Research on accurate modeling and control for pneumatic electric braking system of commercial vehicle based on multi-dynamic parameters measurement

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Abstract
Accurate pressure control and fast dynamic response are vital to the pneumatic electric braking system (PEBS) for those commercial vehicles that require higher regulation precision of braking force on four wheels when braking force distribution is carried out under some conditions. Due to the lagging information acquisition, most feedback-based control algorithms are difficult to further improve the dynamic response of PEBS. Meanwhile, feedforward-based control algorithms like predictive control perform well in improving dynamic performance but because of the large amount of computation and complexity of this kind of control algorithm, it cannot be applied in real-time on the single-chip microcomputer, and it is still in the stage of theoretical research at present. To address this issue and for the sake of engineering reliability, this article presents a logic threshold control scheme combining analogous model predictive control (AMPC) and proportional control. In addition, an experimental device for real-time measuring PEBS multi-dynamic parameters is built. After correcting the key parameters, the precise model is determined and the influence of switching solenoid valve on its dynamic response characteristics is studied. For the control scheme, numerical and physical validation is executed to demonstrate the feasibility of the strategy and for the performance of the controller design. The experimental results show that the dynamic model of PEBS can accurately reflect its pressure characteristics. Furthermore, under different air source pressures, the designed controller can stably control the pressure output of PEBS and ensure that the error is within 0.08 bar. Compared with the traditional control algorithm, the rapidity is improved by 32.5%.

Keywords
Vehicle dynamic, pneumatic electric braking system, brake system modeling, proportional control

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Introduction
Commercial vehicles have the characteristics of a large mass, long body, high centroid, and variable load, their braking safety and driving stability are easily affected by an uneven distribution of braking force, which leads to frequent accidents such as the rear-end collision and rollover.1 As an important component exerting braking force then implementing active stability control for commercial vehicles, PEBS adopts the combination of electronic control and traditional braking technology for braking. Compared with traditional braking, it can greatly improve the response speed during vehicle braking and reduce the braking distance. Furthermore, driving safety has a high requirement for the precise pressure control of PEBS when a vehicle operates under extreme conditions such as sharp turns and sudden stops in which minor braking force fluctuation can...
lead to dynamic instability.\textsuperscript{2,3} Besides, considering the characteristics of PEBS such as rapid response, high amenity, electronic control, and mechanical redundancy, it is expected to become the basic module of driverless commercial vehicles in the future.\textsuperscript{4} Therefore, research on how to get accurate and stable pressure of PEBS especially in extreme conditions is significant to improve the safety of commercial vehicles.

In the PEBS of commercial vehicles, proportional servo valves and high-speed switch solenoid valves are two common pneumatic valves. According to their characteristics, their control mode is to control different pressures according to the input analog linear signal\textsuperscript{5–7} and pulse width modulation signal, respectively. Among them, the high-speed switching solenoid valve is widely used because of its simple structure and low price. Pneumatic Brake-by-Wire Valve (PBWV) as the basic actuator of PEBS, uses a high-speed switching solenoid valve group to control pressure.\textsuperscript{7,8} Considering the inherent opening and closing characteristics of the solenoid valve, it is difficult to adjust the gas flow continuously and infinitely.\textsuperscript{7} In addition, compressed air as the power-transmitting medium in PEBS, the braking pressure is susceptible to working conditions such as temperature and humidity. Moreover, different tubing and air source pressure also have different effects on the braking pressure.\textsuperscript{9,10} Aiming at the above non-linear and air source pressure also have different effects on the temperature and humidity. Moreover, different tubing pressure is susceptible to working conditions such as the power-transmitting medium in PEBS, the braking accuracy of the model, and because of its large amount of calculation, it is difficult to be applied in the single-chip microcomputer. So how to establish an accurate model and fast calculation has become a difficult problem to accurately control the braking pressure.

