STUDY ON THE CONTROL METHOD OF CONSTANT DECELERATION BRAKE FOR MINE HOIST

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This paper first introduces the control method of deceleration in hoist emergency braking, introduces the working principle of constant deceleration braking system, and establishes the mathematical model of every component of braking system. Then the transfer function of the braking system is obtained. The transfer function shows that the braking system is a high order and type 0 system. The control of braking system needs to be corrected. At last, the simulation model is built in Simulink, the method of double PID in series is used. The stability and robustness of the control system are verified by simulation and test.

Keywords: Mine hoist; Braking system; Constant deceleration; Transfer function; Closed-loop control system

1. Introduction

As the key equipment in the mine production process, the hoist constructs the connection between the ground and the underground, and undertakes the important task of lifting coal or ore, lowering materials and elevating personnel or equipment. In case of emergency or accident of hoist, the brake device should quickly and safely brake the brake disc of hoist, so that the hoist can stop quickly (also called emergency brake) to avoid the malignant expansion and spread of the accident [1]. The braking system of hoist plays a very important role in the safe

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operation of hoist. Therefore, scholars and scientific research institutions at home and abroad have carried out positive research on it [2-6].

In the lifting process of mine hoist, the lifting load is large, the speed is fast, and the working condition is complex [1]. In emergency braking, the commonly used braking methods are constant torque braking and constant deceleration braking. During constant torque braking, braking deceleration may exceed the allowable range of deceleration, which may lead to wire rope slipping and reduce the safety performance and service life of the equipment. When braking with constant deceleration, the braking deceleration does not change with the change of load and working condition, and the braking is always according to the preset deceleration value. With the faster response, the better stability, and the higher safety, the constant deceleration braking system is mostly used in braking system of mine hoist. In order to improve the performance of constant deceleration braking safety level of mine hoist, a research method combining theoretical research, calculation simulation and test is proposed is proposed in this paper to study the dynamic characteristics and braking performance of the hoist constant deceleration control system, so as to improve its safety level.

Using simulation method to study the dynamic and static characteristics of the system has been widely used. Krus, P. [7] and Qinhe, G. & et al. [8] simulated the dynamic characteristics of the pipeline in the hydraulic system. Zhao, L. [9] established the mathematical model of electro-hydraulic proportional servo variable pump, and simulated it based on MATLAB. Li Juanjuan et al. [10] used AMESim software to simulate and analyze the fault of hydraulic brake system of mine hoist. Yang Erfu et al. [11] studied the failure mode of liquid rocket propulsion system in China by numerical simulation. In this paper, E141A type constant deceleration braking system of JKMD4.5 × 4 type mine hoist is chosen as the research object. Firstly, the working principle of constant deceleration braking system is introduced, then the mathematical model of each component is established, and the transfer function is obtained. Through the analysis of transfer function, double PID control is selected. Finally, the stability and robustness of double PID control are verified by simulink simulation and test.

The rest of this paper is structured as follows: Section 2 introduces the working principle of constant deceleration braking system. Section 3 models the components and obtains the transfer function. Section 4 selects the simulation parameters, then optimizes the two PID parameters, and finally carries out the simulation experiment. Section 5 introduces the test-bed, and the practicability is
verified on the test bed of the ultra-deep well hoist of CITIC HEAVY INDUSTRIES. Section 6 summarizes the advantages and disadvantages of the method, and puts forward the further research ideas.

2. Working Principle of Constant Deceleration Braking System

The schematic diagram of E141A type constant deceleration braking system is shown in Fig. 1.

Before the hoist working, the accumulator is filled firstly. After the pressure of the accumulator reaches the required value, the pressure relay JP1 action, G1,
G2, G3, G4 solenoid directional control valves power on, and the proportional overflow valve is lowered to zero. Subsequently, the mine hoist begin to work normally. In the case of the hoist sudden fault, a safe braking is required. The motor and proportional overflow valve power off, variable pump stop to supply oil, solenoid directional control valves G1, G2 power off, and braking system pressure quickly drop to the pressure set by relief valve 19. Then according to the comparison between the actual deceleration signal and given deceleration signal, the controller send the control signal to electro-hydraulic proportional directional valve, making the valve spool moving right or left, which means the oil discharges to the tank, the system pressure decreasing, or charges by the accumulator, the system pressure increasing. When the control current is zero, the spool of electro-hydraulic proportional directional valve is in the middle position and in a fully closed state which makes the system pressure maintain constant [12,13]. Relying on this control mode keeps the braking deceleration constant until the hoist is fully parked. Then the solenoid valves G3, G4 power off and the hoist is in the state of full locking.

