Modeling and Research of Fuzzy PID Control Strategy for Hydraulic Rock Drill Propulsion System

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Abstract. According to the coupling characteristic of propulsion system and impact system of hydraulic rock drill, deduced the calculation formula of optimal axial thrust for the propulsion system of hydraulic rock drill. On the basics of the calculation formula before, the paper constructed compound driven propulsion system of hydraulic rock drill based on electro-hydraulic proportional pressure-reducing valve and high speed on-off valve, and established its dynamics model. Finally, the paper considered optimal axial thrust as the goal, and adopted the fuzzy PID control strategy to realize on-line, intelligent control to axial thrust of the propulsion system of hydraulic rock drill. Through mathematical modelling and simulation study show that fuzzy PID control method has its advantages as the fast response follow by target, high control precision, and small fluctuation, and provided theory reference and technology method for intelligent control of the propulsion system of hydraulic rock drill.

1. Introduction

The propulsion system of hydraulic rock drill is a device to drive the rock drill body forward and make it well contact with the rock, so the rock drill can achieve good crushing efficiency in the process of drilling. The axial thrust output by hydraulic cylinder is closely related to the working parameters and working medium output by impactor. In order to improve the efficiency of rock drilling and crushing, the working parameters (impact energy and working frequency) of hydraulic rock drill will be adjusted according to the working medium impacted with different characteristics. At the same time, in order to prevent the rock drill body from retreating and affecting the effective shock wave (or shock energy) transmission between drill head and rock, the axial thrust applied to the rock drill must be effectively controlled. Therefore, the coupling characteristics of propulsion system and the impact system of hydraulic rock drill are fully considered to realize the adaptive control of axial thrust of the hydraulic rock drill propulsion system. This can effectively improve the efficiency of rock drill and reduce energy consumption\textsuperscript{[1]}. Many scholars have applied electro-hydraulic control technology to hydraulic propulsion device and achieved good results. Liu Zhong and Wu Jinsong proposed using high speed on-off valve to realize adaptive control of axial thrust of hydraulic rock drill\textsuperscript{[2]}. Guo Yan studied the rotary system of submersible drilling, and proposed to realize the adaptive output of axial thrust by controlling the
The electro-hydraulic control technology of shield tunneling system is studied by Shihu. The proportional relief valve and proportional speed control valve are used to control the propulsion pressure and speed. Based on the previous research results, proposed in this paper, with straight moving type electro-hydraulic proportional pressure reducing valve to the adjustment of the hydraulic cylinder propulsion, at a high speed on-off valve to realize advance speed regulation of hydraulic rock drill. The fuzzy PID control strategy is applied to the control of the thrust of the propulsion system of rock drill, so as to realize the adaptive control of the output parameters of propulsion system of rock drill under different drilling conditions.

2. Determination of Optimum Axial Thrust of Hydraulic Rock Drill Propulsion System

The axial thrust is the key parameter of hydraulic rock drill propulsion system. When the axial thrust is less than the recoil force, the rock drill will move backward, with rock drill ineffectively to drill into and break the rock. When the axial thrust is greater than the rear seat force (the difference is large), the focus point of the drill head will be deviated with producing slip and increasing the resistance of rotary drill, which is bound to reduce the working efficiency of rock drill and the life of drill head. Since the shaft thrust of the rock drill has a serious impact on the working performance of the hydraulic rock drill, and at the same time, it will seriously accelerate the wear of the rotary drill blade, it is necessary to determine the optimal axial thrust under specific working conditions and control the adaptive output of the axial thrust.

The following equation can be obtained by impulse theorem in the acceleration stage of rock drill:

\[
(m_i v_{i0} + m_{v_i}) - (m_i v_{i0} + m_{v_i}) = F_{T_{min}} \Delta T
\]

where \( m \) and \( m_i \) are the mass of the rock drill body and piston respectively (Kg); \( v_0 \) and \( v_f \) are the initial speed and final speed of the rock drill body respectively (m/s); \( v_{i0} \) and \( v_{i} \) are the initial impact velocity and the final impact velocity of the piston, respectively (m/s); \( F_{T_{min}} \) is the minimum axial thrust on hydraulic impactor (N); \( \Delta T \) is the piston stroke acceleration time (s), \( \Delta T = \alpha / f \), where \( \alpha \) is an abstract design variable, \( f \) is the impact frequency of piston (Hz).

