

Article

Design and Implementation of Active Brake Pedal Simulator Integrating Force Feedback and Energy Optimization

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Abstract

Brake pedals and wheel braking units are mechanically decoupled in brake-by-wire systems. This causes the driver to lose the familiar pedal feel. To address this issue, this paper designed an active braking pedal simulator based on the long-travel Halbach-array linear motor. Firstly, this paper conducted both qualitative and quantitative analyses on the pedal characteristics of a traditional hydraulic braking system and used them as a reference. A dual-coil independent control strategy was designed in order to overcome the thrust instability at the junction of the Halbach-array magnetic field. This enables the linear motor to achieve smooth and continuous thrust output throughout the entire travel range. Secondly, this paper also designed a “linear motor + spring” solution to reduce energy consumption and peak motor thrust. By conducting a quantitative analysis of the relationship between the spring stiffness, motor work and peak thrust, the spring stiffness was optimized. The results show that when the spring stiffness is 3.73 N/mm, the motor work can be reduced to 5.92 Joules while significantly reducing the peak thrust. Finally, this paper also established a testing platform. It was used to verify the performance of the proposed pedal simulator under low-intensity, medium-intensity, and high-intensity braking conditions as well as an anti-lock braking system intervention. The testing results show that the pedal simulator can actively adjust the pedal characteristics according to the braking intensity, and it can provide clear vibration feedback during the anti-lock braking system intervention. Therefore, the proposed pedal simulator effectively simulates the pedal feel of hydraulic braking systems while improving energy efficiency and operational stability. It provides a feasible solution for enhancing the driver–vehicle interaction and the driving comfort of brake-by-wire systems.



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Keywords: brake-by-wire; brake pedal simulator; linear motor; spring compensation; pedal feel feedback

1. Introduction

Electric vehicles (EVs) are gradually replacing conventional internal combustion vehicles and have become one of the primary means of transportation [1,2]. This change will reduce the consumption of gasoline and diesel, thereby decreasing pollutant emissions [3].

In electric vehicles, as internal combustion engines are replaced by electric motors, vacuum-assisted hydraulic braking systems are also being replaced by electric brake-by-wire (BBW) systems [4,5].

BBW systems have many advantages. However, the mechanical decoupling between the brake pedal and wheel brake units prevents the driver from experiencing intuitive and realistic pedal feel feedback during braking [3,6]. For example, the brake pedal lacks the characteristic of “kickback” feedback during anti-lock braking system (ABS) intervention [7,8]. As the critical interface between the driver and vehicle, the brake pedal is essential for driving safety and comfort [9]. Therefore, addressing the issue of missing pedal feel in BBW systems is of great significance.

In hydraulic braking systems, the pedal undergoes certain travel after overcoming the resistance of the pedal linkage, vacuum booster and hydraulic system when the driver depresses the brake pedal. There is a corresponding relationship between the pedal force and pedal travel, which is known as the brake pedal characteristic curve, as shown in Figure 1 [10]. When the pedal travel is less than 10 mm, the braking clearance and the mechanism clearance are eliminated. The vacuum booster does not function during this stage, and the pedal force and pedal travel show a slight increase. When the pedal travel is between 10 mm and 40 mm, the vacuum booster begins to function. Due to the effect of the vacuum booster, the pedal travel increases rapidly, but the pedal force increases very slightly. When the pedal travel is between 40 mm and 80 mm, the assistance provided by the vacuum booster gradually reaches its maximum value, and both the pedal force and pedal travel increase simultaneously. When the pedal travel exceeds 80 mm, the vacuum booster is unable to provide any additional assistance. The force applied by the driver is directly transmitted to the master cylinder. As a result, the pedal force increases rapidly, but the pedal travel increases relatively slightly.

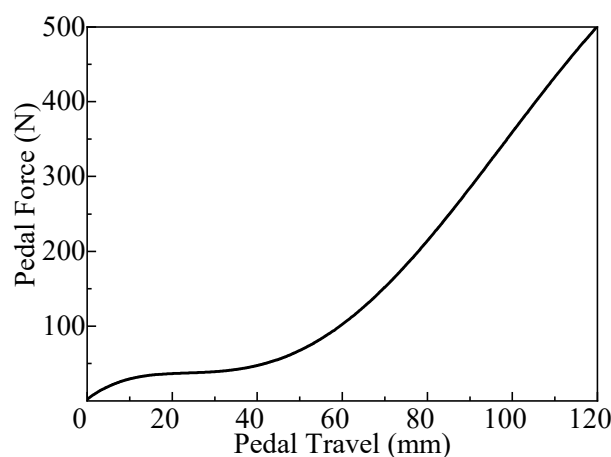


Figure 1. Brake pedal characteristic curve.

Brake pedal characteristics are the key parameters that must be carefully designed in braking systems, as they directly affect the driver’s perception and control of the brake system [11,12]. In hydraulic braking systems, the pedal characteristic curve involves multiple complex mechanical parameters, including brake fluid demand, master cylinder size, pedal leverage ratio, vacuum booster assistance ratio and initial operating force. Once a hydraulic braking system is designed, it is difficult to adjust or modify it. Therefore, a dedicated design must be developed for each vehicle model. The design process is also time-consuming, labor-intensive, and expensive [13,14].

The brake pedal and wheel braking unit are mechanically decoupled in BBW systems. The driver’s braking operation cannot be directly applied to the wheel braking unit through

mechanical or hydraulic components. Similarly, the wheel braking unit also cannot directly provide the driver with the appropriate brake pedal feel and braking status feedback [3]. For drivers, both the brake pedal feel and braking status feedback are of vital importance. Drivers need to perceive the state and response of the braking system [5].

To ensure driving safety, BBW systems often employ a pedal simulator to perform functions such as perceiving the driver's braking intention, converting braking signals, and simulating the brake pedal feel [15]. The brake pedal simulator precisely simulates the brake pedal characteristic curve, as shown in Figure 1, through specific mechanical structures and control methods, and it ensures that the driver can obtain a pedal feel similar to that of a hydraulic braking system. According to the adjustability of pedal feel, brake pedal simulators are divided into two types: passive pedal simulators and active pedal simulators [10,15]. Passive pedal simulators often use simple structures and components such as springs or rubber to simulate pedal force. Their advantages are their simple structure and low cost. However, passive pedal simulators lack flexibility. They can only simulate fixed pedal characteristic curves and cannot make changes or adjustments according to working conditions or vehicle models. They also cannot provide active feedback forces similar to those when ABS is involved [14,15].