This paper proposes a control architecture that combines MPC with traditional feedback control. Next, the precise model MPC will need is built using an experimental measuring device to obtain and modify several dynamic parameters. Based on the accurate model, the map diagram reflecting the system dynamics is established by simulating, and this map can be used to conduct the MPC. The rest of the paper is organized as follows. Section 2 analyzes the working mechanism of the PEBS during braking and introduces the PEBS system model. Section 3 designs a controller based on the PEBS model. Section 4 sets up several experimental measuring devices for multiple dynamic parameters of PEBS to modify the model and verifies the control effect of the controller. Numerical study and physical testing are performed to validate the presented design in Section 5. Finally, the conclusion is presented in Section 6.

Braking principle and PEBS model

Braking principle

In PEBS, there are mainly PBWV and ABS valves to control the braking pressure. To study the dynamic characteristics and control effects of PBWV more specifically, it is assumed that the circuit is working normally and the wheels are not locked (ABS valve not working). The working principle of PEBS is shown in Figure 1.

Figure 1 shows a general pneumatic electric braking system. The system consists of four components; namely, a control unit, an air supply system, a PBWV, and an actuator. The PBWV includes an inlet valve, a release valve, a backup valve, and a relay valve. The inlet valve and release valve belong to the high-speed switch valve that together is controlled by PWM (Pulse Width Modulation) signal. The backup pressure valve is a switch valve that closes once the inlet valve or release valve is energized. The piston of the relay valve moves axially to increase or decrease the pressure of the actuator (brake air chamber).

Figure 2 defines the position of the relay valve piston and its corresponding working state. When the electric control circuit is working normally, the backup valve does not work, so the backup valve is not drawn in the figure. In Figure 2, three working states of the PBWV correspond to three working processes; namely, pressure increasing process, pressure holding process, and pressure decreasing process.

Solenoid valve model

In PEBS, as a high-speed switch valve, the solenoid valve’s dynamic response characteristics directly affect the performance and working efficiency. In the
working process of the solenoid valve, the parameters related to the movement of the spool mainly include voltage, current, inductance, spool displacement, etc. After a lot of simulation in Ansys Maxwell, it is found that the inductance value in the system presents a non-linear relationship with the change of the spool displacement and its change rate is small. To roughly obtain the relationship between the current and the voltage in the solenoid valve, the average value of the inductance is taken as the fixed inductance for design. The relationship between the current and voltage in the coil of the solenoid valve is

\[ U = iR + L \frac{di}{dt} \]

where \( i, L, R \), and \( U \) are the current, the inductance, the resistance of the electromagnetic coil, and the voltage, respectively.

The pneumatic brake-by-wire valve is a definite system in this article, its internal parameters (e.g. the electric-magnetic field characteristics of the material, coercivity, etc.) have a fixed effect on the electromagnetic force, while the external parameters such as current and spool displacement is the main external factor affecting electromagnetic force. The specific calculation of electromagnetic force is complicated, but there is a certain relationship between input and output. After a large number of experiments and simulation studies, the relationship between electromagnetic force, current,
and spool displacement can be expressed by a fitting formula. Therefore, assumed that the calculation expression of the electromagnetic force as

$$F_e = f(i, x)$$

(2)

where $F_e$ and $x$ are the electromagnetic force and the spool displacement, respectively.

Figure 3 shows the solenoid valve opening process. Its drawing method is to perform simulation analysis in Ansys Maxwell when the spool displacement is 0.2, 0.4, 0.6, ..., 1 mm, respectively, to obtain the data of current, electromagnetic force, and time under the corresponding displacement. The data is fitted in MATLAB, and the three-dimensional map of displacement, electromagnetic force, and current is obtained. Considering that the material properties, magnetic permeability, coercive force, and other parameters of the solenoid valve studied in this paper are fixed, the three-dimensional data obtained by the above processing methods only partially exist, and boundaries need to be added to distinguish them. Since the power supply voltage of the commercial vehicle is 24 V, and the solenoid valve is controlled by PWM, we collect data with the minimum and maximum duty cycle to determine the boundary of the highlighted part under the condition that the solenoid valve can be opened. Similarly, the three-dimensional diagram when the solenoid valve is closed can be obtained. After analysis and verification, the fitted electromagnetic force expression is equation (3), which can only represent the highlighted part of Figure 3.

$$F_e = 27.66i^2 + 2.04xi + 8.74i - 4.27x - 4.47$$

(3)