Assuming that the rock drill body remains static during the stroke phase, \( v_0 = v_f = 0 \). So there are:

\[
m_i v_{i} = F_{T_{min}} \Delta T = F_{T_{min}} \frac{\alpha}{f}
\]

The impact energy of the piston impact bit is as follows:

\[
W = \frac{1}{2} m_i v_{i}^2
\]

The simultaneous equations (2), (3) can be calculated as follows:

\[
F_{T_{min}} = \frac{f}{\alpha} \sqrt{2m_i W}
\]

In the process of rock drilling and crushing, the axial thrust is also related to the friction force and the nature of the working medium when the body moves forward. For example, the rock with high hardness requires large axial thrust, and the rock with low hardness requires small axial thrust. According to the engineering practice experience, in order to ensure low energy consumption and high drilling speed of rock drill, the optimal axial thrust calculation formula is as follows:

\[
F_T = \left(\frac{\gamma - 8}{100} + 1\right) \left(\frac{f}{\alpha} \sqrt{2m_i W} + F_{\mu}\right)
\]

where \( \gamma \) is the rock P-coefficient, \( F_{\mu} \) is the friction force in the movement of the rock drill body (N).

3. Modeling of Propulsion System of Hydraulic Rock Drill

3.1. The axial thrust control system scheme of rock drill
The rock machine propulsion system is composed of hydraulic propulsion cylinder, high-speed switching valve and direct-acting electro-hydraulic proportional pressure reducing valve. When the rock drill is in a specific state of rock drilling, the controller controls the load pressure by inputting a specific current to the direct-acting electro-hydraulic proportional pressure reducing valve 4. By inputting a specific current pulse width modulation signal to the high-speed switch valve, the flow rate of the system is controlled. The output flow rate is proportional to the duty cycle of the pulse width modulation signal. At the same time, the forward and backward motion of the hydraulic propulsion cylinder can be realized by controlling the high-speed switch valve 2 and 3 separately. Assuming that the rock has viscoelastic properties, a spring damping model is constructed, so the model of shaft thrust control system of rock drill is shown in Figure 1[6-8].

1-Hydraulic propulsion cylinder 2  3-High-speed on-off valve 4-Direct operated electro-hydraulic proportional pressure reducing valve 5-Hydraulic pump 6-Overflow valve

Figure 1 Model diagram of shaft thrust control system for rock drill

3.2. Mathematical model of shaft thrust control system for rock drill

In the process of percussive rock drilling, facing different rock drilling conditions, fully consider the coupling characteristics of the hydraulic propulsion system, the impactor parameter output, and the rock state to establish a mathematical model for rock drill shaft thrust control [9-13].

3.2.1. Mathematical Model of Direct Electro-hydraulic Proportional Valve

The force balance equation of the spool is as follows:

\[ F_i - p_i A = m_x \frac{d^2x}{dt^2} + (B_x + B) \frac{dx}{dt} + (K_x + K) x \]

(6)

The pressure flow equation of pressure reducing valve port after linear transformation is as follows:

\[ Q = \lambda_x x - \lambda_p p_i \]

(7)

The flow continuity equation at zero load is as follows:

\[ Q = \frac{V}{\beta} \frac{dp_i}{dt} - A_i \frac{dx}{dt} \]

(8)

where \( F_i \) is the electromagnetic force of proportional electromagnet (N); \( F_i = K_i i \), \( K_i \) is the current-magnetic coefficient of proportional electromagnet (N/A), \( i \) is the input current of electromagnet (A); \( p_i \) is proportional pressure relief valve outlet pressure (Pa); \( A_i \) is the pressure feedback area of pressure relief valve outlet (m²); \( m_x \) is the quality of pressure valve spool (Kg); \( x \)
is spool displacement of pressure relief valve (m); $B_v$ and $B_t$ are viscous damping coefficient and transient hydrodynamic damping coefficient of spool, respectively (N·s/m); $K_i$ and $K_i$ are the reset spring stiffness and hydraulic spring stiffness of the proportional pressure reducing valve, respectively (N/m); $\lambda_q$ is the flow gain coefficient (m³/s), $\lambda_q = C_i \pi D_i \sqrt{2(p-p_{at})/\rho}$, where $C_i$ is the flow coefficient, $D_i$ is the aperture of the valve seat of the proportional pressure reducing valve (m); $p$ and $p_{at}$ are the stable values of inlet pressure and outlet pressure of the pressure reducing valve, respectively (Pa), $\rho$ is the density of the oil (Kg/m³); $\lambda_p$ is the pressure gain coefficient $(m^3/(s\cdot Pa))$, $\lambda_p = 0.5x_dC_i\pi D_i / \sqrt{\rho(p-p_{at})}/2$, where $x_d$ is the stable value of spool displacement (mm); $V$ is total volume of outlet controlled chamber oil (m³); $\beta_p$ is effective bulk modulus of elasticity (N/m²).

