Research on EMB electro-mechanical brake system control algorithm

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ABSTRACT

Under the background of the development of new energy vehicles, the traditional three major parts of vehicles change with the change of energy supply. The traditional hydraulic brake can no longer adapt to today's electro-mechanical system, and the EMB system has become the inevitable direction of the development of the brake system. The current closed-loop control system of the EMB design is not fully developed, and there is a lack of consideration for the impact of brake clearance. In this paper, a lightweight EMB braking system is designed, and key mathematical models related to it are established. The control output of braking force is optimized by the PID algorithm. At the same time, a brake clearance compensation strategy is proposed. After simulation test, the system has good braking force control performance, and reduces the incidence of motor underspeed, and the overall system has good feasibility and stability.

Keywords: EMB, automotive braking systems, new energy vehicles, PID algorithm, Anti-lock braking

1. INTRODUCTION

With the increasing prominence of the global energy crisis and environmental problems, promoting new energy vehicles has become an effective way for countries to solve the energy crisis and reduce environmental pollution, and it is the only way for sustainable development.

The automotive industry is gradually developing towards the trend of new three modernizations, and the traditional three major parts of vehicles have changed with the change of energy supply mode. After the engine becomes an electric motor, the way of braking force will change, so the traditional chassis brake will inevitably become the by-wire chassis brake, the mechanical brake-by-wire system (electromechanical braking system, that is, EMB) and EHB will become the mainstream, because EHB is a transitional form from the traditional braking system to EMB. So, the future direction must be electro-mechanical braking systems[1].

In the early 90s of last century, some well-known automobile manufacturers began to develop EMB actuators. Although EMB actuators are mainly composed of motors, deceleration torque increasing mechanisms, motion conversion mechanisms, etc., the relevant designs of parts manufacturers and automobile companies are different in the structure of EMB actuators.

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No matter what kind of system safe and efficient operation is the prerequisite for having a set of accurate control algorithms. Chris Linel et al. realized that the important node in the operation of the EMB system is to control the output of the brake clamping force more accurately[2]. To verify the relevant hypotheses, a mathematical model of the EMB system is constructed, and a PI control strategy based on the EMB system with the brake clamping force as the outermost ring is proposed based on series closed-loop control. The results show that when the expected brake clamping force changes significantly, the proposed strategy has a better follow-up to it. In the process of studying the EMB control algorithm, Chihoon Jo et al. considered the friction characteristics between the components in the actuator, and proposed a novel clamping force estimation algorithm, which makes good use of the correlation between the drive motor angle and the real brake clamping force. The results show that the proposed strategy can accurately and effectively estimate the clamping force[3]. Young-Hun Kil et al. proposed a method to estimate the brake clamping force of the brake by referring to the hysteresis characteristics of the motor and the rotor position and used the force-displacement control system to realize the adjustment of the brake clearance[4]. Heeram Park et al. found in the process of studying the EMB system that small changes in the relevant parameters of the system can cause drastic changes in the entire EMB system, and it is difficult to measure the real-time brake clamping force during the working process of the EMB system. Based on the above problems, Heeram Park et al. proposed a new adaptive control algorithm, which can significantly improve the tracking of brake clamping force without the help of external sensors[5].

Although major enterprises and universities have successively carried out research and development of EMB executive institutions, they still have some shortcomings in terms of structure, which require good control algorithms to make up for[6]. At present, the research on EMB control algorithms mainly focuses on accurately output the required brake clamping force, improve the reaction speed of the brake system, and identify relevant critical points. In this paper, we try to improve the dynamic performance of the EMB system through control engineering optimization and additional brake clearance compensation strategies, reduce the steady-state error, and improve the overall stability of the system.

2. EMB BRAKE STRUCTURE DESIGN

In this chapter, we build the basic structure of a small EMB brake suitable for new energy vehicles as the hardware basis for the subsequent EMB control algorithm. The overall structure of the EMB brake is shown in Fig 1 below.

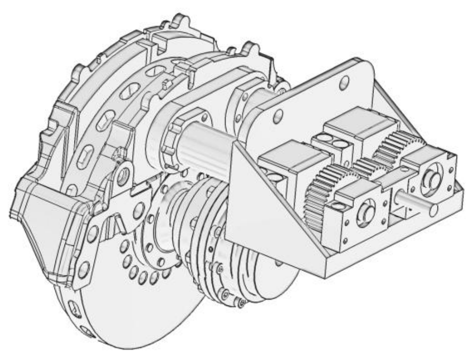

Figure 1. Overall appearance of EMB brakes.