Figure 4 shows the internal structure of the solenoid valve. According to Newton’s second law, the kinematic equation of the moving spool is

$$m\ddot{x} = F_e - pA_0 - c\dot{x} + k_s(x + x_p) \quad (0 \leq x \leq x_{\text{max}})$$

(4)

where $m$, $A_0$, $c$, $x_p$, and $k_s$ are the mass of the moving spool, the effective area of the moving spool, the damping coefficient, spring preload displacement, and spring stiffness, respectively. In addition, “−” represents different processes, “+” represents the solenoid valve opening process, and “−+” represents the solenoid valve closing process.

**Dynamic modeling of PBVV**

Based on Figure 2, Figure 5 shows the corresponding piston position. Similarly, the piston displacement in
the brake chamber can be understood as a simple piston displacement.

According to the above analysis of the dynamic behavior of the PBWV braking process, the compressed air in the control chamber or brake chamber must produce enough force to drive the piston movement of the relay valve or brake chamber. In this process, the volume change equation of the control chamber and the load chamber is

\[ V_t = V_{t0} + A_t x_t \quad (0 \leq x_t \leq x_{rt}) \]  
(5)

\[ V_c = V_{c0} + A_c x_c - A_t x_t \quad (0 \leq x_t \leq x_{rt}, 0 \leq x_c \leq x_{ct}) \]  
(6)

where, \( V_t, V_{t0}, \text{ and } A_t \) are the volume, the initial volume, and the effective area in the control chamber, respectively, \( V_c, V_{c0}, \text{ and } A_c \) are the volume, the initial volume, and the effective area in the load chamber, respectively; \( x_t \) and \( x_c \) are the displacement of the relay valve piston and the brake chamber piston, respectively; \( x_{ct} \) is the total piston displacement of the relay valve during inlet and release; \( x_{ct} \) is the piston displacement of the brake chamber.

According to Newton’s second law, the kinematic equation of the piston in the relay valve and the brake chamber are respectively.

\[
\begin{align*}
\dot{x}_t &= A_t p_t - A_{t1} p_c - F_{ft} \quad (0 \leq x_t \leq x_{rm})  \\
\dot{x}_c &= A_c p_c - K_c x_{oc} - F_{fc} \quad (0 \leq x_c \leq x_{ct})
\end{align*}
\]  
(7)

where, \( m_t, m_{t1}, \text{ and } m_c \) are the mass of the relay valve piston, plug, and the brake chamber piston respectively; \( p_t \) and \( p_c \) are the pressure in the control chamber and brake chamber respectively; \( A_{t1} \) is the effective area of the load chamber; \( F_{ft}, F_{f1t}, \text{ and } F_{fc} \) are the friction between the piston of relay valve, plug, the piston of the brake air chamber, and the side; \( x_{rm} \) is the displacement of the piston during the release; \( x_{or} \) and \( x_{oc} \) are the pre-tension displacement of the relay valve spring and the brake chamber spring respectively; \( K_t \) and \( K_c \) are the spring stiffness of the relay valve spring and the brake chamber respectively.

Generally speaking, because braking is relatively fast and fierce, the variation of the air in the control chamber can be regarded as adiabatic. Assuming that the temperature difference between the gas entering the control chamber or the brake chamber and its original gas is ignored, the aerodynamic process within the system can be expressed as

\[ p(V/m)' = p_0(V_0/m_0)' = C \]  
(9)

where \( p_0, V_0, \text{ and } m_0 \) are the pressure, volume, and mass at the initial condition, respectively, and \( p, V, \text{ and } m \) are the corresponding states at equilibrium, \( r \) is an adiabatic exponent of gas, \( C \) is a proportional constant.

Note: When the closed chamber is the control chamber or brake air chamber, the above parameters correspondingly change to \( p_t, V_t, m_t, p_{t0}, V_{t0}, m_{t0} \text{ or } p_c, V_c, m_c, p_{c0}, V_{c0}, m_{c0} \). Differentiating equation (9) with respect to time,

\[ \dot{p} = (rp/m)Q - (rp/V)\dot{V} \]  
(10)

where \( Q = \dot{m} \) is the mass flow rate of gas in the closed chamber.

