Modeling, Control And Test Of A Novel Electronic Pneumatic Brake System

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Abstract. The brake energy recovery technology has become a key technology for electric vehicles which can significantly increase the energy economy of the electric vehicle. By-wire control of friction braking is the basic requirement for regenerative braking. For commercial vehicles with the pneumatic brake system, this article presents a novel electronic pneumatic brake system actuated by a linear motor. The electronic pneumatic brake system was modeled based on Matlab/Simulink and the accuracy of the model was verified by bench test data. Based on the analysis of model characteristic, PID and fuzzy PID controller are designed. The control effect is verified by the hardware-in-loop simulation platform of the electronic pneumatic brake system. Experimental results show that the overall performance of fuzzy PID control is better than PID control. The research can provide a friction braking force control solution for the braking energy recovery technology of an electric bus or truck using a pneumatic brake system.

1. Introduction
Brake energy recovery technology is one of the critical technologies in electric vehicles which can increase the range of electric vehicles by more than 20%(Wang at al., 2011). It has also become a hot spot in the research and development of automotive technology in recent years. In the energy recovery process, regenerative brake force can only be superimposed on the friction braking force with traditional brake structure which called parallel regenerative braking. It restricts the efficiency of energy recovery.

Many scholars have done a lot of related research to solve these problems. The research object is mainly concentrated on the hydraulic brake system of the car. Wang Jun designed an electronic air pressure control system for braking energy recovery. It was found that a reasonable braking force distribution strategy can achieve better energy recovery by using simulation analysis(Wang at al., 2013). From the above analysis, it can be seen that the existing research focuses on electronic hydraulic brake systems while less research on electronic pneumatic brakes.

Aiming at the above problems, a linear motor driven pneumatic brake-by-wire system is proposed in this paper. The PID control algorithm and the fuzzy PID control algorithm are designed. The control effect of them is tested by the hardware-in-loop simulation platform of the electronic pneumatic brake system.

2. Pneumatic brake system model
To research the characteristics of the pneumatic brake system, mathematical modeling of the pneumatic brake system is required. Brake-by-wire valve architecture is shown in Figure 1.

Port A is connected with the wheel cylinder, port P is connected with high-pressure gas source, and port T is exhaust valve. When braking, the linear motor drives the front valve rod to move and close the
exhaust valve port T. After closing the exhaust valve port, continue to push back the rear valve rod. High-pressure gas flows into port A from port P, and the pressure of the wheel cylinder begins to rise. When the linear motor force decreases, the front valve rod moves to the left under the action of the pressure feedback force. At this time, the front and rear valve rod are separated and the exhaust valve is opened. because of this, the wheel cylinder is released to the outside, and the gas flows out of port A to the atmosphere.

![Image](image1.png)

**Figure 1. brake valve-by-wire principle.**

### 2.1. Simplified Model

Due to the nonlinear factors such as friction hysteresis in the system and the pressure change process is also nonlinear, it is necessary to accurately model the system for accurate air pressure control.

The flow of gas in a pneumatic brake system is a complex variable mass thermodynamic process. The simplifications of the system are as follows.

- The compressed gas is assumed to be an ideal gas, and the gas characteristics at the high-pressure gas tank are considered constant during the braking process;
- It is assumed that the gas flow through the brake valve port is isentropic;
- The pipeline energy loss and gas potential energy change during gas flow are ignored;
- As the braking pipeline is not long, it has little effect on the pressure change, and the influence of the pipeline on the system is ignored;

For simplicity, only a quarter of the braking system is modeled, and only one brake wheel cylinder is calculated. The system is shown in Figure 2.

