Position Control Algorithm of Fuzzy Adaptive PID of Hydraulic Interconnected Suspension Under Load Impact Disturbance

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ABSTRACT Position control accuracy of hydraulic interconnected suspension (HIS) for agricultural vehicles is unsatisfied under load impact conditions. In order to address this problem, advanced control methods were studied in this paper. The technical scheme of HIS was developed firstly. The transfer function from proportional valve spool displacement to hydraulic cylinder piston displacement was derived and then the parameters of the transfer function were also identified. The PID, sliding mode control (SMC) and fuzzy adaptive PID (FAPID) control algorithms were designed. The co-simulation of AMESim and SIMULINK was adopted to compare and analyze the control characteristics of the three algorithms. An experimental platform was also built to verify the improvement effect. The research conclusions show that when the HIS operate under load impact condition, for FAPID algorithm, the rise time is reduced by 0.27s, the maximum overshoot is reduced by 9.83mm, and the absolute value of average error (AVAE) is decreased by 62.62% compared with PID control algorithm. The response speed of SMC is the fastest, but there is a large chattering in the steady-state displacement curve. The designed FAPID control algorithm balances the advantages of fast response, high control precision and good stability, and the control performance is better than that of SMC and PID control algorithms. The research methods in this paper can provide reference value for position control accuracy improvement of hydraulic suspension.

INDEX TERMS fuzzy adaptive PID, hydraulic interconnected suspension, load impact, position control, sliding mode control

I. INTRODUCTION

A. RESEARCH BACKGROUND

Hydraulic interconnected suspension (HIS) is widely used in special or agricultural vehicles, which requires multiple hydraulic cylinders to rise and fall synchronously and have high position synchronization accuracy. Agricultural vehicles are often required to operate on slope surface, and there may be a risk of tipping in the process. HIS can realize the functions of anti-pitching and anti-roll through the synchronous action of multiple hydraulic cylinders. A more popular way is that hydraulic synchronizer, such as synchronous motor or shunt collector valve, is adopted for shunting flow rate forcibly to ensure synchronization accuracy. But this synchronization method has a large pressure loss. Furthermore, it can only realize one kind of anti-pitching and anti-roll function. In order to solve the above problems, electronically controlled HIS which can arbitrarily control the synchronous action of multiple hydraulic cylinders has become a research hotspot in recent years. One of the most significant current discussions in electronically controlled HIS field is that how to improve the synchronization accuracy under load impact disturbance. This is mainly because the load impact disturbance of the hydraulic cylinder may be caused by crossing trenches, climbing ridges and other reasons during the driving process of agricultural vehicles. The synchronization accuracy of electronically controlled HIS may be seriously affected by load impact disturbance. Therefore, it is of great significance to study the control optimization algorithms of HIS to counter the effects of load impact.
B. LITERATURE REVIEW
In recent years, the study of HIS mainly focuses on pitch dynamic, intelligent control algorithm, ride comfort characteristic, global sensitivity analysis, etc. In a study conducted by Mahmoud Omar et al (2018), a new position-composite control method for electro-hydraulic servo system was proposed. The open-loop gain was increased through positive velocity feedback, and the damping ratio was optimized through negative acceleration feedback. It can improve the response speed and reduce the position error. The simulation data show that the dynamic response speed of the system is about 26.9% higher than that of PID if the designed compound control strategy was employed. Andrei-Cristian Pridie et al (2021) studied the synchronous control method of lifting and lowering of vehicle HIS. A simulation model of hydraulic synchronous system in AMESim was established, and the synchronous rising and falling process was simulated under various operating conditions. They found that the maximum displacement synchronization error is 16.5mm. To further improve the control accuracy, Chen Bin et al (2016) carried out a series of studies on the fuzzy adaptive PID (FAPID) control algorithm of hydraulic quadruped robot. The influence of nonlinear and time-varying factors on the control characteristics was analyzed in detail. The simulation results show that the FAPID algorithm is much better than the conventional PID in reducing the adjustment time and suppressing the interference. In addition, a significant discussion on HIS multi-objective optimization was presented by Zhang Nong et al (2020). Taking the performance indexes reflecting the vehicle ride comfort and anti-roll performance as the objective function, the Sobol index method was adopted to optimize the HIS parameters. After optimization, the acceleration of the center of mass was reduced by 20.95%, and the anti-roll acceleration was reduced by 8.05%. The influence of synchronous motor parameters on the synchronization performance of dual hydraulic cylinders was analyzed in detail by Liu Xiliang et al (2020). They found that when the synchronous motor speed is lower than 50rev/min, the synchronization error will be more than 15mm. The synchronization error at 50 °C of hydraulic oil is 2 to 3 mm larger than that of hydraulic oil at 30 °C. So far, however, the method of HIS position control accuracy under the load impact condition has been discussed rarely. The major limitation of previous studies is that these methods may not work, when the external load on the hydraulic suspension actuators changes dramatically.