3. Modeling of Emergency braking system

According to the working principle of emergency braking, the control principle block diagram of hoist brake system can be obtained as shown in Fig. 2.

![Control principle block diagram of hoist brake system](image)

**Fig. 2. Control principle block diagram of hoist brake system**

**Proportional directional Valve** is products of ATOS Company, its type is DKZO-TES-PS-071-L5 and Fig. 3 is the structure diagram. The valve is equipped with LVDT position sensor, uses positive cover valve spool, and adjusts the position of valve in closed loop mode.
When the proportional electromagnet works in the linear region, its output force can be approximately expressed as [11]

\[ F_d(t) = K_i i(t) - K_y X_p(t) \]  

(1)

where \( K_i \) is the current-force gain of the proportional electromagnet, \( N/mA \); \( K_y = \frac{\partial F_d}{\partial i} \); \( K_{sy} \) is Proportional electromagnet zeroing spring stiffness, \( N/m \); \( K_y \) is the sum of displacement-force gain and zero-adjusting spring stiffness of proportional electromagnet, \( N/m \), \( K_y = \frac{\partial F_d}{\partial x} + K_{sy} \).

The force balance equation of proportional directional valve spool is:

\[ F_d(t) = m_f \frac{d^2 x_p(t)}{dt^2} + c_f \frac{dx_p(t)}{dt} + K_{hy} P_f X_p(t) \]  

(2)

where \( m_f \) is the mass of proportional directional valve spool, \( kg \); \( x_p(t) \) is displacement of proportional directional valve spool, \( m \); \( c_f \) is dynamic damping coefficient of proportional directional valve spool; \( P_f \) is the oil pressure of proportional directional valve, \( Pa \); and \( K_{hy} \) is the equivalent stiffness of the hydrodynamic force related to \( P_f \), \( N/m \).

The Laplace transformation of Eqs. (1) and (2) are as following:

\[ F_d = K_i I - K_y X \]  

(3)

\[ F_d = m_f s^2 X + c_f sX + K_{hy} P_f X \]  

(4)
From Eqs. (3) and (4), the transfer function block diagram is obtained with the current of the proportional amplifier as the input and the displacement of the proportional directional valve as the output.

\[
\frac{X_v}{I} = \frac{K_d}{s^2 + \frac{2\xi_m s}{\omega_m} + 1}
\]

(5)

where \( K_d \) is current-displacement gain of spring mass system of armature assembly, \( K_d = \frac{K_f}{K_f y P_f K_y} \); \( \omega_m \) is natural frequency of spring mass system of armature assembly, \( \omega_m = \sqrt{\frac{K_f y P_f}{m_f}} \); \( \xi_m \) is dimensionless damping ratio of armature assembly, \( \xi_m = \frac{c_f}{2} \sqrt{\frac{K_f y P_f}{m_f}} \).

According to Fig. 4, the current-displacement transfer function of proportional directional valve is obtained as follows:

\[
\frac{X_v}{I} = \frac{K_d}{s^2 + \frac{2\xi_m s}{\omega_m} + 1}
\]

Linearized flow equation of electromagnetic proportional directional Valve is:

\[
Q_L = K_{qs} x_e - K_v p_l
\]

(6)

where \( Q_L \) is load flow of electromagnetic proportional directional Valve, m³/s; \( x_e \) is displacement of electromagnetic proportional directional valve spool, m; \( K_{qs} \) is low-displacement gain of electromagnetic proportional directional valve, m²/s; \( K_v \) is flow-pressure coefficient of electromagnetic proportional directional valve, m³/(Pa · s); \( p_l \) is Load pressure, Pa.

The force equation on the piston of brake is

\[
A_p p_l = m_i \frac{d^2 x_p}{dt^2} + B_i \frac{dx_p}{dt} + K_m x_p + K_m x_0
\]

(7)

where \( A_p \) is piston effective working area, m²; \( p_l \) is pressure of the hydraulic chamber, Pa; \( m_i \) is mass of working parts driven by Piston (including pistons, brake
pads and connecting bolts), \( N \); \( x_p \) is piston displacement, \( m \); \( x_0 \) is precompression length of spring, \( m \); \( B \) is viscous damping coefficient; \( K_m \) is spring stiffness, \( N/m \).