The outermost side of the EMB system is a unit housing designed for a lightweight system, with internal space reserved for pushers and connecting plates. The power take-off mechanism is connected to the planetary reduction mechanism. On the other side there is a thrust actuator linked to the brake caliper that performs the braking function of the EMB system.

The main principle of the device includes the following processes. A pair of wedges are erected between the support of the device and the rotating brake disc, which push the brake linings when moving relative to each other, thereby generating braking force, and at the same time, unlike the hydraulic brake control system of conventional cars, the EMB system The servo motor is used to control the movement of the wedge so that it does not completely lock. Under intelligent control, the wedge directly converts the kinetic energy of the car into braking energy and completes the braking work with lower energy consumption and faster corresponding speed.

We build a basic human-computer interaction model for the EMB system here, as shown in Fig. 2 below.

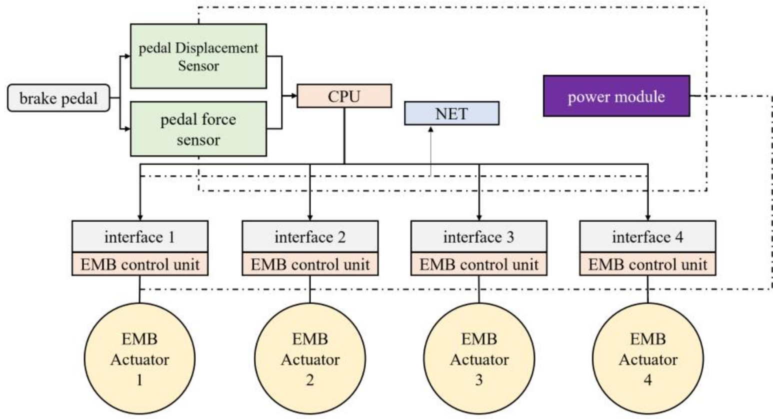

Figure 2. Illustration of the EMB system.

In the interactive system we have established, all brake lines and hydraulic oil are eliminated, and information is exchanged through the electromechanical system. Various sensors measure the output pressure of the wheel-end EMB actuator, and the on-board computer network uses CAN communication. The parking brake switch responds to the driver's service brake request and controls the generation and elimination of the parking brake. The central controller integrates the signals from the vehicle status sensors and calculates the braking torque of the front and rear axles, while transmitting data with the front and rear axle controllers. After the front and rear axle controllers accept the braking torque request, it calculates the correct braking torque according to the ground adhesion coefficient and load and transmits it to the EMB at the left and right wheel ends through the CAN bus Controller.

In the above article, we built the basic framework of the EMB system, and in this chapter, we build the mathematical model of the core components of the EMB actuator. The main components that need to be built include a description model of the drive motor, a motor friction model, and a model of load and brake. The detailed process is given below.

3. EMB ACTUATOR COMPONENTS

3.1 A descriptive model of the drive motor

To simplify the model, we assume that the drive motor meets certain ideal conditions during operation. First, we believe that the drive motor meets the ideal commutation process, that is, ignores the losses caused by mechanical commutation during operation. Secondly, the influence of self-inductance and mutual inductance in the motor winding is ignored, and the intermediate transition process caused by other current changes is ignored. Our model also ignores the effects of armature reactions and rotor fluctuations caused by the air gap distribution magnetic field[7]. Based on the above assumptions, we can give the following form of motor model as Fig. 3.

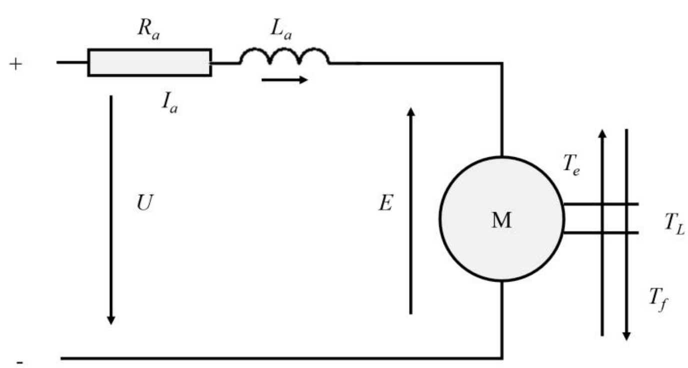

Figure 3. Schematic diagram of the motor model.