According to the formulation between the gas flow and orifice area it can be shown that

\[ Q = \begin{cases} 
\left( \frac{2}{\gamma+1} \right)^{1/(\gamma-1)} \left( \frac{p_2}{p_1} \right)^{2\gamma/(\gamma+1)} \right)^{1/2}, & (0 \leq p_2/p_1 < 0.528) \\
\left( \frac{2}{\gamma+1} \right)^{1/(\gamma-1)} \left( \frac{p_2}{p_1} \right)^{2\gamma/(\gamma+1)} \right)^{1/2}, & (0.528 \leq p_2/p_1 \leq 1)
\end{cases} \]  
(11)

where \( c_d \) is the gas orifice flow coefficient, \( p_1 \) is the upstream air pressure, \( p_2 \) is the downstream air pressure, \( A \) is the effective throttle area, \( T_a \) is the absolute temperature in the control chamber, \( R \) is the specific gas constant, Constant 0.528 is a critical ratio of low-to-high pressure at the two sides of the orifice. When greater than 0.528, the flow is subsonic. Otherwise, the flow is supersonic and the corresponding dynamics are complex which is oftentimes the case when air undergoes throttling.

Note: When the gas flows into the control chamber, \( p_1 = p_s, p_2 = p_i \), when the gas flows out of the control chamber \( p_1 = p_c, p_2 = p_a \), when the gas flows into the brake chamber \( p_1 = p_s, p_2 = p_c \), when the gas flows out of the control chamber \( p_1 = p_c, p_2 = p_a \). Where \( p_s \) is the pressure of supply air, \( p_a \) is the discharge atmospheric pressure.

**Controller design**

The principle of MPC control is to solve the finite-time-domain open-loop optimization problem online at each sampling time according to the measurement information obtained at the current time, and then apply the first element of the solved control sequence to the controlled object. At the next sampling time, repeat the
above process, refresh the optimization problem with new measurements, and resolve it. The process of the algorithm is mainly divided into three steps: predictive dynamic model, rolling optimization, and feedback correction. These three steps can solve the open-loop optimization problem online according to the data update at each sampling time, rather than acting the same feedback control law on the system all the time as the traditional control method. Through rolling optimization and feedback correction, it has the advantages of good control effect and strong robustness, which can effectively overcome the uncertainty, nonlinearity, and parallelism of the process.

According to the dynamic model created in the “Braking principle and PEBS model” section, it can be seen that the PEBS model is more complex and the system has strong-nonlinearity. When the MPC method is directly used to control the pressure of PEBS, because of the large amount of calculation, the result may not be calculated or the calculation speed is too slow. To improve the speed of MPC calculation for efficient control, this paper uses the accurate model to collect data and draw the MAP diagram, and directly uses the MAP diagram as the system dynamics model, which is used as the characteristic of a predictive control system.

The MAP diagram is shown in Figure 6, which refers to the increase/decrease characteristics of PBWV within 1–10 ms. It is obtained by collecting and processing brake pressure data from the PEBS model constructed above. Specific parameters are set to 7.5 bar for air receiver pressure and the period is 0.5 s, according to the control period 1–10 ms, corresponding to the pressure data collected repeatedly when the waiting period is 499–490 ms, then the collected pressure data are analyzed and processed to obtain the relationship between pressure and pressure change during increase/decrease.

Because the control principle is consistent with that of MPC, the control algorithm is named analogous model predictive control (AMPC). The advantage of AMPC is that when the expected pressure deviates greatly from the actual pressure, the algorithm responds faster than the PID algorithm in the same cycle and makes the actual pressure follow the expected pressure earlier. According to MPC theory, the model is modified or compensated in real-time. However, for the sake of engineering reliability, the AMPC method is adopted when the error is large, so that the limited performance of the system can quickly approach the target. When the error is small, using proportional feedback continuous control is more appropriate. That is, a combination of AMPC and P control algorithms is used to control the pressure output. A schematic diagram of the PEBS pressure controller is shown in Figure 7.

