maximum dynamic error system, the maximum allowable value (1 degree), this mainly because when the system damping increases, weakening the system to produce active corrective torque. Figure 5e and f know that when the system friction and damping coefficients are increased by 20 and 50%, severe fluctuations in the system, the system has exceeded the maximum dynamic error is designed to allow, the fluctuations of the system parameters makes the system very strong coupling, the system decoupling controller at this time has failed.

CONCLUSION

Giant Hydraulic Forging press active synchronization system after decoupling control based on state feedback the elimination of the Multi-channel interference between the strong, into several independent subsystems, each subsystem using the classical control algorithm to compensate, to gain a good inhibition effect of synchronization error, eliminating the system of foreign interference on the synchronization precision control. However, because the system is multi-cylinder synchronous control system, although the system of five separate sub-circuit in the hydraulic components and layout are the same program, but because of processing errors hydraulic components and hydraulic system itself nonlinear characteristics, as well as unpredictable external disturbance factors make the system parameters such as: friction coefficient, damping coefficient, activity coefficient of the beam stiffness of the study in different occurred during the large fluctuations. The decoupled controller parameters are obtained under the circumstances the system parameters remained unchanged, when the system parameter fluctuations, through decoupled subsystems may occur as the system parameters corresponding volatility fluctuations. Through the above can be seen after increasing the friction coefficient and damping system the system's synchronization error occurred fluctuations, even beyond the maximum allowable design value. Therefore, fluctuations occur when the system parameters, the stability of the decoupled system is not strong, decoupling controller requires a system of high precision mathematical model .When the system is stable in the working conditions, Small range of load changes and the forging of a range of eccentric torque of the system can predict when the decoupling controller can be used to correct the system so that the system reduced to several independent single-input single-output system , then using the classical control methods to compensate the system error, this method has a clear design concept, easy implementation. However, complex systems working conditions, a large load range and the system can not predict when the eccentricity of foreign interference, still use the simplified system, decoupling control algorithm, the stability of the system will change, the synchronization error of the system will far exceed even allowed value. In this case, inhibited and robust control algorithms to the system parameters such as the robust controller to compensate the system should be considered.

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REFERENCES

Xinliang, L., 2010. Research of Active Synchronization Control System of Giant Forging Hydraulic Press [D]. Central South University, Changsha.

Yu, C., 2005. Modern Control Theory [M]. 3rd Edn., Harbin Institute of Technology Press, Harbin.

Zhong, J., 2007. Coupling Design theory and Methods of Complex Electromechanical Systems [M]. Machinery Industry Press, Beijing.

Zhongwei, L., 2007. Hydraulic synchronous control system and its application on a giant hydraulic press [J]. Hydraulic Pneum. Sealing, pp: 7-10.