Research on Electronic Brake Force Distribution and Anti-Lock Brake of Vehicle Based on Direct Drive Electro Hydraulic Actuator

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ABSTRACT In order to effectively improve the vehicle safety, this paper proposed an electronic brake force distribution scheme and an anti-lock brake scheme for passenger car based on a novel brake-by-wire system. The brake-by-wire system was introduced at first, then the electronic brake force distribution scheme and anti-lock brake scheme are designed after analyzing of brake system and vehicle model. At last, the experiment is accomplished to prove the feasibility of the novel brake-by-wire actuator, and the co-simulations of vehicle based on Matlab/Simulink and AMESim are accomplished for typical braking process to prove the superiority of proposed brake-by-wire system and brake scheme. On the one hand, the experiment and simulation results show that the proposed electronic brake force distribution scheme is able to distribute the braking force according to braking intensity and conditions. So the \( \beta \) curve can as close as possible to the \( I \) curve. On the other hand, the anti-lock brake scheme is able to accurately regulate the wheel slip rate and adapt to different brake intensity. So the anti-lock brake scheme processes strong robust performance.

KEY WORDS: brake-by-wire, linear motor, electric brake force distribution, anti-lock brake, fuzzy control (B1)

1. Introduction

Brake-by-wire (BBW) system is one of the key safety technologies since the invention of anti-lock brake system (ABS), because it provides better performance for passenger cars than traditional hydraulic brake system \( (1) \). At present, two main types of BBW system have been extensively studied, which are electro-hydraulic brake (EHB) and electro-mechanical brake (EMB) \( (2, 3) \). Since the mechanical and hydraulic link parts between the brake pedal and wheel actuator are abandoned, the brake signals and electrical energy are transferred by wires and cables in BBW system. Therefore, the braking force distribution between four wheels is no longer restricted by the hydraulic structure, the brake force of every wheel are able to be adjusted independently and the braking process of vehicle can be flexibly adjusted \( (4) \). Although the performance of BBW is higher than hydraulic brake system, the braking performance and direction stability depend not only on the BBW performance, but also on the braking force distribution scheme and anti-lock brake scheme \( (5) \). In addition, the regenerative efficiency of electric vehicle also depends on the braking force distribution scheme \( (6) \).

In order to take full advantage of BBW, Yong et al. designed a braking force observer and a robust slip rate controller for EHB to effectively improve the wheel’s anti-lock braking performance \( (7) \). Michael et al. also developed an anti-lock braking strategy for EHB, which keeps the slip rate at the optimal value whether the road is ice, snow or wet \( (8) \). For EMB, Yang et al. designed the electronic braking force distribution (EBD) scheme that can ideally distribute the braking force between the front and rear axles, and proposed the fuzzy ABS algorithm that can accurately control each wheel’s slip ratio and has strong robustness \( (9) \). Daegun et al. designed the wheel slip rate controller by utilizing BBW actuator and sliding mode control method, and proposed the slip rate distribution scheme to maintain vehicle stability based on the fuzzy logic controller and direct yaw moment controller \( (10) \). In addition, the vehicle state observer and slip-rate controller were designed by Andre et al. to improve the braking stability \( (11) \). The generic continuous time model predictive control (MPC) algorithm for vehicle was proposed by Chris et al. to enhance the anti-lock braking performance of EMB \( (12) \).

In order to further improve the braking performance and braking stability, a novel BBW actuator based on linear motor is designed in this paper, and a complete BBW system is constructed based on the actuator. Because the proposed BBW system has the advantages of integrating EBD and ABS, the EBD strategy based on ideal braking force distribution and the ABS strategy based on fuzzy controller are studied in detail. Subsequently, the feasibility and performance of BBW actuator are verified by simulation and experiment. Finally, the co-simulation model based on Simulink and AMESim is established to verify the EBD and ABS performance under typical braking condition.

2. The Brake-by-Wire System

2.1. Direct-drive electro-hydraulic brake actuator

The brake-by-wire actuator proposed by our team is driven by a linear motor and is named as direct-drive electro-hydraulic brake (DDEHB) actuator \( (13, 14) \), as shown in Fig. 1.