C. PAPER ORGANIZATION
In this study, FAPID, sliding mode control (SMC) and PID algorithms are developed to solve the problem of poor position control accuracy of HIS under load impact condition. The technical scheme of the HIS is designed, and then the transfer function from the spool displacement of the proportional valve to the piston displacement of the hydraulic cylinder is obtained by theoretical analysis and model identification. The control characteristics of the three algorithms are compared and analyzed by using co-simulation of AMESim and SIMULINK. To verify the correctness and superiority of the algorithms, an experimental platform of HIS is established. The research methods and conclusions of this paper can be extended to other hydraulic cylinder position control system.

II. HIS Design
According to the requirements of HIS, the technical scheme (Fig.1) was developed. Each hydraulic cylinder of the designed HIS is driven by an independent gear pump, and the oil outlet of the gear pump is connected to a three-position and four-way proportional reversing valve with M-type median function which has dual functions of direction switching and flow rate adjustment. When it is in the neutral position, the hydraulic oil output from the gear pump flows directly back to the tank. This can reduce the pressure loss and calorific value. When it is in the left position, the hydraulic cylinder overcomes the external force to extend, and when it is in the right position, the hydraulic cylinder moves in the opposite direction. The size of the throttle opening of the proportional directional valve is proportional to the external control signal, through the internal pressure compensation mechanism. The hydraulic one-way valve group is used to lock the hydraulic cylinder and prevent it from moving due to external load shock. A cable displacement sensor is installed on the double-acting hydraulic cylinder, and its output analog signal can be collected by the controller in real time. After the conversion of the controller, the stroke information of the hydraulic cylinder can be obtained dynamically. At the same time, the controller combines the expected displacement and real-time feedback information, and outputs the electrical signal to control the proportional reversing valve to realize the synchronous action of the hydraulic cylinder.
III. ESTABLISHMENT AND IDENTIFICATION OF CONTROL MODEL

A. DERIVATION OF CONTROL MODEL

The power transfer path of the designed HIS is essentially an asymmetrical hydraulic cylinder controlled by a symmetrical four-way valve. Its physical model is given in Fig. 2. It is necessary that the following assumptions are proposed in the derivation of the transfer function. Firstly, fluid is incompressible and pressure is equal in the connected area. Secondly, pressure loss in the pipeline should be ignored. Lastly, volumetric elastic modulus of hydraulic oil is constant.

In these formulas, \( p_1 \) and \( p_2 \) denote rodless and rod chamber pressure of hydraulic cylinder, respectively. \( F \) is load force. \( q_1 \) and \( q_2 \) is flow rate of rodless and rod chamber of hydraulic cylinder, respectively. \( p_L \) is loading pressure.

The flow rate in the rodless and rod chamber of a hydraulic cylinder can be expressed as Formula 4 and 5 according to the hydrodynamic equation.

\[
q_1 = c_d \omega x_v \frac{z}{\rho} \left( p_s - p_1 \right) \quad (4)
\]

\[
q_2 = c_d \omega x_v \frac{z}{\rho} p_2 \quad (5)
\]

In these two formulas, \( c_d \) is flow rate constant. \( \omega \) is natural frequency. \( x_v \) indicates spool displacement of proportional valve. \( \rho \) denotes density of hydraulic oil and \( p_s \) denotes operating pressure of hydraulic pump.