The flow continuity equation in hydraulic cylinder of brake is:

\[
Q_L = A_p \frac{dx_p}{dt} + C_l x_p + \frac{V_s}{\beta} \frac{dp}{dt}
\]

where \( C_l \) is the leakage coefficient of the piston; \( V_s \) is the volume of oil in the working chamber of the hydraulic cylinder and in the intake line, \( m^2 \); \( \beta_e \) is the volume modulus of oil.

The Laplace transformation of Eqs. (6) (7) and (8) are as following:

\[
Q_L = K_w X_v + K_p P_l
\]

\[
A_p P_l = m_s X_p s^2 + B_s X_p s + K_m X_p
\]

\[
Q_L = A_s X_p + C_p P_l + \frac{V_s}{\beta_e} P_s
\]

From Eqs. (9) (10) and (11), the block diagram of the transfer function of proportional valve controlled brake as shown in Fig. 5 is obtained.

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**Fig. 5** The block diagram of transfer function of proportional directional valve control brake

According to Fig. 5, the transfer function with the displacement of the proportional valve spool as the input and the oil pressure of brake chamber as the output is obtained as follows:

\[
\frac{P_l}{X_v} = \frac{K_m (m_s s^2 + B_s s + K_m)}{(K_m + V_s / 4\beta_e s)(m_s s^2 + B_s s + K_m) + A_p \beta_m s}
\]

\[
= \frac{K_m m_s s^2 + B_m s + K_m}{A_p \beta_m s + \left( \frac{K_m m_s + B_m}{A_p \beta_m} + \frac{4\beta_s}{A_p} \right) s + \left( 1 + \frac{B_m}{A_p \beta_m} + \frac{K_m V_s}{A_p \beta_m} \right) s + \frac{K_m m_s}{A_p \beta_m}}
\]

---
where $K_v$ is total flow-pressure coefficient, $K_v = K_i + C_i$.

Due to $\frac{B K_v}{A_p^2} \ll 1$, $\frac{K_m}{K_h} \ll 1$ and $\left(\frac{K_v \sqrt{K_m m_i}}{A_p^2}\right) \ll 1$, Eq. (12) can be simplified as:

$$\frac{P}{K_v K_m} = \frac{K_v X_v \left(m s^2 + B s + K_m\right)}{\left(\frac{s}{\omega_i} + 1\right) s^2 + 2 \xi \omega_i + 1}$$

(13)

where $\omega_i$ is inertial element corner frequency, $\omega_i = \frac{K_m K_v}{A_p^2}$, rad/s; $\omega_h$ is the natural frequency of valve control cylinder, $\omega_h = \frac{4 \beta \omega^2_e}{V_t} = \frac{K_h}{m_i}$, rad/s; $K_h$ is hydraulic spring stiffness of hydraulic cylinder, $K_h = \frac{4 \beta \omega^2_e}{V_t}$, rad/s; $\xi$ is damping ratio of valve control cylinder, $\xi = \frac{K_v}{A_p} \sqrt{\frac{\beta}{V_t}} + \frac{B_i}{4 A_p} \sqrt{\frac{V_t}{\beta m_i}}$.

When the hoist is braking, the equation of torque balance on the hoist drum is as follows: [1,10]

$$M_Z \pm M_j = M_d$$

(14)

where $M_Z$ is braking torque, $M_j$ is Static resistance torque, and $M_d$ is inertia torque of lifting system.

$$M_Z = 2 \left( K_m x_0 - P_i A_y \right) \mu R_j$$

(15)

$$M_j = k m g R_j$$

(16)

$$M_d = \sum m a R_j$$

(17)

where $K_m$ is spring stiffness, N/m; $x_0$ is spring precompressed length, m; $P_i$ is brake system pressure, Pa; $A_y$ is brake cylinder area, m$^2$; $\mu$ is friction coefficient of brake shoe; $R_j$ is braking radius, m; $k$ is mine resistance coefficient; $m$ is the
load mass, kg; \( g \) is gravity acceleration, m/s\(^2\); \( R \) is drum radius, m; \( r \) is braking radius, m; \( \sum m \) is equivalent mass of lifting system, kg; \( a \) is deceleration of mine hoist, m/s\(^2\).