As seen from Figure 7, when the error is large and exceeds the limit value \( e_p > e_T \), the pressure output of PEBS is controlled by the AMPC algorithm, on the contrary, it is controlled by a proportional control algorithm. Where \( e_p = p_d - p_c \)

\[ p_d \text{ and } p_c \text{ are the target pressure and actual pressure of the brake chamber, respectively. } e_T \text{ is the limit value of the decision selection control algorithm.} \]

Aiming at the AMPC control algorithm, to improve the accuracy of the algorithm, the \( \beta \) value is introduced to correct the error. The corrected error is expressed as

\[ e = e_p + \beta e_m \]

where

\[ e_m = \Delta p_t - \Delta p_t \]

\[ \Delta p_t = p_{c(k)} - p_{c(k-1)} \]

\[ \Delta p_t = p_{t(k-1)} - p_{t(k-1)} \]

\( p_{c(k)} \) and \( p_{c(k-1)} \) are the actual value of the current moment and the previous moment, respectively. \( p_{t(k-1)} \) is the target pressure of the previous moment.
According to the corrected error, the time of pressurization or decompression is found out based on the 
map diagram. Specifically expressed as

\[ t_{\text{increase}} = 10 \times e_p \]

\[ t_{\text{decrease}} = \max(e_{\text{map-increase}}) \]

\[ e_p > \max(e_{\text{map-increase}}) \]

\[ e_p < \min(e_{\text{map-decrease}}) \]

where \( t_{\text{increase}} \) and \( t_{\text{decrease}} \) are the pressurization and decompression time, respectively. \( e_{\text{map-increase}} \) and \( e_{\text{map-decrease}} \) are the maximum and minimum pressure differences of pressurization and decompression in the MAP diagram.

According to the P control algorithm, the calculation equation of booster and decompression time is

\[ u = K \times e_p \]

\[ t_{\text{increase}} = \begin{cases} 10 & \text{if } |u| > |\text{LimitValue}| \\ 10 \times (u/|\text{LimitValue}|) & \text{if } |u| \leq |\text{LimitValue}| \end{cases} \]

where \( K \) and \( \text{LimitValue} \) are proportional gain values and thresholds, respectively.

**Parameter acquisition and model verification**

According to the AMPC control principle, because the MAP diagram is used to predict the characteristics of the system, its drawing comes from the accurate PEBS model, so it is necessary to verify the PEBS model created in the “Braking principle and PEBS model” section. It is concluded from the analysis that if the model can real-time and accurately reflect the dynamic parameters such as solenoid valve opening and closing time, gas flow coefficient, piston displacement of relay valve, control chamber pressure, and brake chamber pressure in PBMV, the model can accurately describe the performance of the PEBS. Therefore, in this section, a variety of experimental devices are built to measure the dynamic parameters in real-time and verify the model according to the experimental data. The schematic diagram of the experimental device is shown in Figure 8.

**Determine the solenoid valve opening and closing delay time**

To better reflect the characteristics of the solenoid valve, it is necessary to know the delay time of the solenoid valve in the opening and closing section. Figure 8(a) shows a measuring device for the solenoid valve delay time. The internal flow passage of the device meets the intake and exhaust of the solenoid valve. Since the moving spool will generate a corresponding back electromotive force once it is displaced, the current change rate will decrease at that moment, so the solenoid valve delay time can be measured according to the current change time. Figure 9 shows the current change...
of the solenoid valve under different voltages. It can be seen from the figure that when the solenoid valve is energized, the current gradually increases during 0–4 ms. At 4 ms, due to the sudden decrease of the current rate, it can be judged that the solenoid valve starts to work, that is, the opening delay time is 4 ms. In the same way, the solenoid valve closing delay time is 1 ms.

Determine the gas flow coefficient

Since the pressure of the control chamber cannot be measured directly, a new valve cover that can install a pressure sensor is redesigned to measure without destroying the performance of the valve. The measuring schematic diagram is shown in Figure 8(b). The pressure of the brake chamber is measured directly by the pressure sensor. According to equation (11), the gas flow coefficient is related to \( c_d, p_1, p_2, k, A, T_a, \) and \( R \). In general, the value of \( c_d \) is unknown and other values can be measured and determined, so the gas flow coefficient depends on \( c_d \). To modify \( c_d \) conveniently and quickly, this paper modifies the \( c_d \) directly according to the delay time and the increasing trend of the pressure data of the brake chamber.