In the Fig. 1, 1 is the inner core of line motor, 2 is the shell, and 8 is the end cover, they are all made of low carbon steel and have good magnetic permeability. The ring-shaped permanent magnets 3 and arc-shaped permanent magnets 5 arranged in Halbach array are attached to the inner surface of shell. The magnetic field direction of the ring-shaped magnet is axial, the direction of the arc-shaped magnet is radial, which are shown in the Fig. 1. Between the magnets and the inner core is coil bobbin 6 that moves axially and has two coils 4 and 7 wound in opposite direction. The current directions during braking are shown in Fig. 1 (a) and (b). The plunger 14 is connected to the coil through a pin 15 and moves with the coil. After receiving the brake signal, the energized coil moves to left due to the electromagnetic force, thereby pushing the plunger to compress the brake fluid in the unequal-diameter hydraulic cylinder 13. Subsequently, the compressed brake fluid pushes the piston 12 to the left. The caliper 9 is rigidly connected to the motor body. The two friction linings 10 are arranged symmetrically on both sides of the brake disc 11. The left friction lining is fixed to the caliper 9 and the right friction lining is fixed to the piston.

After eliminating the braking gap, the positions of the actuator moving parts are shown in Fig. 1 (b). The friction linings
and the brake disc are strongly pressed by the piston. Because the brake disc rotates with the wheel and the brake linings are fixed, the friction force is generated between the disc and friction linings, which slows down the wheel. If the brake state needs to be released, the coil is energized in the reverse direction and the moving parts return to the original position, as shown in Fig. 1 (a).

2.2. The vehicle BBW system based on DDEHB

The vehicle braking system composed of DDEHB is shown in Fig. 2. It consists of brake pedal simulator, four actuators, controllers, drivers and sensors. The brake pedal simulator converts the driver’s actions into electronic signal. And other signals such as velocity, wheel speeds, steering angle and braking force are also converted into electronic signals at the same time. All these signals are delivered to brake system controller. After the controller analyzes and processes these signals comprehensively, the braking force distribution strategy allocates the total braking force to each actuator controller. Thereafter, the actuator controller adjusts the coil current to provide appropriate braking force for every wheel based on the received force signal.

DDEHB utilizes linear motor to convert electrical energy into electromagnetic force, so the structure is simpler than EMB because the motion conversion mechanism is not necessary in DDEHB\(^{(9)}\). The main advantages are:

- The brake force is provided by the linear motor without motion conversion mechanism, so the braking force responds quickly and the control accuracy is high.
- The unequal diameter hydraulic cylinder is used to reduce the size and weight of actuator, so DDEHB is compact and suitable for various vehicles.
- The vacuum booster, hydraulic pipelines, ABS module and other devices are cancelled, so the structure of the brake system is simpler.
- The brake force is independently and flexibly adjusted by DDEHB actuator, so it is easier to adjust the braking process, such as ABS control and vehicle stability control (VSC).

3. Dynamic Model of DDEHB and Vehicle

3.1. Model of DDEHB

The electric circuit of linear motor can be equivalent to a closed circuit consisting of inductor, resistor and counter electromotive force. According to Kirchhoff’s voltage law, the total voltage of closed circuit is zero, which is expressed by the equation (1).

\[
U(t) = E + Ri(t) + L \frac{di(t)}{dt}
\]

(1)

Where: \(i\) is the coil current; \(R\) is the coil resistance; \(U\) is the motor voltage; \(L\) is the coil inductance; \(E\) is the counter electromotive force which is calculated as equation (2).

\[
E = BIN \frac{dx(t)}{dt}
\]

(2)

Where: \(B\) is the magnetic induction intensity; \(N\) is the coil turns; \(l\) is the coil lap length.

The energized coil is subjected to electromagnetic force and moves along the axis. The direction of electromagnetic force is determined by the Faraday’s left-hand rule, and the value is determined by equation (3).
\[ F_e = B I N(t) \]  

The plunger is subjected to electromagnetic force and hydraulic force, which motion is expressed as equation (4). 

\[ F_e = m \frac{d^2x(t)}{dt^2} + C \frac{dx(t)}{dt} + F_d \tag{4} \]

Where: \( m \) is the total mass of coil and plunger; \( x \) is the coil displacement; \( C \) is the coil damping coefficient; \( F_d \) is the equivalent hydraulic force acted on the plunger.

If the secondary factors are ignored, such as process pressure loss and local pressure loss of hydraulic liquid. The model of unequal diameter hydraulic cylinder can be simplified to:

\[ \frac{F_D}{F_d} = \frac{D^2}{d^2} \tag{5} \]

Where: \( F_D \) is the force acted on piston; \( D \) and \( d \) are the piston diameter and plunger diameter respectively.

Finally, the braking torque acting on the wheel is:

\[ M = 2fF_dR \tag{6} \]

Where: \( f \) is the friction coefficient between the braking lings and brake disc, \( R \) is the effective radius of brake disc.