Active pedal simulators can actively adjust the pedal force and pedal characteristics through the control system, and they can provide a braking experience that is closer to traditional hydraulic braking. Active pedal simulators can also simulate more diverse pedal sensations, meeting the personalized needs of different vehicle types and different vehicle weights [6]. Therefore, research on brake pedal simulators generally focuses on active pedal simulators [16,17]. Bosch designed an electric booster with a pedal behavior simulator, which provides braking assistance via a motor and precisely detects the axial movement of the pedal through the pedal behavior simulator, offering the driver a pedal feel consistent with the vehicle's state [3]. Magnetorheological damping pedal simulators utilize the rheological properties of magnetorheological fluid under the influence of a magnetic field to adjust the damping force. When a current passes through the magnetic field coil, the viscosity of the magnetorheological fluid changes, thereby altering the resistance encountered when the pedal moves. This scheme features a fast response, high accuracy, and has the ability to adjust the damping force in real time. It is highly suitable for applications that need dynamic force feedback [18,19]. Electromagnetic spring pedal simulators combine electromagnetic force and spring force to provide pedal force feedback. The electromagnet generates attractive or repulsive forces based on the control signal, and it simulates the reaction of the pedal together with the spring force. Its structure is simple, its cost is low, and it can easily implement force feedback [10,14].

In summary, passive pedal simulators have a simple structure but cannot achieve dynamic or active adjustment of the pedal feel. Active pedal simulators offer higher controllability and the ability for active adjustment. Therefore, this paper designs an active brake pedal simulator that integrates force feedback and energy optimization based on a linear motor. The objective is to achieve realistic and adjustable pedal feel feedback while ensuring a dynamic response, reducing energy consumption, and enhancing structural reliability. The paper first introduces the overall design and system architecture of the active pedal simulator. The dual-coil independent control method is proposed to address the thrust instability of the Halbach-array linear motor. Then, the power consumption characteristics of the "linear motor-spring" structure are analyzed, and the spring stiffness is optimized to achieve the best match. Finally, a pedal simulator test platform is established to conduct experimental validation under varying braking intensities and ABS intervention conditions. This paper provides a novel approach to addressing the issue of missing pedal feel in BBW systems and is of significant importance for improving driving experience and vehicle safety.

2. Design Scheme of Brake Pedal Simulator

2.1. Overall Scheme of Brake-by-Wire System

A typical BBW system is shown in Figure 2. Firstly, the brake pedal simulator electronic control unit (ECU) analyzes and calculates the driver's braking requirements (target deceleration) based on collected pedal signals such as pedal travel or pedal force. Then, the target deceleration is transmitted to the vehicle ECU for calculation and distribution of the braking force for the entire vehicle. Then, the target braking force allocated to each wheel is transmitted to each wheel control unit. Finally, every wheel braking unit flexibly adjusts its brake force under the control of its own controller. The braking force and braking status of each wheel are also fed back to the pedal simulator ECU, and the pedal feedback force can be actively adjusted as needed.

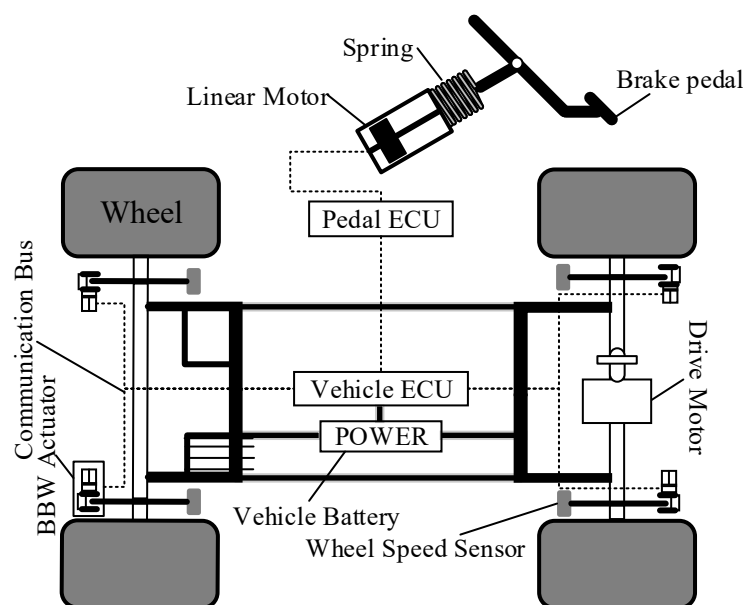


Figure 2. The overall scheme of a brake-by-wire system.

2.2. Brake Pedal Simulator Solution

The brake pedal is connected to a vacuum booster through a connecting rod in a hydraulic braking system. The driver steps on the brake pedal when there is a need for braking. The brake pedal rotates around the fulcrum, transmitting the pedal force applied by the driver to the vacuum booster through the connecting rod. The brake pedal simulator needs to accurately simulate the above process. Many pedal simulators use rotating motors in combination with gear mechanisms and ball screw mechanisms to simulate the above process.

In this paper, a long-travel linear motor is designed based on a Halbach array structure and is applied to a brake pedal simulator, as shown in Figure 3. Linear motors can generate linear thrust without the need for additional motion conversion mechanisms, thus significantly simplifying the structure of the pedal simulator. In addition, the linear motor has high position accuracy and force control accuracy, and it also possesses fast response characteristics, enabling high-speed and high-precision dynamic control. Therefore, the linear motor can accurately and rapidly simulate the brake pedal characteristic curve of traditional hydraulic braking systems.

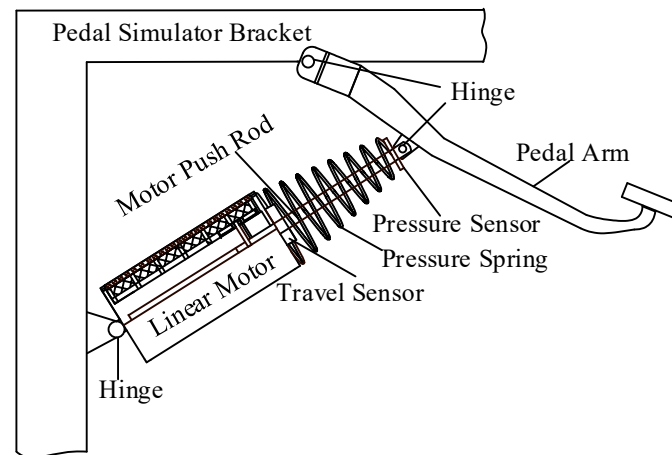


Figure 3. The structure of the active pedal simulator based on linear motors.