Figure 10 shows that the actual braking pressure is compared with the pressure calculated by different \( c_d \) in the model. From the analysis of the figure, it is found...
that when \( c_d = 0.23 \), the calculated braking pressure is closest to the actual pressure.

**Determine piston displacement of relay valve**

The piston of the relay valve controls the filling and outgassing of the whole valve, which is an important index that affects the performance of the whole valve. To meet the requirements of low cost, high measurement accuracy, and simple operation when measuring the piston displacement in the valve, a fixed direct laser sensor bracket are designed based on the PBWV. Specifically, a light pillar is fixed by a thread at the top of the piston, and a circular plate is placed at the other end of the pillar to ensure that the displacement of the piston is reflected by the displacement of the plate irradiated by laser. The measuring schematic diagram and the actual bench are shown in Figure 8(c).

According to the working condition of PBWV, the piston displacement of the relay valve is measured in three starting positions (top position, free position, and central position). In the experiment, the pressure of the air receiver is controlled as 8 bar, the input signal is constant for 0–1 s, the pressure is pressurized for 1–5 s, and the 5–9 s is decompressed. The data are collected and compared as shown in Figure 11. Figure 11(a) shows the displacement comparison of the piston in different initial positions, in which the piston is in the dynamic equilibrium position between 1.5 and 5 s, and Figure 11(b) shows the corresponding pressure change of the brake chamber.

The analysis shows that no matter where the initial position of the piston is, the corresponding pressure change is consistent; when the piston starts at the top, the maximum displacement of the piston is 3.88 mm; when the piston starts in the free position, the displacement is 1.5 mm until the piston is stable; when the piston starts in the central position, there is a dead zone from this position to the dynamic equilibrium position, and the dead zone displacement is 0.4 mm.

**Model verification**

According to the above experimental equipment, a large number of data are measured and the relevant parameters in the model are modified. The actual data in the model are shown in Tables 1 and 2. In this section, under the premise of the same parameters, the simulation and experiment of the PEBS are carried out respectively.

Figure 12 shows the comparison between simulation and experimental data when the air receiver pressure is set to 2, 4, 6, 8 bar. The analysis shows that the simulation data are basically consistent with the experimental data.

Figure 13 shows brake chamber pressure data between simulation and experiment based on P control.
The pressure of the air source is controlled to 7 bar, and the expected pressure is set to step waveform, triangular waveform, and sinusoidal waveform. The analysis shows that the simulation data are basically consistent with the experimental data.

Comprehensive analysis shows that the above model can accurately reflect the characteristics of the PEBS and can provide guidance for the analysis of its performance.

**Experiments and results**

Figure 14 shows the indoor test bench built to validate the controller design. The brake chamber pressure is obtained through the pressure sensor embedded in the PBWV of commercial vehicles. The controller chip is SPC5 series automotive 32-bit micro-control chip designed by ST Co., which provides multi-channel IO ports, AD analog signal acquisition, PWM input and output, and CAN communication. The function configuration and the writing of the underlying code are completed by the SPC5 series special development software SPC5 Studio, and the online debugging is carried out by the supporting debugging software UDE STK. In the process of online debugging, the expected pressure, actual pressure, and other information are transmitted to the host computer through the CANalyst to realize CAN communication. The real-time tracking control and parameter adjustment of a pressure curve are carried out through the MATLAB software in the host computer, so that the pressure control experiment can be carried out efficiently.