The motor voltage *U* can be calculated using the following formula.

$$U = L\frac{dI}{dt} + RI + E \tag{1}$$

The motor back EMF *E* in the equation can be expressed by the following equation.

$$E = K\frac{d\theta}{dt} \tag{2}$$

The motor torque is calculated using the following formula.

$$J\frac{d^2\theta}{dt} = T_m - T_f - T_L \tag{3}$$

The calculation formula of the motor stall torque Tm is as follows.

$$T = K_t I \tag{4}$$

In the above series of formulas, *I* represent the armature current; *R* represents the armature resistance; *K* is the motor back EMF coefficient; K_t represents the motor torque coefficient, *J* represents the moment of inertia; θ indicates the motor rotation angle; T_f represents the friction torque of the motor; T_L represents the motor load torque.

3.2 Motor friction model

In the above model construction, we ignore the loss in the commutation process, and no longer consider the additional effect of the intermediate change process of the current on the armature, so we mainly need to consider the mechanical friction process in the operation of the motor and analyze this process[8].

It should be noted that the friction characteristics of the motor are divided into static friction characteristics and dynamic friction characteristics, considering the actual situation, we use the static friction characteristic model in the general model[9]. In this model, we comprehensively consider the static friction, coulomb friction and viscous friction of the motor, while also avoiding repeated acquisition of various empirical coefficients and critical speed values[10].

In the constructed motor friction, we calculate the magnitude of the frictional moment by the following formula.

$$T_f = \begin{cases} T_e & \left(\dot{\theta} = 0 \cap |T_e| < T_s\right) \\ T_s \mathrm{sgn}(T_e) & \left(\dot{\theta} = 0 \cap |T_e| \geq T_s\right) \\ T_c \mathrm{sgn}\left(\dot{\theta}\right) + K_v\dot{\theta} & \left(\dot{\theta} \neq 0\right) \end{cases} \tag{5}$$

where θ is the relative sliding speed of the motor; T_e is the external torque; T_s is the maximum static friction torque; T_c is the coulomb friction; K_v is the viscous coefficient of friction. For the friction characteristic coefficient in the model, we remove the coulomb friction moment of 0.0192Nm; The maximum static friction moment is 0.0387 Nm; The viscous friction coefficient is 0.0011 Nm (rad/s) [11].

3.3 Load and brake model

In this section, we model the load and brake, the ultimate purpose of the electro-mechanical braking system is to convert the torque of the motor into a clamping force on the brake disc through a series of structural operations to complete the braking operation. In total, we consider that the clamping force on the brake disc is proportional to the third square of the nut's displacement[12]. If we express the clamping force coefficient in K_F, the clamping force on the brake disc can be expressed by the following formula.

$$F = K_F x^3 \tag{6}$$

When the EMB performs the work, the kinematic mechanism converts the rotation of the motor into nut translation[13]. For the EMB actuator in this article, we can consider that the load moment of the brake clamping force reaction on the lead screw is equal to the load moment of the gear reducer, and the clamping force reaction force received by the lead screw nut is equal to the clamping force received by the brake disc[14]. Through the above analysis, the load torque T_{nL} thrust reverse motor can be affected by the planetary carrier. The above relationship can be expressed with the following formula.

$$T_{nL} = \left(\frac{L}{2\pi}\right) F_{ns} \tag{7}$$

where F_{ns} is the clamping force reaction force to the lead screw nut.

For the purposes of this article, since the EMB brake adopts the floating caliper disc type, both sides of the brake disc are subjected to friction during operation, so the brake model can be described, which is expressed by the following formula[15].

$$T_b = 2Fu_bR_b \tag{8}$$

where T_b is the output torque of the brake.

3.4 EMB system PID control algorithm

In this paper, our goal is that the parameters related to the brake clamping force can be output quickly, accurately, and stably following the expected value, and it can be seen from the model analysis above that the rotational motion of the motor is transmitted through a series of mechanisms, and finally the power is converted into the required brake clamping force. Therefore, the core of the EMB system control is the control of the drive motor[16].

The system we built required motors to be able to respond quickly to system revenue. We chose PID control as the core control method of EMB system.