3.2. Dynamics model of vehicle

The external forces acted on the wheel in longitudinal and vertical during braking are shown in Fig. 3, including vertical ground reaction force \( F_Z \), wheel load \( W \), ground braking force \( F_X \), braking torque \( M \), and axle reaction force \( F_P \).

The braking torque \( M \) is provided by DDEHB actuator. With the effect of braking torque \( M \) and ground braking force \( F_X \), the dynamic model of wheel is described by equation (7):

\[ M - R_t F_X = J_t \frac{d\omega_t}{dt} \tag{7} \]

Where: \( R_t \) is the rolling radius of wheel; \( J_t \) is the wheel inertia; \( \omega_t \) is the wheel speed.

The force acted on the vehicle during braking are shown in Fig. 4. If the vehicle vibration, roll, pitch and other secondary factors are ignored, and the height of vehicle gravity center is \( h \), steering angle is \( \alpha \), then the vehicle dynamics model is simplified as equation (8) to (13)\(^{16,17}\).

\[ m(\ddot{v} - \nu \dot{v}) = (F_{X1} + F_{X2}) \cos \alpha + (F_{Y1} + F_{Y2}) \sin \alpha + F_{X3} + F_{X4} \tag{8} \]

\[ m(\dot{v} + \nu \ddot{v}) = (F_{Y1} + F_{Y2}) \cos \alpha - (F_{X1} + F_{X2}) \sin \alpha + F_{Y3} + F_{Y4} \tag{9} \]

\[ mg = F_{Z1} + F_{Z2} + F_{Z3} + F_{Z4} \tag{10} \]

\[ -m(\ddot{v} + \nu \dot{v})h = (F_{Z1} - F_{Z2}) \cdot \frac{b_f}{2} + (F_{Z3} - F_{Z4}) \cdot \frac{b_r}{2} \tag{11} \]
\[ m(\dot{u} - \nu r)h = -(F_{x1} + F_{z2})l_f + (F_{x3} + F_{z4})l_r \]  
\[ I_{x\parallel} = \frac{b_f}{2}[(F_{x1} - F_{x2}) \cos \alpha + (F_{y1} - F_{y2}) \sin \alpha] + \frac{b_r}{2}(F_{x3} - F_{x4}) + \frac{l_f}{2}[(F_{y1} + F_{y2}) \cos \alpha - (F_{x1} + F_{x2}) \sin \alpha] - l_r(F_{y3} + F_{y4}) \]

Where: \( m \) is the vehicle mass; \( u \) is the longitudinal velocity; \( \nu \) is the lateral velocity; \( r \) is the yaw rate; \( F_{x1} \) is the longitudinal adhesion force of i wheel; \( F_{y1} \) is the lateral adhesion force of i wheel, \( l_f \) is the vertical force of i wheel, \( l_r \) is the horizontal distance between the front axle and gravity center; \( l_t \) is the horizontal distance between the rear axle and gravity center; \( b_l \) is the front wheel base, \( b_r \) is the rear wheel base; \( I_{x\parallel} \) is inertia moment of vehicle body around the z axis

\[ F_{x\parallel} = \frac{1}{2} \left[ \frac{G}{h} \left( l_t^2 + \frac{4(l_t + l_r)h}{G} F_{x\parallel} \right) - \left( \frac{G l_r}{h} + 2F_{x\parallel} \right) \right] \]

Do not consider the steering temporarily, the braking force allocated on front and rear axle will be evenly distributed to the left and right wheels:

\[ F_{x1} = F_{x2} = \frac{F_{x\parallel}}{2} \]

\[ F_{x3} = F_{x4} = \frac{F_{x\parallel}}{2} \]

Whether to control the slip-rate depends on the wheels status, the ABS controller will be activated if the slip-rate of any wheel is greater than 0.5. thereafter, the slip-rate is regulated back to its ideal value or set value by adjusting the DDEHB current \( I_{EF} \).

\[ x_1 = \Delta s = s - 0.2 \]
\[ x_2 = \frac{ds}{dt} \]
\[ s = \frac{u - \omega_t R_t}{u} \]

The output of controller is the increment of braking force \( \Delta F \), the structure of ABS controller is shown in Fig. 5.

According to equation (22), the variation range of slip ratio \( s \) is: 0 < \( s \) < 1, so the domain of \( x_2 \) is (-0.2, 0.8). And according to experience, the domain of \( x_2 \) is (-15, 15), the domain of \( \Delta F \) is (-90, 90). The fuzzy controller uses fuzzy mind similar to human brain to regulate the slip rate. In other words, the further the values of \( x_1 \) and \( x_2 \) are from the stable value 0, the value of \( \Delta F \) is greater.