2.3. Linear Motor Scheme

The spatial magnetic field of permanent magnets can be optimized by arranging them in a Halbach array. The Halbach array significantly enhances the magnetic field strength on one side of the coil and weakens the magnetic field on the other side, thereby enabling more efficient utilization of magnetic energy and magnetic yoke materials [20–23]. The special structure of the Halbach array also causes the magnetic field directions of the upper and lower parts of coil to often be in opposite states, as shown in Figure 4a. The assembled permanent magnet and motor housing are shown in Figure 4b. When the linear motor moves a long distance, the coil needs to continuously pass through the reverse magnetic field during its movement. When the coil passes through the boundary area of the magnetic field loop, the output force becomes unstable due to the sudden change in the direction of the magnetic field. This may even cause sudden changes in the direction of motion or mechanical vibrations, which will seriously affect the operational stability and accuracy. To address the above issues, the moving winding adopts a double-coil design. Two sets of coils are independently wound on the same coil frame and controlled, respectively, as shown in Figure 4c. This dual-coil independent control scheme was developed to suppress thrust fluctuations. When the mover coil crosses the magnetic field boundary, the control system precisely determines the coil position. Then, whether the two sets of coils are energized and the direction of energization are determined based on the direction of the magnetic field. After experimental verification, this control method has enhanced adaptability and reliability in complex magnetic fields, and the linear motor can achieve smooth operation throughout the entire length of travel. The two sets of coils are denoted as coil A and coil B respectively, as shown in Figure 4c. The positive current, negative current and power-off are denoted as “+”, “−” and “0” respectively. Based on these definitions, there are a total of nine operating conditions for linear motors, as summarized in Table 1. The parameters of the final designed linear motor are shown in Table 2.

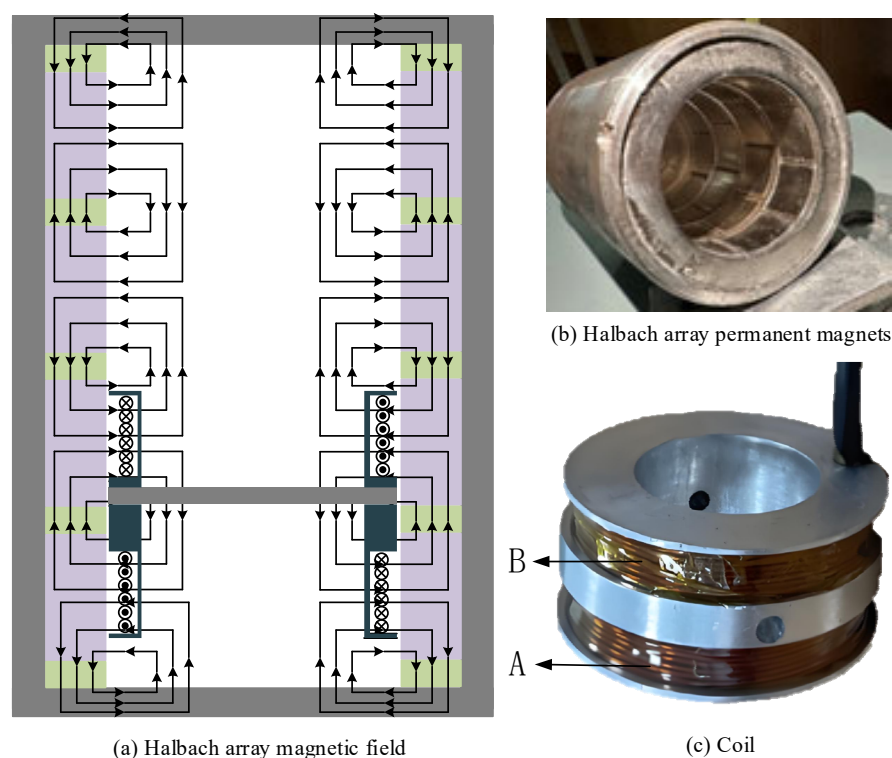


Figure 4. Linear motor scheme.

Table 1. The working state of the linear motor.

Working Condition	Coil A	Coil B	Description (Based on Magnetic Field Direction)
1	+	+	Both coils energized in the same polarity, generating greater thrust.
2	+	−	Coils energized with opposite polarities, generating greater thrust.
3	+	0	Coil B is located at the boundary of the magnetic field and is not energized, generating smaller thrust.
4	−	+	Coils energized with opposite polarities, generating greater thrust.
5	−	−	Both coils energized in the same polarity, generating greater thrust.
6	−	0	Coil B is located at the boundary of the magnetic field and is not energized, generating smaller thrust.
7	0	+	Coil A is located at the boundary of the magnetic field and is not energized, generating smaller thrust.
8	0	−	Coil A is located at the boundary of the magnetic field and is not energized, generating smaller thrust.
9	0	0	Neither of the two coils are energized, so the linear motor does not operate.

Table 2. Parameters of linear motor.

Parameter	Value
Type	Cylindrical permanent magnet moving coil
The number of closed magnetic field circuits	5
Power voltage	DC 24 V
Mover travel	90 mm
Coil number	2
Peak thrust	350 N
Peak current	35 A
Full-travel response time	380 ms
The time to reach maximum thrust	166 ms

2.4. Spring Replenishment Scheme

The linear motor thrust is formed by the superposition of two coil thrust sets. When the two sets of coils provide thrust in the same direction, the linear motor thrust is relatively large. However, due to the particularity of the Halbach array, it will cause the two sets of coils to alternately pass through the intersection area of the two magnetic field directions

when the coil moves. At this point, the linear motor thrust will fluctuate sharply if both of the two sets of coils are still energized simultaneously. When the coils pass through the area where the magnetic fields converge, the two sets of coils must be alternately energized according to their positions to ensure that the linear motor operates smoothly. The test results show that the linear motor's stable peak thrust is between 220 N and 350 N when the coil is at different positions. This thrust is relatively small and cannot provide the driver with sufficient pedal reaction force during high-intensity braking or emergency braking. To solve this problem, a spring mechanism was introduced as a supplement. Finally, the "linear motor-spring"-type active pedal simulator was designed, as shown in Figure 3.

3. Power Consumption Optimization of the Pedal Simulator

3.1. Ideal Brake Pedal Characteristic Curve

In order to accurately simulate the characteristic curve of a traditional pedal, it is necessary to analyze the characteristic curve and derive its precise mathematical expression. The fitted data used is publicly available industry data [12,24]. The Polyfit function is based on the principle of least squares. It calculates the polynomial coefficients that best fit the given data and the order of the polynomial. In this paper, a fifth-degree polynomial fitting is adopted, and the curve obtained through fitting is:

$$Y_P = 5.3425 \times 10^{-8}x^5 - 2.52 \times 10^{-5}x^4 + 0.003907x^3 - 0.1999x^2 + 4.3499x + 2.2335 \quad (1)$$

where x denotes the pedal travel in millimeters (mm), and Y_P denotes the pedal force in newtons (N), as shown in Figure 1.

3.2. Spring Stiffness Optimization

After adopting the "linear motor-spring" as the active pedal simulator solution, it is necessary to optimize the appropriate spring stiffness in order to reduce the energy consumption of the pedal simulator. Figures 5 and 6 show the schematic diagrams of the pedal force with spring stiffness values of $K = 2.1$ N/mm and $K = 5$ N/mm respectively. The solid line in the figure represents the pedal characteristic curve, and the dashed line represents the spring force. If the spring force is lower than the pedal characteristic curve, linear motor thrust is required; if the spring force is higher than the pedal characteristic curve, linear motor pulling force is required.