The transfer function between pressure output and current of proportional pressure reducing valve can be obtained by simultaneous equations (6), (7) and (8).

$$P(s) = \frac{K_i}{s^2 + bs + cs + d}$$

where $a = \frac{V}{\beta_v}m_2$; $b = (B_v + B_t)\frac{V}{\beta_v} + m_2\lambda_p$; $c = (B_v + B_t)\lambda_p + (K_i + K_i)\frac{V}{\beta_v} + A \lambda_p$; $d = A_2\lambda_p + (K_i + K_i)\lambda_p$.

### 3.2.2. Mathematical Model of High Speed Switch Valve

The flow pressure equation of high-speed switching valve is as follows:

$$Q = \lambda C_2 \pi D_2 h_o \frac{c}{R} \frac{2(p_1 - p_2)}{\rho}$$

where $Q$ the actual flow rate of high-speed switching valve (L/min), $\lambda$ is the duty cycle of electromagnetic pulse width modulation input; $C_2$ is the flow coefficient of high speed switch valve; $D_2$ is the aperture of high-speed switch valve seat (m); $c$ is the opening length of high-speed switch valve (mm); $R$ is the steel ball radius of high speed switch valve (mm); $h_o = \sqrt{R^2 + (D_2/2)^2}$; $p_2$ is the outlet pressure of high-speed switch valve (Pa).

The results of Laplace transform for formula (10) are as follows:

$$P_2(s) = P_1(s) - \frac{\gamma Q^2}{2\gamma^2}$$

where $\gamma = \lambda C_2 \pi D_2 h_o \frac{c}{R}$.

### 3.2.3. Mathematical model of hydraulic cylinder

The leakage of the hydraulic cylinder, the viscous resistance and axial deformation of the piston are ignored, and the pressure of the rod cavity is assumed to be 0. The force analysis of the axial thrust valve control cylinder model of the rock drill is carried out. According to the force balance equation of the hydraulic cylinder, the axial thrust of the rock drill is as follows:

$$F = p_2 A_2 = m \frac{d^2 y}{dt^2} + \eta \frac{dy}{dt} + K_2 y + F_e$$

where $A_2$ is the effective area of hydraulic cylinder rod-less cavity (m²), $\eta$ is rock penetration viscosity coefficient (N·s/m), $K_2$ is rock stiffness (N/m), $F_e$ is the external force on the piston rod (N).

The results of Laplace transform for formula (12) are as follows:

$$P_2(s)A_2 = m_2 s^2 Y(s) + \eta s Y(s) + K_2 Y(s) + F_e(s)$$

### 3.3. Mathematical Model of Feedback System


The system measures the pressure of the front and rear chambers of the hydraulic cylinder and feedbacks its pressure relationship to the controller. Then, the current of the proportional flow valve is compared with that of the set proportional flow valve to control the flow output of the proportional pressure reducing valve. According to the characteristics of pressure sensor and controller current output, the mathematical expression of the feedback system is as follows:

\[ i = Hp \]  

where \( H \) is the current - pressure gain (A/MPa).

### 3.4. Rock drill propulsion control system model

The connecting the above mathematical models of proportional pressure reducing valve, high-speed switching valve and hydraulic cylinder in the form of data flow, the mathematical model of shaft thrust control system of rock drill can be obtained, as shown in Figure 2.

#### 4. Simulation Analysis

PID control has been widely used in industrial control systems because of its simple structure and convenient operation. The shaft thrust control system of hydraulic rock drill has the characteristics of nonlinear, large disturbance and time-varying. Therefore, in this paper, fuzzy PID control method is used to take full advantages of fuzzy control and PID control, in order to quickly follow the shaft thrust response, which can effectively improve the robustness and control accuracy of the system.

#### 4.1. Fuzzy PID controller design

The fuzzy PID controller adjusts the \( K_p, K_i \) and \( K_d \) parameters online to meet the precise control of the target object. The structure of the fuzzy PID controller is shown in Figure 3.