As one of the widely used control methods in control engineering, PID control has the most important advantages including simple and clear structure, easy adjustment, and high stability. For the controlled object, PID control will become the most effective means when we do not understand a lot of its information, or when the specific and effective parameters cannot be obtained.

For the PID system, based on the error of the controlled system, the effective output value of the control quantity can be calculated by continuously adjusting the three parameter values of proportion, integration and differentiation and then input to the system to achieve normal and efficient operation of the system. PID controller is to use the deviation signal to obtain the specific size of the proportional value, integral value and differential value, to complete the relevant regulation of the input parameters in the controlled system, which can be summarized as the use of deviation, eliminate deviation, the basic working principal diagram shown in Fig. 4 below.

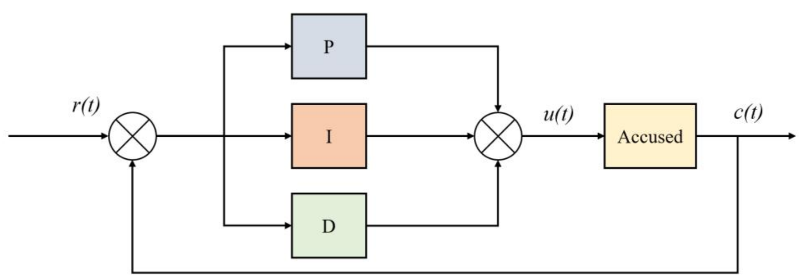

Figure 4. PID basic control principle.

PID controller consists of proportional, integral, differential coefficients, respectively expressed as KP, KI and KD, PID controller by controlling the deviation of input and output results to complete the adjustment of the system, the controller It is expressed by the following formula. where u represents the controlled quantity.

$$u(t) = K_p e(t) + K_I \int_0^t e(t)dt + K_d \frac{de(t)}{dt} \tag{9}$$

During the application, the operation effect of the entire control system is affected by the selection of three coefficient values to be determined. Furthermore, when building this controller, the selection of PID parameter values should be considered from multiple perspectives. The actual value of the parameter value is influenced by the mechanical dynamic characteristics of the system itself, which in this case is influenced by the physical characteristics of the EMB actuator components[17]. In actual engineering, the tuning methods of related parameter values generally include two types, calculation tuning and engineering tuning. We use engineering tuning here to determine the PID controller parameters.

The most important control variable for the EMB system in this article is the brake clamping force[18]. Therefore, it is necessary to build a three-stage system of speed-current-clamping force, in which the clamping force ring, as the outermost ring, controls the brake clamping force of the braking system. The speed and current loop are the inner loops of the system, which play the role of timely adjustment of the interference in the loop, constrain and guarantee the motor, to effectively promote the control of the outer loop. The frame diagram of the force-speed-current three-loop closed-loop control system is shown in Fig. 5 below.

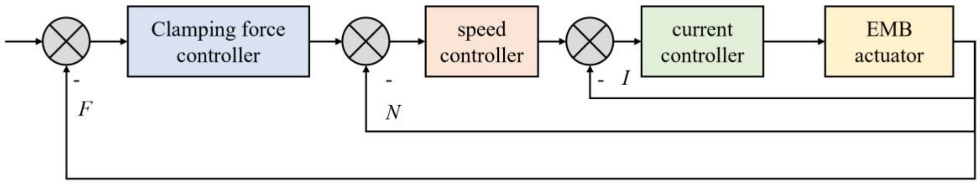

Figure 5. EMB three-closed-loop control system schematic.

In the above schematic model, we can see that the process of three-closed-loop control mainly includes the following stages.

First, the system inputs a target clamping force to the EMB system and compares it to the actual clamping force calculated from the nut displacement in the system. The difference between the two is used as the final input of the clamping force control loop, and the adjusted output value is calculated by the PID controller in the clamping force control loop as the desired speed in the EMB system.

The calculated desired speed is then compared with the actual speed in the EMB system, using the difference between the two as the final input to the velocity loop. The output value calculated and adjusted by the PID controller in the speed loop is used as the expected current in the EMB system.

Finally, the expected current in the EMB system is compared with the actual current, and the difference between the two is used as the final input of the current loop, and the adjusted output value calculated by the PID controller in the current loop is used as the input value of the final EMB actuator.

