Control of Electronic Mechanical Brake System on Automobile Electromechanical Brake

Yiqing Chen *, Youtong Li
School of Guang'an Vocational and Technical College, Guang'an, China

* Corresponding author e-mail: Yiqing@gavtc.cn

Abstract. To verify the brake effect and the mechanism of exerting the brake of the electronic mechanical brake (EMB) system in the brake process of electronic vehicles, the working environment and function of each component of EMB are analysed based on the current relevant laws and regulations. Each component of EMB is calculated and selected. The mechanical relationship between the components of the EMB system are analysed, the preliminary EMB structure is designed, and the key parameters such as EMB brake pressure and EMB brake clearance elimination time are tested. Results show that the difference between the experimental value and theoretical value of the brake pressure test of the preliminary designed EMB structure aren’t large, and the brake clearance force increases with the increase of the locked-rotor voltage. When the voltage increases to 12V, the theoretical brake clearance force is 16510N and the experimental value is 15998N. During the EMB brake clearance elimination time test, the maximum speed of the brake block running at a constant speed during the elimination of brake clearance phase is 2.39mm/s, which is similar to the theoretical value $V_{max}=2$mm/s. The experimental and theoretical motor brake speed is reduced to 88 rpm/min at 0.15s and at 0.13s. With all indicators considered, the EMB system can be used to control electromechanical brakes and provide certain guidance for its development.

Keywords. EMB system; automotive electronic machinery; brake clearance; brake elimination.

1. Introduction
With the continuous advancement of economic globalization and the rapid development of modern science and technology, people's attention to cars is not limited to price and appearance, but to their eco-friendly, lightweight and safety properties [1]. The traditional hydraulic brake system presses the hydraulic oil from the master cylinder to the brake disc or brake drum through the hydraulic line with the assistance of the vacuum booster, to realize the brake of the vehicle [2]. As a new significant brake system, EMB is of small size and light weight and it uses wire lines as a signal and energy transmission medium [3]. Environmentally friendly electrical energy provides the energy source for the entire brake system. The fast-response and high-efficiency motor is used to drive the end brake actuator to brake [4].

Based on the advantages of EMB brake, the mechanical relationship between each component of EMB are analyzed and calculated to preliminarily design the EMB system, and the key parameters such
as EMB brake pressure and EMB brake clearance elimination time are tested to provide guidance for further development.

2. Methodology

2.1. EMB actuator’s selection and design
As a brand new brake system, the EMB system must conform to the current domestic automobile brake regulations in selection and design, which means the brake clearance time is within 0.1-0.3mm, the brake clearance elimination time is within 0.05s-0.15s, and when a full-loaded car are driving on a clean concrete road and emergency brakes at 80km/h, the braking distance is less than 50.7m. Based on the actual parameters of the target model, the actuator’s target brake force and the brake clearance elimination time of this EMB system are determined according to the current car brake regulations and the maximum brake clearance force provided by the hydraulic brake system of the original vehicle. Then the ball screw pair, planetary gear reduction mechanism and permanent magnet DC torque motor are selected and designed.

2.2. Calculation methods of brake designing parameters
To achieve a large reduction ratio in a limited space, the sun gear and the planet gears are carburized and quenched via 20Cr Mn Ti, the hardness of the tooth surface is within 56~60HRC, $\sigma_{H_{lim}} = 1500\text{MPa}$.

The outer ring gear is quenched and tempered via 42Cr Mo, the hardness of the tooth surface $\geq 260\text{HBS}$. Gear accuracy is 7 levels. The equation for calculating the center distance of the planetary reduction gear according to the contact strength is as follows.

$$a \geq 483(u+1)\sqrt{\frac{K\tau_a}{\phi_a\sigma_{H_{lim}}}}$$

(1)

The input torque of the sun gear $T_a = 24\text{N.m}$, the gear ratio $u = 2.5294$, the comprehensive coefficient $K = 2$, and the tooth width coefficient $\phi_a = 0.5$.

The equation for calculating the torsion shaft diameter is shown in equation (2), where $[\tau] = 48\text{N/mm}^2$, motor input torque $T = 24\text{N.m}$, and plane scroll spring input torque $T = 20\text{N.m}$.

$$d \geq \sqrt[3]{\frac{ST}{[\tau]}}$$

(2)

The equation for calculating the circumferential force received by the sun gear is as follows, where the input torque of the sun gear $T_a = 24\text{N.m}$, the number of planet wheels $n_w = 3$, and the radius of the sun gear index circle $r_a = 12.75\text{mm}$.

$$F_{wca} = \frac{1000T_a}{n_w r_a}$$

(3)