Defines the fuzzy set: [NB, NM, NS, ZE, PS, PM, PB] = [negative big, negative median, negative small, zero, positive small, positive median, positive big], and fuzzification map \( x_1 \), \( x_2 \) and \( \Delta F \) to the above 7 fuzzy set. The fuzzy controller uses Mamdani fuzzification inference and gravity anti-fuzzification inference to
calculate the output, and the designed fuzzy control surface is shown in Fig. 6 [22].

![Fuzzy control surface](image)

**Fig. 6 Fuzzy control surface**

The output ΔF is responsible for adjusting the wheel slip rate. If the slip rate is always maintained at its optimal value, the actual deceleration will be greater than the required deceleration during normal braking. Therefore, the actual braking force acted on the wheels is shown in Fig. 7, the minimum value should be taken from the output of slip rate controller and braking force distribution controller.

**Fig. 7 Braking force selection**

The force tracking performance is verified in this paper. Due to the small space of caliper, it is not possible to directly measure the clamping force using large-sized force sensor, so the pressure sensor is used to measure the hydraulic pressure of cylinder, thereby indirectly measuring the clamping force. During simulation and experiment, the target of hydraulic pressure is a sine wave signal, which the maximum value is 6 MPa, the minimum value is 0, and the cycle time is 1s.

![Tracking performance of DDEHB](image)

**Fig. 8 Tracking performance of DDEHB**

5. Simulation and Experiment of EBD/ABS

5.1. Simulation and experiment of DDEHB

In order to verify the feasibility of the braking scheme proposed in this paper, the performance of DDEHB actuator was first verified by simulation and experiment. The caliper of DDEHB used in the experiment was modified from a traditional hydraulic brake unit. The linear motor was machined according to Fig. 1, and then the caliper and linear motor were rigidly connected as a whole by bolts. The main parameters of the DDEHB and linear motor are shown in Table 1 and Table 2 respectively. At the same time, the Simulink model is also established to simulate the performance of DDEHB [23].

| Parameter          | Value     |
|--------------------|-----------|
| Piston diameter (mm)| 38        |
| Plunger diameter (mm)| 6         |
| Max pressure (MPa)  | 10        |
| Max braking force (N) | 10343 (μs=0.38) |
| Max piston pressure (N) | 27219     |
| Electromagnetic force (N) | 339       |

**Table 1 Parameters of DDEHB**

| Parameter          | Value     |
|--------------------|-----------|
| EMLA diameter (mm) | 60        |
| EMLA length (mm)   | 70        |
| EMLA voltage (V)   | 24        |
| Coil resistance (Ω) | 0.7615    |
| Coil inductance (μH) | 279.8     |
| Peak current (A)   | 20        |

**Table 2 Parameters of linear motor**

The experiment results agree well with simulation results, as shown in Fig. 8. And the actual pressure tracks the target pressure well whether it is simulation or experiment.

![The enlarged view of the local area](image)

**Fig. 9 The enlarged view of the local area**

The experiment results agree well with simulation results, as shown in Fig. 8. And the actual pressure tracks the target pressure well whether it is simulation or experiment.

Fig. 9 is the enlarged view of the local area in Fig. 8. The Fig. 9 shows that there is a certain deviation between the simulation results and the test results when the pressure is very low, because the simulation model does not accurately reflect the friction of actuator. However, the hydraulic pressure is very low in this case, which beyond the normal working conditions of actuator, so it has little impact on the DDEHB performance.

5.2. Simulation of Vehicle

In order to verify the proposed braking scheme, a co-simulation model based on Simulink and AMESim was established. Simulink is used to construct EBD/ABS strategy, which distribute and adjust the braking force. AMESim is used to construct the
vehicle and DDEHB model. In the simulation, the road adhesion coefficient is set to 0.4, the target deceleration of the first 3 seconds is set to 2m/s² which verifies the performance of braking force distribution scheme, and the target deceleration after 3 seconds is changed to 5m/s² which verifies the performance of anti-lock braking scheme. The other main parameters used in the simulation are shown in Table 3.

| Parameter                        | Value  |
|----------------------------------|--------|
| Vehicle mass (Kg)                | 1430   |
| Front axle base (mm)             | 1210   |
| Rear axle base (mm)              | 1230   |
| Gravity height (mm)              | 490    |
| Target slip-rate                 | 0.2    |
| Brake disc effective radius (mm) | 95     |
| Wheel rolling radius (mm)        | 280    |
| friction lining factors          | 0.38   |
| Initial velocity (m/s)           | 30     |

Table 3 Simulation parameters

The velocity and deceleration are shown in Fig. 10, the velocity changes evenly and smoothly whether in light braking or heavy braking. The actual deceleration was accurately maintained at 2m/s² in the first 3 seconds, indicating that the deceleration controller accurately controls the braking strength. The actual deceleration attempts to reach 5m/s² after 3 seconds, but the road does not allow because the adhesion coefficient is only 0.4. The anti-lock controller was activated in time when controller detects that the wheels are going to be locked, so the actual deceleration was adjusted and maintained at 4 m/s².