For the performance indicators of this system, we propose the following standards: the stability error of the clamping force ring does not exceed 5%, and the speed control error does not exceed 5% of the maximum speed[19].

3.5 Brake clearance compensation strategy

Due to the operating characteristics of the EMB system itself, two critical points need to be identified during braking. The two zero thresholds are the contact critical point and the separation critical point. The brake clearance compensation strategy in this chapter is based on the three indicators of clamping force, vehicle speed and lead screw nut displacement of the EMB system, and determines whether the brake clearance is eliminated, whether the braking process ends, and whether a fixed brake clearance is generated through the analysis of relevant parameters to complete the identification and judgment of the above two critical points, and then complete the control of the brake clearance. The overall process can be represented by Fig. 6.

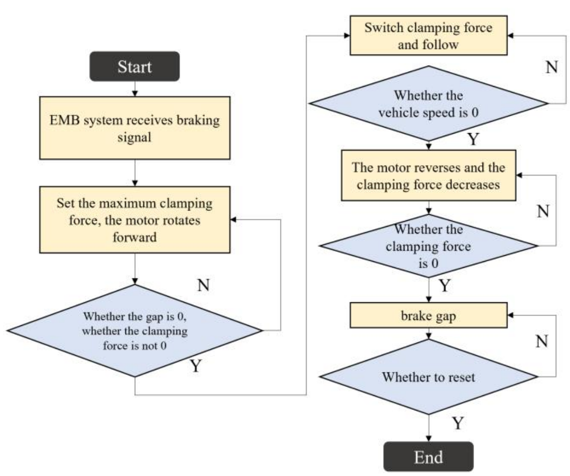

Figure 6. Brake clearance compensation process.

To quickly eliminate the brake clearance, the target clamping force is set to the maximum clamping force Fmax that the EMB system can provide during the brake clearance elimination phase, and the motor eliminates the brake clearance in the shortest possible time. When x reaches the set value range and the brake clamping force is just not O, the brake clearance is eliminated. At this time, the real-time target brake clamping force of the brake is adjusted according to the driving conditions of the vehicle and according to the system settings until the vehicle speed is 0. When the vehicle stops, the reverse voltage is input at both ends of the motor until the brake clamping force is 0, and the ball screw pair returns to the zero initial position, a fixed brake clearance is generated, and the control ends.

4. ACCURACY TEST AND RESULT INSPECTION

In this chapter, we build a simulation platform to test the overall performance indicators of the built EMB system.

We first verified whether the EMB model we built was able to quickly respond to and maintain stable braking force when the upper controller decided to brake thrust. Here we construct the brake thrust step case and measure the dynamic correspondence of the entire braking system when working, and the overall result is shown Fig. 7 and Fig. 8 below.

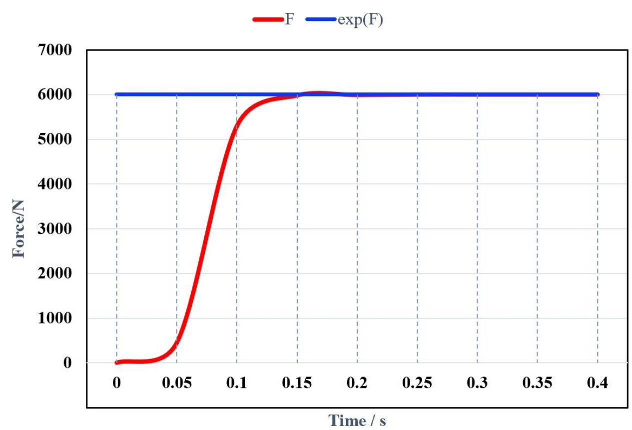

Figure 7. 0-6 kN step signal input curve.

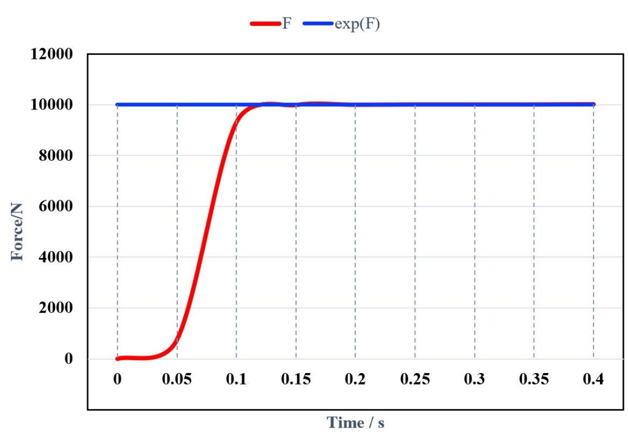

Figure 8. 0-10 kN step signal input curve.