The equation for calculating the radial force received by the sun gear is as follows, where the normal angle of the sun gear pressure $\alpha_n = 20^\circ$ and the helix angle on the index circle of the sun wheel $\beta = 0^\circ$.

$$F_{rcs} = F_{wca} \frac{\tan \alpha_n}{\cos \beta}$$

(4)
The calculation of the maximum brake pressure of a single brake is as follows, where $P$ is the maximum hydraulic pressure of the brake line in Mpa, $d$ is the piston diameter in mm.

$$N_{\text{max}} = P \times \frac{1}{4} \pi d^2$$  \hfill (5)

The calculation of the surficial brake force is as follows, where $T_\mu$ is the friction torque between the brake disc and the friction pad, and $r$ is the wheel radius.

$$F_{\text{sb}} = \frac{T_\mu}{r}$$  \hfill (6)

The equation for calculating the maximum surficial brake force of a single-sided wheel is as follows, where $F_z$ is the normal force of the wheel from the ground, $\varphi$ is the ground adhesion coefficient, $\beta$ is the power distribution coefficient.

$$F = \frac{1}{2} F_z \varphi \beta$$  \hfill (7)

The calculation for the clamping force received by the one-sided brake disc during parking brake is as follows, where $R_e$ is the effective radius of the wheel, $m$ is the mass of the car, $\alpha$ is the slope, $f$ is the coefficient of friction between the brake pad and the brake disc, $R$ is the average radius of the fan-shaped friction pad.

$$F_o = \frac{1}{4} mg R_e \sin \alpha$$  \hfill (8)

2.3. Calculation methods of driving motor designing parameters

When EMB is working, the torque of the motor is converted into the clamping force of the brake pad and brake disc via the ball screw [5]. To get enough clamping force, a motor with enough torque is chosen by calculating the clamping force. The conversion relationship between the torque of the motor and the thrust of the ball screw is as follows.

$$F_a = \frac{2 \pi \eta T_a}{L}$$  \hfill (9)

$F_a$ is the thrust generated by the ball screw, $T_a$ is the input torque, and $\eta$ is the transmission efficiency of the ball screw, which is 90% taking into account the actual situation. $L$ is the ball screw lead.

2.4. Tests for EMB system brake performance

The slider is fixed on the guide rail and keep still, the brake block is applied directly to the head of the pressure sensor, and locked-rotor voltage includes 2V, 4V, 6V, 8V, 10V and 12V are applied to the torque motor respectively. All the data on the pressure indicator is recorded.

Whether the EMB system elimination brake clearance time meets the usage requirements is tested by the indirect measurement. The slider is fixed on the guide rail to keep still. Displacement sensors are installed in parallel at both ends of the brake block inside the brake disc. The time that the two displacement sensors used to move 2mm, 3mm, 4mm and 5mm are taken down for many times, and 8kN target clamping force is applied, the motor speed is observed and compared with the theoretical value.
3. Results and discussion

Based on the actual parameters of the target vehicle model and the current car brake regulations, the target brake force and brake clearance elimination time of the EMB system are determined according to the maximum brake clearance force that the original vehicle hydraulic brake system provides. Then the ball screw pair, planetary gear reduction mechanism and permanent magnet DC torque motor are selected and designed. The selection design process is shown in Figure 1. First, parameters such as the brake gap of the target model and the time to eliminate the brake gap are determined, the parameters of the specific components are calculated according to the relevant calculation method to obtain the maximum brake force of the vehicle. The specific parameters of the EMB motion conversion mechanism are determined according to the national standard. The appropriate type of reducer and motor power and torque are determined according to the usage occasion. The time to eliminate the brake gap is calculated according to the motion conversion mechanism, reduction ratio and motor. The reduction ratio should be adjusted until it meets the requirements. The relevant parameters of the reducer can be designed and checked for intensity. Then the EMB structure diagram is drawn.

EMB is an implementation form of wire-controlled brake, the motor will rotate and drive the planetary gear reducer after the brake signal is input. After the planetary gear reducer reduces the torque, the rotary motion of the motor turns into the linear motion of the brake block via the ball screw, then the torque of the motor turns into the clamping force of the brake [6]. Based on the electronic parking brake structure, combined with the working principle of EMB, the EMB structure diagram is shown in Figure 2, which is mainly composed of a motor, a deceleration mechanism, a brake mechanism, and a motion conversion mechanism.

![Figure 1. EMB design flow chart.](image1)

![Figure 2. EMB structure diagram.](image2)
The difference between the experimental value and the theoretical value of the brake clearance force is not large, indicating that the selection and design fully meets the usage requirements. The brake clamping force increases with the increase of the locked-rotor voltage, which changes approximately in a proportional manner, which means the brake force can be adjusted while adjusting the voltage. When the voltage is increased to 12V, the theoretical brake clamping force is 16510N, and the experimental value is 15998N. The difference between the experimental value and the theoretical value is relatively small, possibly due to the installation error in the test process, the motor’s mechanical properties, the complex environmental conditions of the outside world [7].