When the hydraulic cylinder moves at a constant speed, Formula 6 and 7 can be deduced.

\[
q_1 = \frac{A_1}{A_2} q_2 \quad (6)
\]

\[
p_s = \frac{p_2}{\varepsilon} + p_1 \quad (7)
\]

Formulas 8 and 9 can be obtained by solving Formulas 3 and 7. And the loading flow rate \( q_L \) can be defined as Formula 10

\[
p_1 = \frac{p_L + n^3 p_s}{1 + n^3} \quad (8)
\]

\[
p_2 = \frac{p_L + n^3 p_L}{1 + n^3} \quad (9)
\]

\[
q_L = c_d \omega x_v \frac{z}{\rho(1 + n^3)} \quad (10)
\]

Formula 11 and 12 can be calculated based on the flow rate continuity equation.

\[
q_1 = A_1 \frac{dy}{dt} + \frac{v_{e1}}{\rho_e} \frac{dp_1}{dt} + (C_{ec} + C_{ic}) p_1 - C_{ic} p_2 \quad (11)
\]

\[
q_2 = A_2 \frac{dy}{dt} - \frac{v_{e2}}{\rho_e} \frac{dp_2}{dt} - (C_{ec} + C_{ic}) p_2 + C_{ic} p_1 \quad (12)
\]
In these two formulas, $y$ denotes hydraulic cylinder displacement. $V_{01}$ and $V_{02}$ are rodless chamber volume and rod chamber volume at the initial moment. $\beta_e$ denotes bulk modulus of elasticity. $C_{ec}$ and $C_{ic}$ are external and internal leakage coefficients of hydraulic cylinder.

Further, Formula 13 can be solved according to force balance equation of hydraulic cylinder.

$$A_1p_1 - A_2p_2 = m\frac{dy}{dt^2} + B_e \frac{dy}{dt} + Ky + F_r$$  \hspace{1cm} (13)

$m$ is loading mass. $B_e$ is damping coefficient. $K$ is stiffness coefficient. $F_r$ is interference force.

Formulas 10 to 13 are performed by Laplace transform. In this way, the transfer function (formula 14) from the spool displacement of the proportional valve to the piston displacement of the hydraulic cylinder can be calculated.

$$G(s) = \frac{k_h\omega_h^2}{s^2 + 2\xi_h\omega_h s + \omega_h^2}$$  \hspace{1cm} (14)

In the transfer function,

$$\omega_h = \sqrt{\frac{4\beta_e A_1^2}{MV_t}}$$  \hspace{1cm} (15)

$$\xi_h = k_{fp}\sqrt{\frac{MV_t}{A_1}}$$  \hspace{1cm} (16)

$G(s)$ denotes the calculated transfer function. $k_h$ denotes total coefficient. $\omega_h$ and $\xi_h$ denote natural frequency and damping ratio of the whole system. $M$ and $V_t$ are equivalent mass and volume of the whole system. $k_{fp}$ is gain coefficient of flow rate and pressure.

### TABLE 1. Control variables configuration

| name of the variables                  | type               |
|----------------------------------------|--------------------|
| piston displacement of hydraulic cylinder | observed state variable |
| rodless chamber pressure of hydraulic cylinder | free state variable |
| rod chamber pressure of hydraulic cylinder | free state variable |
| spool displacement of proportional valve | free state variable |
| spool speed of proportional valve       | free state variable |
| input signal of the proportional valve  | control variable   |

### TABLE 2. Parameter configuration of hydraulic system

| component          | parameter             | value   |
|--------------------|-----------------------|---------|
| hydraulic oil      | density               | 850kg/m³ |
| hydraulic cylinder | piston diameter        | 32mm    |
|                    | piston rod diameter   | 16mm    |
| proportional valve | rated current         | 630mA   |
|                    | rated flow rate       | 40L/min |

The AMESim simulation was conducted and the calculation step was 2ms. Then the Jacobi data file generated by AMESim calculation was imported into SIMULINK. A program of model identification was written in SIMULINK to obtain the standard form of transfer function from the spool displacement of the proportional valve to the piston displacement of the hydraulic cylinder. The calculation result is presented in Formula 17.