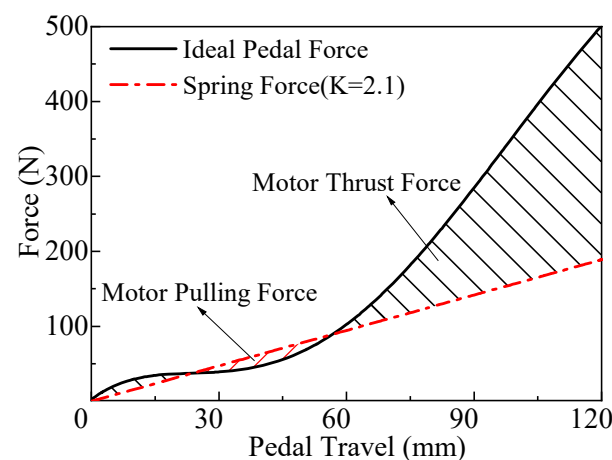


Figure 5. Schematic diagrams of the pedal force with a spring stiffness of $K = 2.1$ N/mm.

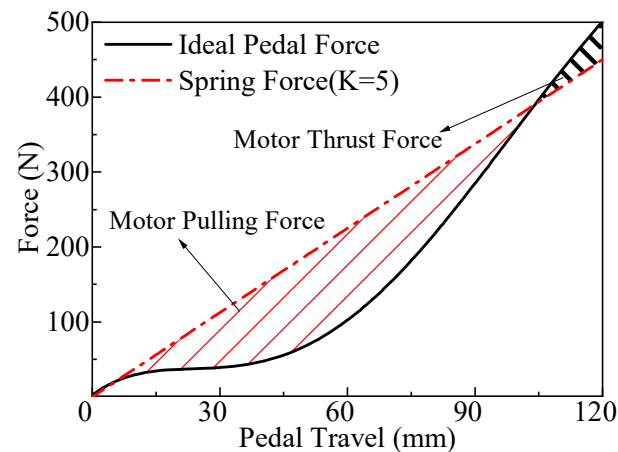


Figure 6. Schematic diagrams of the pedal force with a spring stiffness of $K = 5 \text{ N/mm}$.

As shown in Figures 5 and 6, the shaded area enclosed by the spring force and pedal characteristic curve is the work provided by the linear motor. By comparing Figures 5 and 6, it can be seen that when the pedal characteristic curve remains fixed, the spring stiffness directly determines the work provided by the linear motor throughout its full travel. According to the technical data, the full travel of the brake pedal is usually 110 mm–130 mm, and we take the middle value of 120 mm as the design parameter [11]. In addition, the total travel of the linear motor is 90 mm. The spiral spring is installed coaxially with the motor push rod, and the compression amount of the spring is also 90 mm. Therefore, it can be determined that the ratio of the linear motor travel to the pedal travel is $3/4$, which means the lever ratio of the brake pedal is $3/4$. For ease of illustration and direct comparison, the spring compression amount is converted into the corresponding pedal travel, and both the spring force and the pedal characteristic curve are presented in the same coordinate system, as illustrated in Figures 5 and 6. The spring force in the figure is named the equivalent spring force. If the original stiffness of the spring is K and the stiffness of the equivalent spring force is A , then the relationship between K and A is:

$$A = \frac{3}{4}K \quad (2)$$

The equivalent spring force Y_S can be expressed as:

$$Y_S = Ax \quad (3)$$

After calculation, the x-coordinates of the intersection points of the ideal pedal characteristic curve and the equivalent spring force curve are 5.1 and 104.5 respectively if the spring stiffness is $K = 5$, as shown in Figure 6. Then, the work W that the linear motor needs to do within its full travel range is:

$$W = \int_0^{5.1} (Y_P - Y_S) + \int_{5.1}^{104.5} (Y_S - Y_P) + \int_{104.5}^{120} (Y_P - Y_S) = 9.50\text{J} \quad (4)$$

The intersection point x-coordinates of the equivalent spring force and the ideal pedal force curve are calculated in the same way when A takes different values. Then, the work W that the linear motor needs to do within its full travel is also calculated according to Equation (4). The specific results are shown in Table 3.

Table 3. The x-coordinates of different spring stiffnesses and the work provided by the linear motor.

Spring Stiffness K	2.1	3.33	3.47	3.6	3.73	3.87	4	5
Equivalent spring stiffness A	1.575	2.5	2.6	2.7	2.8	2.9	3	3.75
The first point x -coordinate x_1	24	13.4	12	12.2	11	9.6	10	5.1
The second point x -coordinate x_2	56.82	76.5	78.5	80.7	82.5	84.8	86.7	104.5
Work done by the linear motor $W(J)$	9.50	6.72	6.58	6.48	5.92	6.38	6.39	7.70

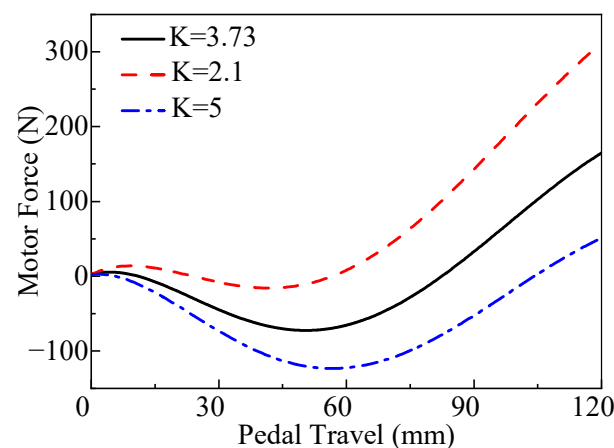
The linear motor work W and the equivalent spring stiffness A in Table 3 are fitted by Matlab 2020b. The fitting curve is:

$$W = 1845A^2 - 10,674A + 21,758 \quad (5)$$

The value of W is the smallest when $A = 2.8$ in Equation (5). That is, the power consumption of the linear motor is the lowest when $A = 2.8$. The corresponding spring stiffness is $K \approx 3.73$ mm/N, and the work W provided by the linear motor is approximately 5.92 J.

In addition to considering the work provided by the linear motor, the peak thrust of the linear motor also needs to be taken into account. On the one hand, severe heat generation will increase additional energy losses if linear motors operate in a high-thrust mode for long time. On the other hand, the peak thrust is an important performance indicator for linear motors, and the linear motor is more expensive if the peak thrust is higher. Therefore, it is necessary to avoid excessive thrust of the linear motor as much as possible.