Substitution Eq. (14) with Eqs. (15) (16) and (17) we obtain:

\[
a = \frac{K_x x_0 - P_i a_j}{\sum m R_j} \mu R_j \pm kmgR_j
\]

(18)

The Laplace transformation of Eq. (18) is as following:

\[
\frac{a}{P_i} = \frac{-A_x \mu R_i}{\sum m R_i}
\]

(19)

The transfer function with \( p_i \) as input and \( I_r \) as output is proportional link:

\[
\frac{I_r}{P_i} = K_p
\]

(20)

where \( p_i \) is brake pressure, and \( I_r \) is the feedback current of the pressure comparer.

The transfer function with \( a_{err} \) as input and \( I_e \) as output is proportional link:

\[
\frac{I_e}{a_{err}} = K_p
\]

(21)

where \( a_{err} \) is deceleration deviation, and \( I_e \) is the control current.

The transfer function with \( I_{err} \) as input and \( I \) as output is proportional link:

\[
\frac{I}{I_{err}} = K_e
\]

(22)

where \( I_{err} \) is current deviation, and \( I \) is the control current.

In summary, the transfer function block diagram of the hoist braking system is shown in Fig. 6.
4. Simulation experiment

It can be seen from the block diagram of hoist braking system that the braking system is 0 type system. 0 type system can not follow the ramp signal, and has steady-state deviation when follow the unit step signal. This kind of system usually needs PI or PID correction. In this paper, the control method of double PID is adopted. Because the acceleration is negative, we should pay attention to positive and negative sign when modeling simulation model [1]. The simulation model is shown in Fig. 7.

![Simulation model in Simulink](image)

The calculation process of each parameter in the simulation model is omitted, and the values of each parameter are given in Table 1.

### Table 1. Parameter values in simulation

<table>
<thead>
<tr>
<th>Serial number</th>
<th>Symbol</th>
<th>Unit</th>
<th>Quantity</th>
<th>Serial number</th>
<th>Symbol</th>
<th>Unit</th>
<th>Quantity</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>$K_v$</td>
<td>$m^3/(mA\cdot s)$</td>
<td>8.34E-05</td>
<td>12</td>
<td>$\xi_0$</td>
<td>-</td>
<td>0.0488</td>
</tr>
<tr>
<td>2</td>
<td>$K_v$</td>
<td>$m^3/(Pa\cdot s)$</td>
<td>1.3E-09</td>
<td>13</td>
<td>$\omega_n$</td>
<td>rad/s</td>
<td>753.6</td>
</tr>
<tr>
<td>3</td>
<td>$C_i$</td>
<td>$m^3/(Pa\cdot s)$</td>
<td>4.5E-13</td>
<td>14</td>
<td>$\xi_m$</td>
<td>-</td>
<td>0.63</td>
</tr>
<tr>
<td>4</td>
<td>$\beta_r$</td>
<td>Pa</td>
<td>1.60E+09</td>
<td>15</td>
<td>$K_d$</td>
<td>m/mA</td>
<td>0.001</td>
</tr>
<tr>
<td>5</td>
<td>$A_p$</td>
<td>$m^2$</td>
<td>0.26944</td>
<td>16</td>
<td>$m$</td>
<td>kg</td>
<td>28 000</td>
</tr>
<tr>
<td>6</td>
<td>$B_t$</td>
<td>$N\cdot s/m$</td>
<td>0.45</td>
<td>17</td>
<td>$R_j$</td>
<td>m</td>
<td>4.54</td>
</tr>
<tr>
<td>7</td>
<td>$K_w$</td>
<td>$N/m$</td>
<td>3.3E+08</td>
<td>18</td>
<td>$R_i$</td>
<td>m</td>
<td>2.45</td>
</tr>
<tr>
<td>8</td>
<td>$V_i$</td>
<td>$m^3$</td>
<td>0.0025</td>
<td>19</td>
<td>$\mu$</td>
<td>-</td>
<td>0.3</td>
</tr>
<tr>
<td>9</td>
<td>$m_r$</td>
<td>kg</td>
<td>160</td>
<td>20</td>
<td>$k$</td>
<td>-</td>
<td>1.15</td>
</tr>
<tr>
<td>10</td>
<td>$\omega_r$</td>
<td>rad/s</td>
<td>5.918</td>
<td>21</td>
<td>$\sum m$</td>
<td>kg</td>
<td>238863.56</td>
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<td>11</td>
<td>$\omega_h$</td>
<td>rad/s</td>
<td>34081.76</td>
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</tr>
</tbody>
</table>