$$G(s) = \frac{117.7}{s^2 + 4.041s + 25720}$$  \hspace{1cm} (17)

It is obvious that the total coefficient is 0.0046, and the natural frequency and damping ratio are 160.3748rad/s and 0.012599 by calculating the corresponding variables in Formula 14 and 17. The step response results calculated from the transfer function and the AMESim simulation model are displayed in Fig.4.
A. DESIGN OF SMC

The essence of SMC is a kind of special nonlinear control. In SMC, the system structure can be changed continuously in the dynamic process according to the current state parameters, such as error and each order derivative. As a result, system parameters are forced to move according to a predetermined state trajectory. The trajectory of SMC can be mainly divided into two aspects. Firstly, the system parameters move from any initial state to the designed state parameters to keep the system state near the sliding surface all the time. The latter part is used for bring the system closer to the sliding surface to weaken chattering phenomenon.

Formula 25 can be calculated by deriving Formula 22.

\[ \dot{s} = c_1 \dot{e}_1 + c_2 \dot{e}_2 + \dot{e}_3 \]  

Formula 23 is brought into Formula 25, and the equivalent control part can be expressed as Formula 26 when \( \dot{s} \) is equal to 0.

\[ u_{eq} = (2c_1 \xi_h \omega_h - c_2) e_1 + (1 - \omega_h^2) e_2 \]  

The exponential approach law is applied for the switching control part. As results, the switching control part can be calculated as Formula 27.

\[ u_{sw} = -\varepsilon \text{sgn}(s) - ks \]  

In this formula, \( \varepsilon \) and \( k \) are constant parameters which are greater than 0.

To sum up, the control rate of the designed SMC can be expressed as Formula 29.

\[ u_{sm} = (2c_1 \xi_h \omega_h - c_2) e_1 + (1 - \omega_h^2) e_2 - \varepsilon \text{sgn}(s) - ks \]

B. DESIGN OF FAPID CONTROL ALGORITHM

FAPID algorithm is mainly composed of a fuzzy controller and a PID controller. The fuzzy controller takes error and error variation rate as input parameters, and the proportional, integral and differential parameters of the PID controller are adaptively adjusted by using designed fuzzy rules to maintain required dynamic and static characteristics. Compared with PID control, FAPID has better control characteristics for large time-varying and nonlinear controlled systems.

The FAPID controller should be established according to the following steps. Firstly, fuzzy control rules are established. Secondly, error and error variation rate are fuzzified by using fuzzy rules to obtain corresponding membership function. Lastly, according to the input parameters, defuzzification calculation is conducted to get output data. The control rate of FAPID can be expressed as Formula 30.
\[ U_f = (K_p + \Delta K_p)e(k) + (K_i + \Delta K_i) \sum_1^k e(k) + (K_d + \Delta K_d)(e(k) - e(k-1)) \] 

The control rate of FAPID, \( U_f \), is expressed as a function of the error signal \( e(k) \) and its past values. \( K_p, K_i \), and \( K_d \) are proportional, integral, and differential coefficients, respectively. \( \Delta K_p, \Delta K_i \), and \( \Delta K_d \) are the proportional, integral, and differential coefficient increments calculated by fuzzy rules. \( e(k) \) is the error of the k moment and \( e(k-1) \) is the error of the k-1 moment.

### C. CO-SIMULATION OF AMESIM AND SIMULINK

The system model of an asymmetrical hydraulic cylinder controlled by a symmetrical four-way valve (Fig. 5) was established in AMESim. In addition, the joint simulation interface needs to be created in AMESim. The load impact disturbance of the hydraulic cylinder was simulated by using a random signal. One of the random signals is depicted in Fig. 6. It is apparent from this figure that the variation frequency of the load is 4Hz. Then, the simulation models of SMC and FAPID were established in SIMULINK which are displayed in Fig. 7 in detail. The two designed control rates were written as S functions according to Formulas 29 and 30.