Based on the previous power consumption analysis, springs stiffnesses K of 2.1 N/mm, 3.73 N/mm and 5 N/mm were respectively selected for analysis of extreme linear motor thrust, as shown in Figure 7. In the figure, when the value is positive, it indicates that the linear motor provides thrust; when the value is negative, it indicates that the linear motor provides pulling force. The force range and extreme values that the linear motor needs to provide under different spring stiffnesses are shown in Table 4.

**Figure 7.** Thrust curves of linear motors under different spring stiffnesses.**Table 4.** The thrust range and maximum value of linear motors under different spring stiffnesses.

Spring Stiffness K (N/mm)	2.1	3.73	5
Thrust range (N)	−15~311	−71~169	−123~48
Maximum thrust (N)	311	169	−123

It can be determined from Figure 7 and Table 4 that the maximum linear motor thrust is 311 N when the spring stiffness is 2.1 N/mm. This value is already close to the maximum

thrust of the linear motor, and the full-travel work is 9.50 joules. The maximum linear motor thrust is 169 N when the spring stiffness is 3.73 N/mm. This value is approximately half of the maximum linear motor thrust, and the full-travel work is 5.92 joules. The maximum linear motor pulling force is -123 N when the spring stiffness is 5 N/mm. This value is approximately one-third of the maximum linear motor thrust, and the full-travel work is 7.70 joules.

A spring with a stiffness of 3.73 N/mm was ultimately adopted after comprehensively considering the power consumption and peak thrust of the linear motor. At this point, the thrust provided by the linear motor can meet the requirements of the brake pedal simulator, and the work has reached the minimum of 5.92 joules. The optimized spring force curve and the work provided by the motor are shown in Figure 8. The area of the shaded part represents the work provided by the linear motor under this scheme.

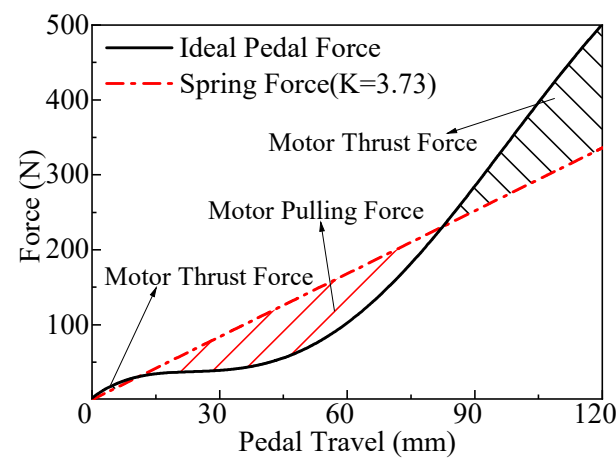


Figure 8. The optimized spring force curve and the motor work.

Adding appropriate springs in the pedal simulator can not only reduce the work and peak thrust of linear motor but also bring other advantages. On the one hand, the spring can still provide partial feedback force even if the linear motor malfunctions during braking. It will not cause the driver to completely lose the braking sensation. On the other hand, the spring can provide reset force to keep the brake pedal in its initial position when the pedal simulator stops working.

In conclusion, adding appropriate springs not only enhances the redundancy capacity of the pedal simulator but also reduces the power consumption of the motor, effectively minimizing heat generation and extending its service life.

4. Brake Pedal Simulator Test

4.1. Introduction to the Brake Pedal Simulator Test Platform

To verify the feasibility of the proposed scheme, a brake pedal simulator test platform was built, as shown in Figure 9. The fixed frame is made of carbon steel to simulate the fixed structure of the pedal simulator on the vehicle. A force sensor is installed at the position of the pedal. When the driver steps on the brake pedal, the pedal force acting on the simulator can be directly measured. A travel sensor is installed parallel to the linear motor. When the pedal rotates around the fulcrum, the travel of the motor push rod can be measured, and then the travel of the pedal can be measured. The spring is coaxially installed with the motor push rod. The bottom of the linear motor is fixed to the carbon steel frame with a hinge, and the push rod is also connected to the pedal lever with a hinge. The motor is powered by a 24 V battery, and the motor thrust is regulated by a pulse width modulation (PWM) control method. The controller strategy is shown in Figure 10. The

control board integrates a H-bridge drive circuit, a signal acquisition and conditioning circuit, and an analog-to-digital conversion (ADC) module. The pedal force sensor uses a strain sensor. This sensor is based on the resistive strain effect and outputs a voltage signal. Then, the millivolt-level signal is amplified by the transducer to a range suitable for the ADC chip. The pedal travel sensor uses a resistive linear displacement sensor. This sensor is based on the potentiometer principle and outputs a voltage signal. The current sensor uses a Hall current sensor. These analog signals, such as the pedal force, pedal travel and current, are all converted into digital signals by the ADC module, which is convenient for the ECU to calculate and process. The clock frequency of the ADC module is set to 25 MHz, and the sampling frequency is 12.5 KSPS.

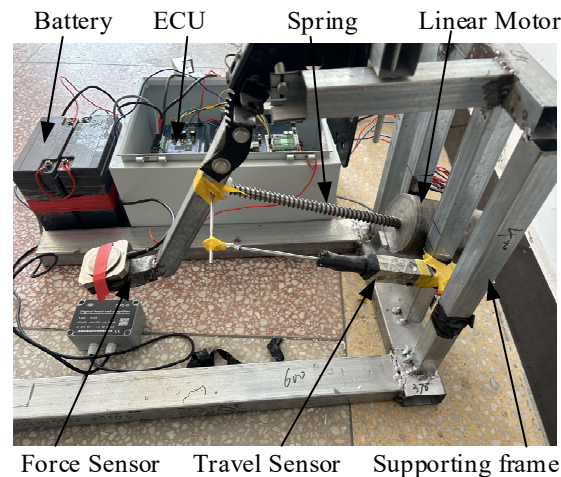


Figure 9. Experimental setup of the active brake pedal simulator.