Based on the existing situation, this paper controls the pressure output of the PEBS under different air

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**Table 1.** The parameters used in solenoid valve.

| R/Ω | L/H | m/kg | c/(N m/s) | K/(N/m) | x₀/m |
|-----|-----|------|-----------|---------|------|
| 15  | 0.05| 0.007| 10        | 110     | 5e-3 |

**Table 2.** The parameters used in the system.

| Parameters | Values | Parameters | Values |
|------------|--------|------------|--------|
| pᵢ/bar    | 8      | mᵢ/kg     | 0.046  |
| pᵢ/bar    | 1      | mᵢ/kg     | 5e-4   |
| Tᵢ/K      | 300    | pᵢ₀/bar   | 1      |
| cᵢ        | 0.23   | Vᵢ₀(m³)   | 2.2e-5 |
| Rᵢ/[kg(K)]| 287.1  | Aᵢ/[m²]   | 3.2e-3 |
| k          | 1.4    | Aᵢ₀/[m²]  | 3.1e-3 |
| xᵢ₀/m     | 0      | Kᵢ₀/[N/m] | 2500   |
| xᵢ₀/m     | 1.5e-3 | xᵢ₀/m     | 0.02175|
| xᵢ₀/m     | 3.88e-3| mᵢ/kg     | 0.65   |
| pᵢ₀/bar   | 1      | Vᵢ₀/m     | 7.5e-5 |
| Aᵢ/[m²]   | 9.5e-3 | Kᵢ₀/[N/m] | 2150   |
| xᵢ₀/m     | 0.055  | xᵢ₀/m     | 0      |
| xᵢ₀/m     | 0.012  | xᵢ₀/m     | 0      |

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**Figure 13.** Dynamic verification: (a) step response, (b) triangular wave response, and (c) sin response.
receiver pressure. Figure 15 shows that under different expected pressures, the pressure output of PEBS is controlled by a simple proportional control algorithm by changing the pressure of the air source.

The analysis shows that if an air source pressure is set and the proportional control parameters are adjusted, the valve pressure can be effectively controlled, but there will be local jitter under other air receiver pressure. In addition, the P control algorithm can control the error within 0.2 bar.

Figure 16 shows the real-time control of the valve by using the control logic of AMPC + P and proportional control based on the same conditions as proportional control. From the control results, the actual pressure can well follow the step and the triangle wave target pressure within the error is 0.08 bar, so it is verified that the combination of AMPC and proportional control algorithm proposed in this paper can well regulate and control the target pressure under different air receiver pressure.

According to the comparison between Figure 15 and Figure 16, the AMPC + P algorithm is more stable and has smaller errors than feedback control. To compare the control effects of the two methods more intuitively, this article compares the control effects of the two control methods under the same air source pressure. The specific data collection method is to keep the air source pressure at 8 bar, and collect five sets of data for each method and take the average value. Figure 17 shows that under the same conditions, the P control algorithm and the AMPC + P control algorithm are used to control the PBWV, and the response time is 77 and 52 ms.
respectively, that is, the rapidity of the new algorithm is 32.5% higher than that of the traditional algorithm.

**Conclusions**

In this paper, according to several experimental measuring devices of dynamic parameters, the accurate model of PEBS is determined by real-time measurement and correction of parameters. The accuracy of the model is verified by comparing the simulation and experimental data, and the dynamic model of PEBS can accurately reflect its pressure characteristics. In addition, based on the accurate model, a logic threshold control scheme based on the combination of AMPC
control and feedback control is proposed for the time-varying parameters of the system. The experimental results show that under different air source pressures, the designed controller can stably control the pressure output of PEBS and ensure that the error is within 0.08 bar and the rapidity is improved by 32.5%.