The speed curve changes evenly in overall view, as shown in Fig. 11. But it decreased rapidly at 3 second because the target deceleration changes to 5m/s², especially the front wheel speed decreases more rapidly even to be locked. After the ABS being activated, the wheel speeds are adjusted to normal and uniformly descend until the end of brake process.

Both front and rear slip-rate are low and stable in the first 3 seconds, as shown in Fig. 12, indicating that the vehicle is in a stable braking conditions. The slip-rates increase rapidly after 3 second due to the abruptly change in the target deceleration, especially the slip-rate of front wheel quickly exceeded the set threshold of 0.5. Then the ABS is activated, both the front and rear slip-rate are adjusted to and maintained at the set value of 0.2.

The brake force provided by DDEHB are shown in Fig. 13. After 3 seconds, the braking force increase quickly result in the wheels being locked. After that, the ABS is activated and the braking force is adjusted back to the optimal value of current road.

The actual braking force distribution curve (β curve) and ideal braking force distribution curve (I curve) are shown in Fig. 14. The β curve should be below and as close as possible to I curve. The actual braking force changes along the arrow, where the braking force distribution ratio in first 3 seconds and after 3 seconds are mainly concentrated in area 1 and area 2 respectively. Both regions are below and close to I curve, which indicates that the proposed distribution scheme is reasonable and feasible. In the Fig. 14, a part of β curve is higher than I curve, which corresponding time is 3.24s - 3.37s, it is the braking force adjustment stage after the anti-lock is activated and the effect on
the braking performance can be ignored because the adjustment time is very short.

![Diagram of braking force distribution curve](image1)

*Fig. 14* Braking force distribution curve

![Diagram of deceleration and slip-rate](image2)

*Fig. 15* Deceleration and slip-rate of conventional hydraulic brake system with logic threshold control

For better comparison, the conventional hydraulic brake system and DDEHB system with logic threshold control were simulated under the same braking conditions with same vehicle parameters, same initial velocity and same braking strength. The deceleration and slip-rate of these two system are shown in Fig. 15 and Fig. 16. Compared with Fig. 10 and Fig. 12, there are no significant differences between the three brake systems before the ABS is activated. However, they are significantly different once the ABS is activated. The DDEHB system with proposed EBD and ABS scheme in this paper accurately controls and stabilizes the slip-rate, so the deceleration and slip-rate are very stable although in the anti-lock braking condition. However, the other two braking schemes adopt logic threshold control strategy, and the pressure in wheel cylinder continue in “buck-keep-boost-keep” cycle to prevent the wheels from locking. Therefore, the deceleration, wheel speed and slip rate change dramatically. In addition, Fig. 15 and Fig. 16 show that the deceleration and slip rate of DDEHB are not as dramatic as those of conventional hydraulic system although both systems adopt logic threshold control methods, because DDEHB responds faster and adjusts more accurately than hydraulic brake system. The final braking distance and braking time are listed in Table 4, the braking distance and braking time of proposed scheme are shortened 6.25% and 11.36% compared with conventional hydraulic brake system with logic threshold control; and they are shortened 4.06% and 5.47% compared with DDEHB system with logic threshold control.

### Table 4 Slip-rate of DDEHB with logic threshold control

| Braking system | Braking distance (m) | Braking time (s) |
|----------------|----------------------|------------------|
| A              | 161.76               | 9.95             |
| B              | 158.07               | 9.33             |
| C              | 151.65               | 8.82             |

Improvement (C vs. A) 6.25% and 11.36%.

Improvement (C vs. B) 4.06% and 5.47%.

6. Conclusion

(1) The EBD and ABS brake scheme based on DDEHB actuator are proposed, and the braking process is studied and analyzed, which is of great significance for the applying of BBW in vehicle.

(2) The simulation results show that the EBD scheme is able to distribute the braking force as close as possible to I curve and the ABS scheme is able to adjust the slip-rate on its target value.

(3) The simulation results under the same braking conditions show that the proposed scheme effectively reduces the braking time and braking distance, compared with conventional hydraulic brake system with logic threshold control and DDEHB with logic threshold control.

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