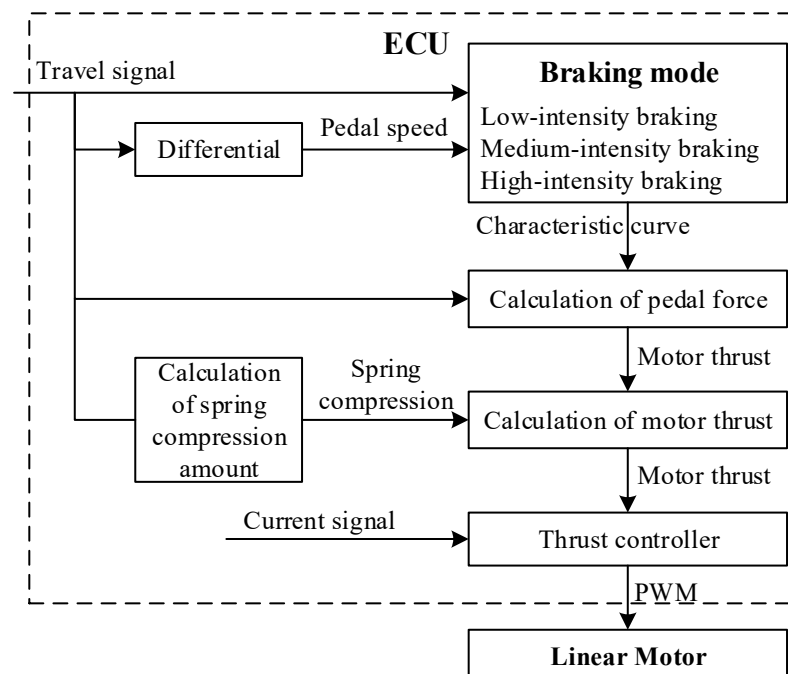


Figure 10. Control strategy of the active brake pedal simulator.

To achieve different braking pedal feel sensations at different braking intensities, three “pedal force–pedal travel” characteristic curves have been preset in the ECU, corresponding to low-intensity, medium-intensity and high-intensity braking scenarios respectively. During the operation, the ECU first collects the pedal travel signal and then calculates the pedal

speed based on this signal. Then, the ECU determines the braking mode based on the pedal speed and pedal travel and selects the corresponding target characteristic curve according to the braking mode. Then, the ECU calculates the required pedal force and motor thrust based on the target characteristic curve, pedal travel and spring compression amount. Finally, the PWM signal is output by the force control strategy to adjust the motor current. The specific control strategy is shown in Figure 10. Through this target characteristic curve judgment and switching mechanism, the simulator is able to produce different braking feel sensations without altering the mechanical structure.

4.2. Brake Pedal Simulator Test Under Different Braking Conditions

Experiments under four braking conditions, namely low-intensity braking, medium-intensity braking, high-intensity braking and ABS intervention braking, were completed on the established brake pedal simulator test platform. During low-intensity braking, the pedal is depressed at a speed of approximately 20 to 30 mm/s, and the pedal force is generally no more than 150 Newtons. This situation corresponds to gentle deceleration during normal driving. During medium-intensity braking, the pedal is depressed at a speed of approximately 40 to 60 mm/s, and the pedal force typically ranges from 200 to 450 Newtons. This situation corresponds to moderate deceleration or normal parking. During high-intensity braking, the pedal is rapidly depressed at a speed exceeding 80 mm/s. At this point, the pedal force can exceed 600 Newtons. This situation corresponds to an emergency braking scenario. When ABS is activated, the pedal simulator superimposes periodic vibration signals on the pedal force to simulate the feedback effect of hydraulic ABS.

In terms of control methods, two independent PWM drive signals are adopted to control coil A and coil B respectively. Four general purpose input/output (GPIO) ports are utilized to achieve flexible control of the forward and reverse directions of current in two sets of coils. To further enhance the smoothness of the linear motor, its total travel of 90 mm is divided into 18 sections, with each section having a travel length of 5 mm. After numerous tests and verifications, each small section of 5 mm travel was able to meet the smoothness requirements of linear motors. This can not only ensure control accuracy but also avoid the problems of program complexity and hardware resource consumption caused by overly fine travel.

During operation, the ECU continuously acquires real-time pedal travel data. Based on the real-time travel data, the ECU determines the magnetic field direction in which coils A and B are located. Based on the coil position and the braking mode selected according to Figure 10, the ECU determines the corresponding operating state from the nine preset states listed in Table 1. Each operating state defines specific current control modes for coils A and B, including the current direction and activation status. Finally, the ECU uses an incremental PID controller to complete the final thrust control based on the target thrust and the coil control mode. The detailed control process is shown in Figure 11.

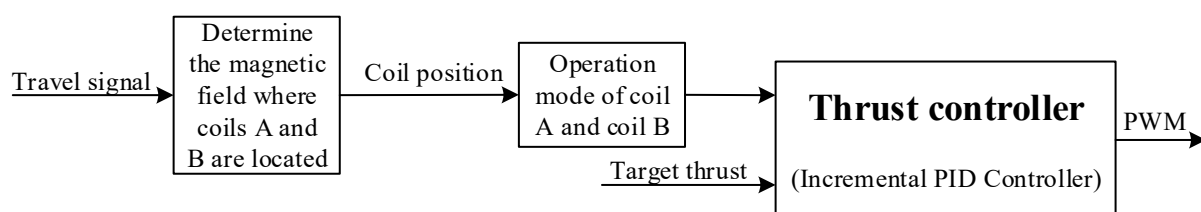


Figure 11. Control process of motor thrust.

4.2.1. Low-Intensity Braking

When braking with low intensity, the driver steps on the brake pedal slowly and almost at a constant speed. The pedal characteristic curve shows distinct two-segment variation characteristics, as shown in Figure 12. When the pedal travel is less than 65 mm, the slope of the curve is relatively small, and the pedal force gradually increases to approximately 100 N. When the pedal travel exceeds 65 mm, the slope of the curve increases rapidly, and the pedal force accordingly grows to approximately 350 N. In actual braking, low-intensity braking is generally used for slow deceleration, and the driver does not press the brake pedal too much. That is, the brake pedal travel is usually less than 65 mm. At this stage, the pedal force is relatively small, which is conducive to saving the driver's strength and improving driving comfort.

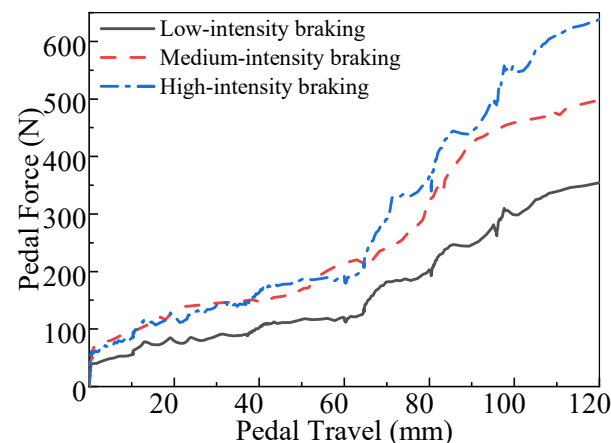


Figure 12. The pedal characteristic curve during low-intensity braking, medium-intensity braking and high-intensity braking.