The error \( e \) and the error change rate \( ec \) of the controlled variable are used as the input of the controller, and \( \Delta K_p, \Delta K_i \) and \( \Delta K_d \) are used as the output of the fuzzy controller to realize the tuning of the parameters of \( K_p, K_i \) and \( K_d \)\cite{14-15}. The parameters of \( K_p, K_i \) and \( K_d \) after setting are as follows:

\[ K_p = K_{p0} + \Delta K_p \]  

(15)

\[ K_i = K_{i0} + \Delta K_i \]  

(16)

\[ K_d = K_{d0} + \Delta K_d \]  

(17)

where, \( K_p \) is the proportional coefficient, \( K_i \) is the integral coefficient, \( K_d \) is the differential coefficient, \( K_{p0} \) is the initial proportional coefficient, \( K_{i0} \) is the initial integral coefficient, \( K_{d0} \) is the initial differential
coefficient, $\Delta K_p$ is the variation of the proportional coefficient, $\Delta K_i$ is the variation of integral coefficient; $\Delta K_d$ is the variation of differential coefficient.

4.1.1. Determination of Language Variable and Membership Function of Fuzzy PID Controller
According to the working condition of the rock drill, the domain of the hydraulic cylinder load pressure error $e$ and error change rate $ec$ is set to [-6,6], the fuzzy subset of the linguistic value is {negative large, negative medium, negative small, zero, positive small, positive medium, positive large}, abbreviated as {NB, NM, NS, ZO, PS, PM, PB}. The domain of $\Delta K_p$, $\Delta K_i$ and $\Delta K_d$ is set to [-6,6], the fuzzy subset of the linguistic value is {negative large, negative medium, negative small, zero, positive small, positive medium, positive large}, abbreviated as {NB, NM, NS, ZO, PS, PM, PB}. In order to make the system have good robustness and control performance, the membership functions of fuzzy subsets NB and PB adopt Gaussian function, and the membership functions of other fuzzy subsets adopt uniform distribution triangle function with high resolution.

4.1.2. Control rules of fuzzy PID controller
In order to achieve precise control of PID parameters ($K_p$, $K_i$ and $K_d$), the relationship between input variables ($e$ and $ec$) and output variables ($\Delta K_p$, $\Delta K_i$ and $\Delta K_d$) must be accurately grasped. According to the PID parameter tuning principle [8] and the experts' long-term accumulated practical knowledge and experience, a fuzzy control rule library for setting $\Delta K_p$, $\Delta K_i$ and $\Delta K_d$ is established, as shown in Table 1.

| $ec$     | NB    | NM    | NS    | ZO    | PS    | PM    | PB    |
|----------|-------|-------|-------|-------|-------|-------|-------|
| NB       | PB/NB/PS | NB/NB/NS | PM/NM/NB | PM/NS/NB | PS/NS/NB | ZO/ZO/NM | NS/ZO/NS |
| NM       | PB/NB/PS | NB/NB/NS | PM/NM/PM | PS/NS/PM | ZO/ZO/NS | ZO/ZO/NM | PS/NM/PS |
| NS       | PM/NM/ZO | PM/NM/NS | PS/NS/NS | PS/NS/ZO | ZO/ZO/NS | ZO/ZO/NS | PS/NS/PS |
| ZO       | PM/NM/ZO | PM/NM/NS | ZO/NS/NS | ZO/NS/ZO | ZO/NS/NS | ZO/NS/PS | PS/NS/PS |
| ZO       | PS/NM/ZO | ZO/ZO/NS | ZO/ZO/NS | ZO/ZO/NS | ZO/ZO/NS | ZO/ZO/NS | NS/NS/NS |
| PM       | ZO/NS/PM | ZO/NS/NS | ZO/NS/NS | ZO/NS/NS | ZO/NS/NS | ZO/NS/NS | ZO/NS/NS |
| PB       | ZO/NS/PS | ZO/NS/PS | ZO/NS/PS | ZO/NS/PS | ZO/NS/PS | ZO/NS/PS | ZO/NS/PS |

4.2. Parameter setting of propulsion system
Taking a hydraulic rock drill as the research object, the basic parameters of its propulsion system are shown in Table 2.