![Figure 3. Comparison of curves of brake clearance force.](image)

As shown in figure 3, the brake pad runs at a constant speed whose peak value is 2.39mm/s during the brake elimination clearance stage through. The result is close to the theoretical value $V_{\text{max}}=2\text{mm/s}$. The results both meets the usage requirements.

| Test times | Sensor displacement | Time (s) | Brake pad’s speed(m/s) |
|------------|---------------------|---------|------------------------|
| 1          | 2mm                 | 0.85    | 2.38                   |
|            | 3mm                 | 1.26    |                        |
|            | 4mm                 | 1.7     |                        |
|            | 5mm                 | 2.13    |                        |
| 2          | 2mm                 | 0.79    | 2.27                   |
|            | 3mm                 | 1.19    |                        |
|            | 4mm                 | 1.61    |                        |
|            | 5mm                 | 2.07    |                        |

Inputting a target clamping force of 8kN to the EMB system, the differences between the experimental motor speed and the theoretical motor speed are analyzed. It can be seen from Figure 4 that when the brake clearance is adjusted, the theoretical motor speed reaches 900 r/min within 0.075 s. The test and theoretical motor speed are reduced to 88 rpm/min at 0.15s and at 0.13s respectively, and are close to 0 at 0.24s, entering the stall phase. The change of brake clamping force is also analyzed within the same amount of time. It can be seen from the figure 5 that no clamping force is generated and the stroke of the brake block continues to increase at 0.075s. The brake clearance is eliminated at 0.24s, and the clamping force increases rapidly with the increase of the travel of the brake pad. The clamping force reaches the target at 0.205s, and the brake clearance is eliminated within 0.13s. The experimental and theoretical brake elimination time differ little, the differences may be caused by the mechanical properties of the motor itself.
4. Conclusion
The mechanical relationship between each component of EMB is analysed and calculated according to the relevant regulations and standards. After the EMB brake pressure and EMB brake clearance elimination time and brake clamping force are analysed, it is discovered that there is little difference between the experimental value and the theoretical value. And the errors may be caused by the installation error and its own properties. Therefore, the EMB system designed in this study can be used in automotive electronic mechanical control. However, the stability of the brake efficiency of the EMB system isn’t predicted in this test yet, and the results of the effects of the friction pair properties and brake torque fluctuations on the brake vibration noise during brake are inconclusive. In our follow-up tests, the brake efficiency stability of the EMB system will be conducted, and the impact of the friction pair properties and brake torque fluctuations on the brake vibration noise of the EMB system during brake will be verified further. In summary, the EMB structure is designed based on the advantages of the EMB brake system in this study, and the result is close to the theoretical value.

Acknowledgments
This work was financially supported by Guang’an Vocational and Technical College Research Projects in 2019: Research and application of intelligent control system for lifting mechanism of dump truck (GAZYKY-2019A01); Sichuan education informatization application and Development Research Center Project (Project No: JYXX20-029)

References
[1] V. Žuraulis, G. Garbincius, P. Skackauskas, et al. Experimental Study of Winter Tyre Usage According to Tread Depth and Temperature in Vehicle Brake Performance. Iranian Journal of Science and Technology-Transactions of Mechanical Engineering. 44(2020) 83-91.
[2] Y. F. Fu, X. H. Hu, W. R. Wang, et al. Simulation and Experimental Study of a New
Electromechanical Brake with Automatic Wear Adjustment Function. Int J Automot Technol. 21(2020) 227-238.

[3] P. Wasilewski. Frictional Heating in Railway Brakes: A Review of Numerical Models. Archives of Computational Methods in Engineering. 27(2020) 45-58.

[4] Y. Boukadida, F. Marignetti, G.M. Casolino, et al. Emulation and Testing for Automotive Propulsion Drive Using Two Cascaded Inverters. IEEE Transactions on Industry Applications., 56(2020) 1766-1783.

[5] A. Harifi, F. Rashidi, F. V. Takaloo, et al. Design of an Adaptive Fuzzy Controller for Antilock Brake systems. International journal of automotive engineering. 10(2020) 3158-3166.

[6] M. Moradian, V. Modanloo, S. Aghaiee, et al. Comparative analysis of multi criteria decision making techniques for material selection of brake booster valve body. Journal of Traffic and Transportation Engineering. 6(2019) 526-534.

[7] S. Venkatesh, K. Murugapoopathiraja. Scoping Review of Brake Friction Material for Automotive. Materials Today: Proceedings. (2019) 927-933.