4.2.2. Medium-Intensity Braking

When braking with medium intensity, drivers tend to press the brake pedal at a relatively fast speed. The pedal characteristic curve shows distinct three-segment variations, as shown in Figure 12. When the pedal travel is less than 20 mm, the slope of the curve is relatively large, and the pedal force rapidly increases to approximately 120 N. When the pedal travel is between 20 and 60 mm, the slope of the curve is relatively small, and the pedal force gradually increases to approximately 200 N. When the pedal travel exceeds 60 mm, the slope of the curve is relatively large, and the pedal force rapidly increases to approximately 450 N. When applying medium-intensity braking, the pedal characteristic curve is closer to that of a hydraulic braking system. Medium-intensity braking is generally used for rapid deceleration or stopping. Compared with low-intensity braking, the driver needs to step on the pedal with greater travel and also requires the pedal simulator to provide a stronger pedal reaction force. Therefore, the pedal force will be greater under the same pedal travel. This strategy can take into account both the pedal feel and the driver's strength. It can not only provide clear braking feedback but also help improve driving comfort.

4.2.3. High-Intensity Braking

When braking with high intensity, the driver steps on the brake pedal at a very fast speed. The pedal characteristic curve shows distinct two-segment variation characteristics, as shown in Figure 12. When the pedal travel is less than 65 mm, the pedal force increases at a relatively small rate to approximately 200 N. When the pedal travel exceeds 65 mm, the slope of the curve increases rapidly, and the pedal force also grows to approximately

650 N. Under high-intensity braking conditions, the pedal simulator needs to provide a stronger pedal reaction force, giving the driver a pedal feel that the pedal is “hard”.

By comparing the pedal characteristic curve during low-intensity braking, medium-intensity braking and high-intensity braking, it can be seen that the active pedal simulator based on linear motors can actively adjust the pedal characteristic curve according to the braking conditions. When braking at low intensity, the pedal simulator can provide relatively “soft” pedal feel. It can save the driver’s strength and improve the driving comfort performance. When braking at medium intensity, the pedal simulator can provide pedal characteristics consistent with the ideal characteristic curve. This can not only provide clear braking feedback feel but also help improve driving comfort. When braking at high intensity, the pedal simulator can provide a relatively “hard” pedal feel. This can provide drivers with stronger braking feedback feel on their feet.

4.2.4. ABS Intervention Braking

When braking in an emergency or on slippery road, the wheels may lock up and slide. If the wheels lock up, it will cause the vehicle to skid sideways, fishtail or lose steering ability, which is a very dangerous braking condition. Most passenger cars’ braking systems are equipped with ABS. When a hydraulic braking system activates ABS, the brake pedal will continuously vibrate. The driver will also feel the shaking and bouncing of the brake pedal, which indicates that the ABS function has been activated. The wheel braking unit and brake pedal are mechanically decoupled in BBW systems. Passive pedal simulators cannot actively feed back the wheels’ status to the driver. That is, they cannot provide the feel of shaking and bouncing. The active pedal simulator based on linear motors can achieve the above-mentioned rebound feel. The pedal will vibrate just like the pedal of a hydraulic braking system when ABS intervenes. On the one hand, it can indicate that the ABS has intervened through pedal vibration, reminding the driver to focus on driving and enhancing driving safety. On the other hand, it can also actively adjust the frequency and amplitude of vibration according to the tendency of wheels locking up, avoiding the driver’s discomfort.

When ABS is activated, the ECU will superimpose a periodic thrust signal on the original steady-state thrust command of the linear motor. This periodic signal can simulate the pressure oscillation characteristics of the hydraulic braking system, thereby generating significant pedal vibration feedback when ABS is activated. Since the linear motor can precisely and directly control the thrust, this method can accurately reproduce in real time the ABS feedback that is felt in a traditional braking system.

Figure 13 shows the low-frequency jitter with a frequency of 10 Hz. The maximum amplitude difference in the pedal jitter force is approximately 60 N. During the test, the driver could clearly feel the pedal shaking, which was completely different from the pedal feel during normal braking. Therefore, the driver can be aware that the ABS system has been activated. When ABS is involved, the shaking sensation of the pedal is rather subjective. Moreover, different vehicles or roads have different effects on the vibration of the pedal, and the frequency and amplitude of the vibration are also not consistent [25]. Figure 14 shows the high-frequency jitter with a frequency of 20 Hz. The maximum amplitude difference in the pedal jitter force is still 60 N. Two tests with different frequencies both verify that the pedal simulator can promptly alert the driver when the ABS intervenes. In conclusion, it has been verified through experiments that the active pedal simulator based on linear motors can provide drivers with different braking foot sensations according to braking conditions.

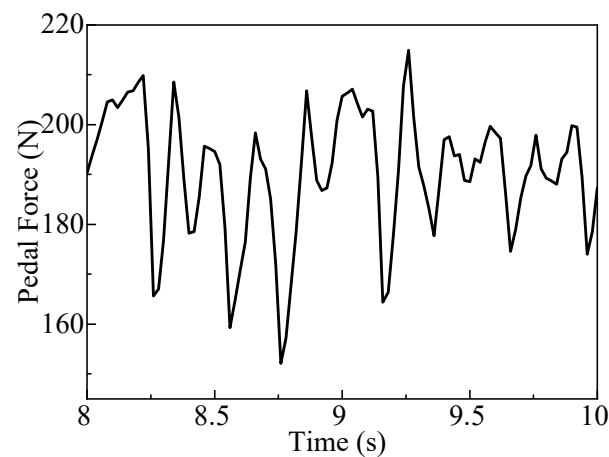


Figure 13. The pedal vibrates at low frequency when ABS is involved.

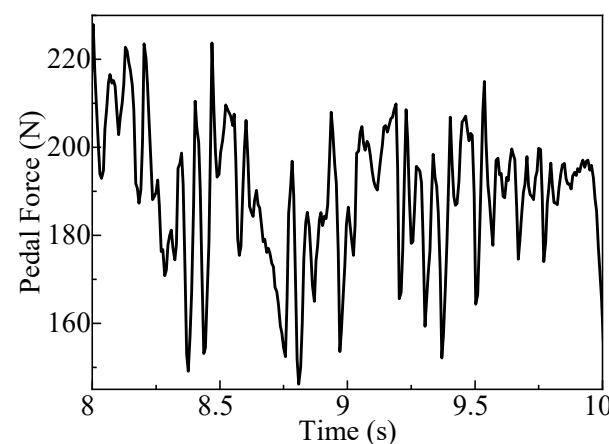


Figure 14. The pedal vibrates at high frequency when ABS is involved.

5. Conclusions

This paper designs an active brake pedal simulator based on Halbach-array linear motors to address the issue of missing pedal feel in BBW systems. The long-travel linear motor used in the pedal simulator can provide high-precision thrust, enabling rapid response and high dynamic control performance. By introducing springs with appropriate stiffness, the optimization of the brake pedal simulator was achieved while taking into account the feedback force and the energy consumption. The dual-coil independent control scheme effectively suppresses thrust fluctuations at the magnetic field junctions, ensuring smooth and reliable operation of the linear motor and pedal simulator.