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References
1. Hussain K, Stein W and Day AJ. Modelling commercial vehicle handling and rolling stability. Proc IMechE, Part K: J Multi-body Dynamics 2005; 219(4): 357–369.
2. Morrison G and Cebon D. Combined emergency braking and turning of articulated heavy vehicles. Veh Syst Dyn 2017; 55: 725–749.
3. Morrison G and Cebon D. Sideslip estimation for articulated heavy vehicles at the limits of adhesion. Veh Syst Dyn 2016; 54(11): 1601–1628.
4. Wu J, Wang X, Li L, et al. Hierarchical control strategy with battery aging consideration for hybrid electric vehicle regenerative braking control. Energy 2018; 145: 301–312.
5. Ning F, Shi Y, Cai M, et al. Research progress of related technologies of electric-pneumatic pressure proportional valves. Appl Sci 2017; 7(10): 1074–1089.
6. Lambeck S and Busch C. Exact linearization control for a pneumatic proportional pressure control valve. In: Proceedings of the IEEE international conference on control and automation. Xiamen, China, 9–11 June 2010. New York, NY: IEEE.
7. Topçu E E, Yüksel and Kamuš Z. Development of electro-pneumatic fast switching valve and investigation of its characteristics. Mechatronics 2006; 16(6): 365–378.
8. You M, Zhang J, Sun D, et al. Characteristics analysis and control study of a pneumatic proportional valve. In: IEEE advanced information technology electronic and automation control conference, 2015, pp.242–247.
9. Wang ZS, Li GY, Wu QQ, et al. Research on pressure characteristics of vehicle air braking system with leakage from pipeline. Appl Mech Mater 2012; 157: 608–611.
10. Bharath S, Nakra BC and Gupta KN. Mathematical model of a railway pneumatic brake system with varying cylinder capacity effects. J Dyn Syst Meas Control 1990; 112(3): 456–462.
11. Wang Z, Zhou X, Yang C, et al. An experimental study on hysteresis characteristics of a pneumatic braking system for a multi-axle heavy vehicle in emergency braking situations. Appl Sci 2017; 7(8): 799–815.
12. Pipan M and Herakovic N. Closed-loop volume flow control algorithm for fast switching pneumatic valves with PWM signal. Control Eng Pract 2018; 70: 114–120.
13. Wu X, Li L, Wang X, et al. Nonlinear controller design and testing for chatter suppression in an electric-pneumatic braking system with parametric variation. Mech Syst Sig Process 2020; 135: 106401.
14. Dousti M and Başlamış ŞC. Experimental validation and robustness analysis of a multiple model switching antilock braking system control algorithm. Int J Veh Des 2016; 71: 226–257.
15. Hodgson S, Tavakoli M, Pham MT, et al. Nonlinear discontinuous dynamics averaging and PWM-based sliding control of solenoid-valve pneumatic actuators. IEEE/ASME Trans Mechatron 2014; 20(2): 876–888.
16. Miller J and Cebon D. A high-performance pneumatic braking system for heavy vehicles. Veh Syst Dyn 2010; 48: 373–392.
17. Palanivelu S, Patil J and Jindal AK. Modeling and optimization of pneumatic brake system for commercial vehicles by model based design approach. SAE technical paper 2017-01-2493, 2017.
18. Kong X, Zhang J, Li T, et al. Research on linear pressure control of force control system of high-speed switch solenoid valve. J Mech Eng 2014; 50: 192–199.
19. Falcone P, Eric Tseng H, Borrelli F, et al. MPC-based yaw and lateral stabilization via active front steering and braking. Veh Syst Dyn 2009; 46: 611–628.
20. Lee SH, Lee YO, Kim BA, et al. Proximate model predictive control strategy for autonomous vehicle lateral control. In: Proceedings of the IEEE American control conference, Montreal, QC, Canada, 27–29 June 2012. New York, NY: IEEE.
21. Xi Y. Predictive control. 2nd ed. Beijing: National Defense Industry Press, 2013.
22. Li L, Lu Y, Wang R, et al. A three-dimensional dynamics control framework of vehicle lateral stability and rollover prevention via active braking with MPC. IEEE Trans Ind Electron 2016; 64(4): 3389–3401.
23. Cheng S, Li L, Guo HQ, et al. Longitudinal collision avoidance and lateral stability adaptive control system based on MPC of autonomous vehicles. IEEE Trans Intell Transp Syst 2019; 21(6): 2376–2385.
24. Du Q, Zhu C, Li Q, et al. Optimal path tracking control for intelligent four-wheel steering vehicles based on MPC and state estimation. Proc IMechE, Part D: J Automobile Engineering Epub ahead of print 25 October 2015. DOI: 10.1177/09544070156045318.
25. Sun X, Cai Y, Wang S, et al. Design of a hybrid model predictive controller for the vehicle height adjustment system of an electronic air suspension. Proc IMechE, Part D: J Automobile Engineering 2016; 230(11): 1504–1520.