#### TABLE 3. Fuzzy rules

| \( e_c \) | \( e\) | NB       | NM       | NS       | ZO       | PS       | PM       | PB       |
|--------|--------|----------|----------|----------|----------|----------|----------|----------|
| NB     | NB     | PB/NB/PS | PB/NB/NS | PM/NM/NS | PM/NM/NS | PS/NS/NB | ZO/ZO/PM | ZO/ZO/PS |
| NM     | NB     | PB/NB/PS | PB/NB/NS | PM/NM/NS | PM/NM/NS | PS/NS/NM | ZO/ZO/NS | ZO/ZO/PM |
| NS     | PM/NM/NS| PM/NM/NS| PM/NM/NS| PM/NM/NS| PS/NS/NS| ZO/ZO/NS| NS/PS/NS| NM/PM/NS|
| ZO     | PM/NM/NS| PM/NM/NS| PM/NM/NS| PM/NM/NS| PS/NS/NS| ZO/ZO/NS| NS/PS/NS| NM/PM/NS|
| PS     | PS/NM/NS| PS/NM/NS| PS/NM/NS| PS/NM/NS| PS/NS/NS| ZO/ZO/NS| NS/PS/NS| NM/PM/NS|
| PM     | PS/PB/NS| PS/PB/NS| PS/PB/NS| PS/PB/NS| PS/PB/NS| PS/PB/NS| PS/PB/NS| NS/PB/NS|
| PB     | ZO/ZO/NS| ZO/ZO/NS| ZO/ZO/NS| ZO/ZO/NS| ZO/ZO/NS| ZO/ZO/NS| ZO/ZO/NS| ZO/ZO/NS|

(Note: \( e \) is the error. \( e_c \) is the change rate. For the A/B/C form, A denotes fuzzy rule of proportional coefficient increment. B denotes fuzzy rule of integral coefficient increment. C denotes fuzzy rule of differential coefficient increment)

#### FIGURE 5. AMESim model

#### FIGURE 6. Load disturbance curve

#### FIGURE 7. Control model established in SIMULINK

In simulation, a sinusoidal displacement signal with an amplitude of 150 mm was taken as the desired displacement of the hydraulic cylinder. The PID control algorithm was added as a contrast to better display the control characteristics. The simulation time is 10s and the calculation step is 0.01s. As results, the displacement curves of the hydraulic cylinder with three control modes are shown in Fig. 8, and the related displacement error curves are presented in Fig. 9.
In order to measure the rapidity and accuracy of the different control algorithms, three evaluation indexes with rise time, maximum overshoot and average error were adopted. Rise time is defined as the time it takes to rise from 0 to 90% of the expected value. The maximum overshoot is defined as the maximum deviation between the desired displacement and the actual displacement. The absolute value of average error (AVAE) can be calculated according to Formula 31.

\[ e_{\text{avr}} = \frac{\sum_{k=1}^{1000} |e_k - e_{sk}|}{m} \]  

(31)

where \( e_{avr} \) denotes AVAE, \( e_k \) and \( e_{sk} \) represent expected and actual displacement, respectively, \( m \) is the total number of calculation points. There are 1000 calculation points in this simulation.

These two figures are quite revealing in several ways. In the first place, the three control algorithms can track the expected displacement effectively. The rise time of PID, SMC and FAPID are 0.52s, 0.18s and 0.29s, respectively. The response speed of SMC is the fastest, but that of PID is the slowest. In the other place, the maximum overshoot of PID, SMC and FAPID are 22.12mm, 16.71mm and 16.80mm. The latter two algorithms have better overshoot control accuracy. In addition, the AVAE parameter of PID, SMC and FAPID are 14.69mm, 8.50mm and 3.66mm. These data recognize that the designed FAPID algorithm has the best control accuracy and stability. It is obvious from Fig.9 that there is large buffeting in the displacement curve for SMC algorithm, which is the common disadvantage of SMC algorithm. This is mainly because the SMC can only approximately keep the controlled parameters near the sliding surface, but it is difficult to keep them on the sliding surface all the time. Based on the above analysis, it can be drawn a conclusion that the FAPID algorithm has the best control characteristics and the SMC is better than PID. Therefore, FAPID was adopted in this study.