The experimental results indicate that the pedal simulator can actively adjust the pedal characteristic curve under different braking intensities and provide obvious foot-bouncing feedback during ABS intervention. Therefore, the active pedal simulator designed in this paper has significant advantages in terms of structural simplification, adjustable pedal feel and rapid response. It provides a feasible solution for the human–vehicle interaction of BBW systems.

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References

1. Alanazi, F.K.R. Electric vehicles: Benefits, challenges, and potential solutions for widespread adaptation. *Appl. Sci.* **2023**, *13*, 6016. [\[CrossRef\]](#)
2. Chen, L.; Ma, R. Clean energy synergy with electric vehicles: Insights into carbon footprint. *Energy Strategy Rev.* **2024**, *53*, 101394. [\[CrossRef\]](#)
3. Ji, Q.; Zheng, L.; Bi, Y.; Pang, H. Review of brake-by-wire technology for low-speed and autonomous vehicles. *World Electr. Veh. J.* **2024**, *15*, 581. [\[CrossRef\]](#)
4. Hua, X.; Zeng, J.; Li, H.; Huang, J.; Luo, M.; Feng, X.; Xiong, H.; Wu, W. A review of automobile brake-by-wire control technology. *Processes* **2023**, *11*, 994. [\[CrossRef\]](#)
5. Zhang, L.; Wang, Q.; Chen, J.; Wang, Z.; Shao, L. Brake-by-wire system for passenger cars: A review of structure, control, key technologies, and application in X-by-wire chassis. *Etransportation* **2023**, *18*, 100292. [\[CrossRef\]](#)
6. Abeyisiriwardhana, W.A.S.P.; Abeykoon, A.M.H.S. Brake-by-wire control with pedal feedback and brake boost. *Bull. Electr. Eng. Inform.* **2021**, *10*, 3042–3051. [\[CrossRef\]](#)
7. Caliskan, U.; Patoglu, V. Efficacy of haptic pedal feel compensation on driving with regenerative braking. *IEEE Trans. Haptics* **2020**, *13*, 175–182. [\[CrossRef\]](#)
8. Bolognesi Prado, W.A.S.; Iombriller, S.; Silva, M.A.R.; Oliveira, L.R. Brake pedal feeling comfort analysis for trucks with pneumatic brake system. *SAE Technical Paper* **2019**, 2019-01-2140. [\[CrossRef\]](#)
9. Spadoni, A. Adaptive Brake by Wire: From Human Factors to Adaptive Implementation. Ph.D. Dissertation, University of Trento, Trento, Italy, 2013.
10. Zehnder, J.W.; Kanetkar, S.S.; Osterday, C.A. *Variable Rate Pedal Feel Emulator Designs for a Brake-by-Wire System*; SAE International: Warrendale, PA, USA, 1999; pp. 881–884.
11. Yin, F.; Wang, M.; Jiang, Y.; Kang, Y. A passenger car brake pedal feel analysis model based on integrated brake-by-wire system. *SAE Tech. Pap.* **2021**, 2021-01-0975. [\[CrossRef\]](#)
12. Lee, S.D.; Kim, S.L. Characterization and development of the ideal pedal force, pedal travel, and response time in the brake system for the translation of the voice of the customer to engineering specifications. *Proc. Inst. Mech. Eng. Part D J. Automob. Eng.* **2010**, *224*, 1433–1450. [\[CrossRef\]](#)
13. Shi, Z.; Yan, Y.; Zhang, S. Variable boost characteristic control strategy of hydraulic systems for brake-by-wire based on driving style. *Sci. Rep.* **2024**, *14*, 29412. [\[CrossRef\]](#) [\[PubMed\]](#)
14. Miao, J.; Yao, M.; Wen, P.; Li, S.; Xiong, H.; Huang, Z. Optimization for springs of electronic brake pedal simulator. In *IOP Conference Series: Materials Science and Engineering*; IOP Publishing: Bristol, UK, 2019; Volume 612, p. 032064. [\[CrossRef\]](#)
15. Wang, D.; Wang, B.; Zi, B.; Zhou, Q.; Zheng, X.; Shen, J. Development and control of a magnetorheological damper-based brake pedal simulator for vehicle brake-by-wire systems. *Chin. J. Mech. Eng.* **2022**, *35*, 136. [\[CrossRef\]](#)
16. Farshizadeh, E.; Steinmann, D.; Briesse, H.; Henrichfreise, H. A concept for an electrohydraulic brake system with adaptive brake pedal feedback. *Proc. Inst. Mech. Eng. Part D J. Automob. Eng.* **2015**, *229*, 708–718. [\[CrossRef\]](#)
17. Gumiel, J.Á.; Mabe, J.; Burguera, F.; Jiménez, J.; Barruetaña, J. Next-generation pedal: Integration of sensors in a braking pedal for a full brake-by-wire system. *Sensors* **2023**, *23*, 6345. [\[CrossRef\]](#)
18. Kikuchi, T.; Noma, J.; Akaiwa, S.; Ueshima, Y. Response time of magnetorheological fluid-based haptic device. *J. Intell. Mater. Syst. Struct.* **2015**, *27*, 859–865. [\[CrossRef\]](#)
19. Kang, P.; Liu, S.; Zeng, T. Magnetorheological fluid-based haptic feedback damper. *Appl. Sci.* **2024**, *14*, 3697. [\[CrossRef\]](#)
20. Hofmann, T.; Huepfel, M.; de Haas, O.; Jocher, A. Design, scalable construction, and test of optimal linear Halbach arrays for mobile applications. *J. Magn. Magn. Mater.* **2024**, *596*, 171967. [\[CrossRef\]](#)
21. Liu, J.; Shang, M.; Gong, C. Analytical modeling and analysis of Halbach array permanent magnet synchronous motor. *World Electr. Veh. J.* **2025**, *16*, 413. [\[CrossRef\]](#)
22. Xiao, F.; Gong, X.; Lu, Z.; Qian, L.; Zhang, Y.; Wang, L. Design and control of a new brake-by-wire actuator for vehicle based on linear motor and lever mechanism. *IEEE Access* **2021**, *9*, 95832–95842. [\[CrossRef\]](#)
23. Bogaychuk, A.; Ivshin, E. Improving the homogeneity of Halbach arrays by optimizing magnet combinations using a genetic algorithm. *Rev. Sci. Instrum.* **2024**, *95*, 055107. [\[CrossRef\]](#)

24. Prado, W.B.; Iombriller, S.F.; da Silva, M.A.R.; Oliveira, L.R. Commercial vehicle pedal feeling comfort ranges definition. *SAE Tech. Pap.* **2020**, 2019-36-0016. [[CrossRef](#)]
25. Zhang, H.; Qiao, J.; Zhang, X. Nonlinear dynamics analysis of disc brake frictional vibration. *Appl. Sci.* **2022**, *12*, 12104. [[CrossRef](#)]

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