| Parameter                                      | Value  |
|------------------------------------------------|--------|
| Quality of piston rod $m_1$ (Kg)               | 6.6    |
| Valve core quality of pressure reducing valve $m_2$ (Kg) | 0.024  |
| Rock drilling viscosity coefficient $\eta$ (N·s/m) | 5976   |
| Reset spring stiffness of pressure reducing valve $K_1$ (N/m) | 2750   |
| Hydraulic spring stiffness $K_1$ (N/m)         | $1.03\times10^5$ |
| Rock rigidity $K_2$ (N/m)                      | $2\times10^7$ |
| Flow coefficient of pressure reducing valve $C_1$ (m³/s) | 0.62   |
| Flow coefficient of high-speed on-off valves $C_2$ (m³/(s·Pa)) | 0.70   |
| Current – magnetic force coefficient of pressure reducing valve $K_3$ (N/A) | 90     |
| Viscous damping coefficient of pressure reducing valve core $B_3$ (N·s/m) | 7.3    |
| Transient hydrodynamic damping coefficient of pressure reducing valve core $B_4$ (N·s/m) | 21.17  |
| Effective volume elastic modulus of oil $\beta_e$ (Pa) | $1.2\times10^9$ |
| Pressure feedback area of pressure reducing valve outlet $A_1$ (m²) | $7.065\times10^9$ |
| Oil volume of controlled chamber at outlet of pressure reducing valve $V$ (m³) | $2.5\times10^4$ |
Pressure reducing valve seat aperture $D_1$ (m) \[ 9 \times 10^{-3} \]
Seat aperture of high-speed on-off valve $D_2$ (m) \[ 6 \times 10^{-3} \]
Steel Ball Radius of high-speed on-off valve $R$ (m) \[ 4 \times 10^{-3} \]
Oil density $\rho$ (Kg/m$^3$) \[ 860 \]

The simulation model of shaft thrust control based on fuzzy PID control can be obtained by the mathematical model of shaft thrust control system and the parameter value of table 2, as shown in Figure 4.

4.3. System Simulation and Analysis
The initial proportional coefficient $K_{p0}=1.4$, the initial integral coefficient $K_{i0}=0.2$, and the initial differential coefficient $K_{d0}=0.0056$ are set. According to the actual range of error, $ec$, $K_p$, $K_i$ and $K_d$, the quantitative factors of $e$ and $ec$ are $K_e=125$ and $K_{ec}=1000$, respectively. The proportional factors of $\Delta K_p$, $\Delta K_i$ and $\Delta K_d$ are $K_{\Delta p}=0.45$, $K_{\Delta i}=0.02$ and $K_{\Delta d}=0.0025$, respectively. The sampling time is set to $10^{-4}$s, and the step signal is used as input to obtain the pressure output curve and displacement curve of the hydraulic propulsion cylinder, as shown in Figure 5 and Figure 6.
The curves 1, 2 and 3 in Figure 5 are the pressure output curves of hydraulic propulsion cylinder when using conventional closed-loop control, PID control and fuzzy PID control respectively. When the conventional closed-loop control is adopted, the overshoot of pressure is 11.1 % and the adjustment time is 0.0146 s; for PID control, the pressure overshoot is 0.53 %, the adjustment time is 0.0067 s; for fuzzy PID control, the overshoot of pressure is 0.13 %, and the adjustment time is 0.0065 s. It can be seen that the oscillation and fluctuation that caused by the adjustment of axial thrust by fuzzy PID control method are significantly reduced, taking the shortest time to achieve a stable state and obtaining higher control accuracy.

The curves 1, 2 and 3 in Figure 6 are the pressure output curves of hydraulic propulsion cylinder when using conventional closed-loop control, PID control and fuzzy PID control respectively. When PID control and fuzzy PID control are adopted, the displacement output of hydraulic propulsion cylinder is stable. If the amplification curves are 2 and 3, the effect of fuzzy PID control is more stable. Therefore, the fuzzy PID control can reduce the vibration of the hydraulic propulsion cylinder, improve the good contact between the rock drill and the rock, and further improve the crushing efficiency of the rock drill. In the figure, the transient pulse interference is input to the system at 0.02 s, which shows that the anti-interference ability of PID control and fuzzy PID control is relatively high.

5. Conclusion
The propulsion system of hydraulic rock drill based on proportional pressure reducing valve and high-speed switching valve control was proposed, and the corresponding mathematical model was established to realize the control of axial thrust of rock drill by proportional pressure reducing valve and the control of flow rate of propulsion system by high-speed switching valve.

The fuzzy PID control method is adopted to adjust the axial thrust of the rock drill propulsion system, which reduces the pressure fluctuation in the process of axial thrust adjustment, and can reach the target state smoothly. Meanwhile, this method also reduces the vibration of the propulsion cylinder, which is conducive to improving the efficiency of rock drilling.

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