V. EXPERIMENT

A new experimental platform, including a HIS (Fig.10) and some hydraulic control valve groups (Fig.11) was researched and developed to verify the correctness of above theoretical analysis results. The composition of the experimental platform is the same as that shown in Fig.1. The executive components are 4 hydraulic cylinders. The upper and lower ends of each hydraulic cylinder are articulated. A center universal hinge is arranged in the middle of the four hydraulic cylinders. Theoretically, the frame can rotate around any axis, but the rotation angle is limited by the hydraulic cylinder stroke and the articulation mechanism.

Additionally, a displacement sensor with the measuring range of 1000mm was installed on each hydraulic cylinder to obtain cylinder stroke information dynamically. The electrical signal generated by the displacement sensor was transmitted to the isolation transmitter, which divides the original signal into two channels. One was transmitted to the main controller and the other one was transmitted to the data acquisition card. The sampling frequency of data acquisition card was set to 100Hz. It should be noted that the two new signals are identical to the original signal. Four proportional relief valves, whose opening pressure is proportional to the
external control electrical signal were used to load hydraulic cylinders. In experimental process, one hydraulic cylinder was loaded with a constant load, and the other cylinders was subjected to an increased load impact disturbance signal on the basis of the constant load. The change of load impact disturbance signal was consistent with that of Fig.6 and the expected displacement signal was consistent with that in simulation. In order to verify the effect of FAPID algorithm, PID algorithm was introduced as a comparison. The measuring displacement curves and displacement error curves of two control algorithms are shown in Fig.12 and 13.

As can be recognized from the figures that both PID and FAPID algorithms can track the expected displacement curve accurately when the external load of hydraulic cylinder is constant. The rise time of FAPID and PID algorithms are 0.26s and 0.32s, respectively. The response speed of FAPID algorithm is slightly better than that of the PID algorithm. The maximum overshoot of FAPID and PID algorithms are 11.33mm and 17.21s and the AVAE of FAPID and PID algorithms are 4.90mm and 6.53mm. When the external load of the hydraulic cylinder has impacted disturbance, the gap of control effect will be further enlarged. The rise time of FAPID and PID algorithms are 0.31s and 0.58s, respectively. It is apparent from this data that the response speed of PID control is obviously slower than that of FAPID. In addition, the maximum overshoot of FAPID and PID algorithms are 14.75mm and 24.58mm and the AVAE of FAPID and PID algorithms are 5.23mm and 13.99mm. Under the condition of load disturbance, the control characteristic of FAPID is obviously better than that of PID control.

VI. CONCLUSIONS

This study set out to design an intelligent FAPID algorithm to improve the position control accuracy of HIS under the operating condition of external load impact. A series of methods such as theoretical design, model identification, simulation optimization and comparative test were adopted. The position control accuracy of hydraulic cylinder based on PID, SMC and FAPID algorithms was analyzed emphatically. The study results are quite revealing in several ways. It can be known from co-simulation of AMESim and SIMULINK that the transfer function from the spool displacement of the proportional valve to the piston displacement of the hydraulic cylinder obtained by model identification has high accuracy. The designed FAPID algorithm takes response speed, control accuracy and stability into account, and its control performance is better than that of SMC and PID algorithms. In addition, the experimental data indicate that FAPID algorithm has a small optimization improvement compared with PID, when the external load is constant. The rise time is decreased by 0.06s and the AVAE is reduced by 24.96%. The advantage of FAPID algorithm is more obvious, when there are impact disturbances in external load. The rising time is decreased by
0.27s and the AVAE is reduced by 62.62%. This proves that FAPID algorithm has better anti-disturbance performance.

In general, there are two innovative points in this study. The hydraulic system pressure loss is large and the efficiency is reduced, if the hydraulic synchronization element is employed to shunt the flow rate forcibly. This problem is avoided by using the FAPID algorithm to improve the position control accuracy of the hydraulic cylinder. In addition, the preferred FAPID algorithm has ideal anti-interference ability. This means that the method designed in this paper can obtain better synchronization effect when the external load is disturbed.

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