The calculation process of each parameter in the simulation model is omitted, and the values of each parameter are given in Table 1.
The Simulink Design Optimization/Signal Constraint--Check Step Response Characteristics is used to optimize the inner and outer PID parameters respectively. The PID parameters of inner ring are 12.69, 1.75 and 1. The PID parameters of outer ring are 43.58, 1 and 1.7. The deceleration is set to 3.5 m/s². The simulation results are shown in Fig. 8(a). It can be seen from the diagram that the double PID control system can meet the control requirements of deceleration during emergency braking.

In order to verify the stability and robustness of the system, disturbance is added to the inner ring and the outer ring respectively. The disturbance is used to simulate the sensor measurement error, the noise in the signal transmission process or the additional force caused by the flexibility of the wire rope. The module of Band-Limited White Noise is used to simulate the disturbance, and the parameter of noise power is set to 1e5. The simulation model and deceleration simulation results after adding disturbance are shown in Fig. 8(b). It can be seen from Fig. 8(b) that when disturbance exists in the system, it can also meet the requirement of constant deceleration of the braking system. The stability and robustness of the double PID control system are verified by the simulation of adding disturbance.

5. Test Verification

In this paper, the test bed is the test platform system of the ultra-deep well hoist of CITIC HEAVY INDUSTRIES. According to the similarity theory, the test bed is reduced to 0.1 of the actual hoist, and the main parameters of the test bed are shown in Table 2. The design drawing and site photos are shown in Fig. 9.
Table 2.

<table>
<thead>
<tr>
<th>Name</th>
<th>Specifications</th>
</tr>
</thead>
<tbody>
<tr>
<td>drum diameter</td>
<td>800mm</td>
</tr>
<tr>
<td>width</td>
<td>160mm</td>
</tr>
<tr>
<td>hoisting height</td>
<td>47m</td>
</tr>
<tr>
<td>diameter of wire rope</td>
<td>10mm</td>
</tr>
<tr>
<td>payload</td>
<td>1t</td>
</tr>
<tr>
<td>volume weight</td>
<td>1t</td>
</tr>
<tr>
<td>hoisting speed</td>
<td>1.8m/s</td>
</tr>
<tr>
<td>motor powers</td>
<td>75×2kW</td>
</tr>
<tr>
<td>brake number</td>
<td>4×2</td>
</tr>
</tbody>
</table>

The test-bed. (a) The design drawing; (b) The site photos.

The test is carried out with no extra loading, which is equivalent to the no-load safety braking with high speed. The safety braking occurs with a deceleration of 2 m/s² when the hoist speed reaches 20 m/s. In the experiment, the Tektronix MDO4024C type oscilloscope is chosen for collecting data. Sample frequency is set to 2500 Hz. The data obtained from sampling is shown in Fig. 10. As shown in Fig. 10, the blue curve represents setting speed, the red for actual speed and the green for hydraulic system pressure. It can be seen from the figure that for the hoist with dual PID control, the error between the preset speed and the actual measured speed is very small. The deceleration converted from the actual speed value of the braking system is shown in Fig. 11. As shown in Fig. 11, the deceleration soon reaches the set value, meeting the requirements of the hoist brake system.
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Fig. 11. The setting speed, actual speed and hydraulic system pressure obtained by sampling.

Fig. 12. The deceleration value of the braking system

6. Conclusion

In this paper, the mathematical model of every component of braking system of mine hoist is established, and the transfer function of emergency braking system is obtained. Through the analysis of transfer function, it is shown that the emergency braking system belongs to type 0 system. When the unit step signal is input, the system has steady-state deviation, so it is necessary to correct the system. The double-closed loop control correction of double PID is adopted in this paper, the stability and robustness of the constant deceleration control system are verified by Simulink simulation, and the practicability is verified on the test bed of the ultra-deep well hoist of CITIC HEAVY INDUSTRIES. This paper provides the theoretical basis for the constant deceleration braking control of hoist and has certain academic and engineering value.
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