From the test results, it can be concluded that under the medium thrust step condition of 6 kN, the system starts the corresponding expected braking thrust in 36 ms, which is less than 0.1 of the system requirement indicator s, indicating that the braking force of the designed system has good dynamic characteristics, and the braking system establishes braking reasoning that meets the requirements at 143 ms, and the steady-state error is less than 3%, which meets the requirements of the designed system.

Under the high thrust step condition of 10 kN, the system starts to expect braking thrust accordingly in 42ms, which is less than 0.1 s of the system requirement specification. It shows that the braking force of the designed system has good dynamic characteristics, and the braking system establishes braking reasoning that meets the requirements at 133.6 ms, and the steady-state error is less than 3%. Combined with the comprehensive conclusion of the two, it shows that the EMB system designed in this paper can meet the requirements of medium and large braking force in terms of dynamic response ability and steady-state error index. The overall characteristics of the system are good.

Subsequently, we tested the impact of the brake clearance compensation system on the EMB system. We give a low target clamping force and record the speed response of the EMB system via simulation data. The result obtained is shown in Fig. 9 below.

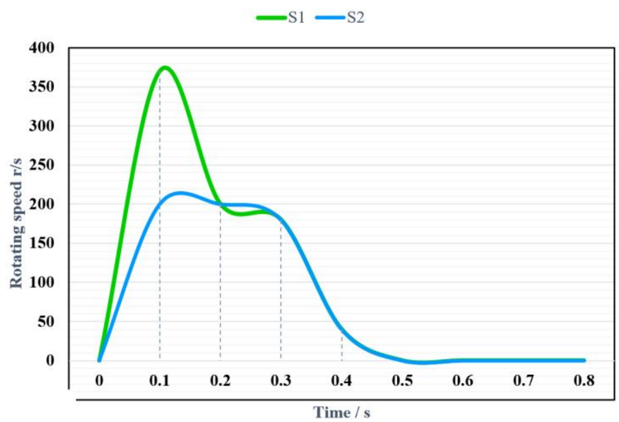

Figure 9. Comparison of EMB system response under brake clearance compensation.

The S1 series represents an EMB system with added clearance compensation, and the S2 represents an EMB system without clearance compensation. It can be seen from the results in the figure that in the absence of brake clearance control strategy in the EMB system, when the target clamping force of the brake is small, The maximum speed that the motor can reach in the process of eliminating the brake clearance is lower than the rated maximum speed of the motor, and the time to eliminate the brake clearance increases, which affects the safety of the EMB braking process. When we adopt the designed brake clearance control strategy, when eliminating the brake clearance, the system automatically sets the expected clamping force to the maximum braking force that the EMB system can provide, so that the motor reaches the highest speed and quickly eliminates the brake clearance, which can ensure the reliability and safety of the EMB system.

5. CONCLUSION

With the further development of new energy vehicles, the limitations of traditional hydraulic braking systems are becoming more and more difficult to overcome, and in view of this practical problem, we propose an EMB electronic braking system scheme in this paper. Through the mechanical design of the system, it can be controlled by electromechanical systems away from the hydraulic system used in conventional automobiles. In line with the development direction of new energy vehicles. For the EMB actuator structure we designed, we build a mathematical model for the actuator. The main influencing factors of the related system are analyzed, and the corresponding three-loop PID control algorithm is constructed for the most important three factors of speed-current-clamping force. While ensuring good dynamic response, it has high clamping force control accuracy. At the same time, we provide a brake clearance compensation strategy to improve the stability and safety of the overall EMB braking system for the wear problem of the braking system.

For the above design scheme, we carried out the system simulation and performance index test, and the test results show that the overall system we designed has a high response speed under the premise of medium and high target braking force and can effectively control the overall adjustment error after the control system reaches steady state. At the same time, the compensation strategy of the braking system can effectively avoid the speed bottleneck of the motor under low braking force, quickly eliminate the brake gap, quickly recover the motor speed, and effectively improve the stability of the braking process.

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