After the compressed gas in the air tank passes through the brake valve, it flows into the wheel cylinder whose pressure is $P_b$. The flow rate $u_0$ in the gas tank is approximately zero. Assume that the compressed air parameter pressure $P_b$, temperature $T_0$, and density $\rho_0$ in the gas tank remain constant. The cross-sectional area at the nozzle outlet is $A$, the orifice flow coefficient is $C_d$, and the air flow rate is $u$. The air pressure is $P_b$, the temperature is $T_b$, the density is $\rho_b$, the mass flow rate is $q_m$, and the adiabatic exponent of the ideal gas is $k$. It is assumed that the gas flow through the brake valve port is isentropic. The flow rate of the air at the exit is obtained from the gas isentropic flow energy equation. In addition according to the continuity equation and the unary isentropic flow equation. The mass flow equation can be obtained in the equation (1).

$$ q_m = C_d A \rho_0 \left( \frac{P_b}{P_0} \right)^{\frac{1}{k}} \sqrt{\frac{2k}{k-1} \frac{P_a}{\rho_0} \left[ 1 - \left( \frac{P_b}{P_0} \right)^{\frac{k-1}{k}} \right]} $$

(1)

By integrating the mass flow rate with time, the mass change of the gas in the wheel cylinder can be obtained. Then according to temperature and pressure in an adiabatic state and ideal gas equation(2), the pressure equation in the wheel cylinder can be obtained.

$$ \frac{T_a}{T_b} = \left( \frac{P_a}{P_b} \right)^{\frac{k-1}{k}} \frac{P_b}{P_a} + \frac{mR}{V_b} $$

(2)

In the same way, the exhaust model from the wheel cylinder to the outside can be established.

The valve opening has a great influence on the system characteristics. We perform force analysis on the brake valve. To simplify the analysis, we make the following assumptions.
• The force produced by the conical spring has a linear relation with the displacement;
• Ignore the mass the rear rod;

We can get the force analysis from Figure 3.

The force equation of the brake valve can be obtained as shown in the equation (3).

\[
\begin{bmatrix}
    m & 0 \\
    0 & M
\end{bmatrix} \begin{bmatrix}
    \ddot{x}_x \\
    \ddot{x}_3
\end{bmatrix} + \begin{bmatrix}
    -c_1 & -c_1 \\
    c_1 & c_1 + c_3
\end{bmatrix} \begin{bmatrix}
    \dot{x}_x \\
    \dot{x}_3
\end{bmatrix} + \begin{bmatrix}
    k_1 & -k_1 \\
    -k_1 & k_1 + k_3
\end{bmatrix} \begin{bmatrix}
    x_x \\
    x_3
\end{bmatrix} + \begin{bmatrix}
    F_2 + F_f + F_{air} \\
    -(F_2 + F_f + F_{air})
\end{bmatrix} = \begin{bmatrix}
    F_T \\
    0
\end{bmatrix}
\]

Here, 
- \( m \) and \( M \) are respectively the front rod mass and valve body mass.
- \( c_1 \) and \( c_3 \) are equivalent dampings between front and rear rod and body respectively.
- \( k_1 \) and \( k_3 \) are the equivalent stiffness of the return spring and the supporting spring respectively.
- \( F_2 \) is the force on the rear rod.
- \( F_f \) is the friction between the front rod and the valve body.
- \( F_T \) is the thrust acting on the front rod by the linear motor. is \( F_{air} \) air pressure.

2.2. System characteristics

In Matlab/Simulink, we can also excite the system through sine waves of different frequencies to obtain open loop characteristics. The bode diagram of the system is shown in Figure 4.

It can be seen from Figure 4 that the system bandwidth is about 15rad/s, the phase margin is 100° and the amplitude margin is 83dB. The system crossover frequency is 15rad/s. The characteristics of the system are closed to second order systems in the middle and low frequency.

The pneumatic system hardware-in-the-loop test platform as shown in Figure 5 was used to verify the accuracy of the model. The host computer uses Matlab/Simulink-RTWT to ensure the real-time performance of the system. The host computer collects the real-time air pressure signal of the wheel cylinder feedback and the thrust signal of the linear motor through the data acquisition system. The continuous step response curve from the initial state 0 to the gas source pressure and from the gas source pressure to the initial state is obtained through experiments.

From Figure 6, it can be seen from the system inflation response curve that the air pressure does not rise much during the first step. The reason is that the effective opening of the air inlet valve is reduced since the motor needs to push a distance to close the exhaust valve T. From the system deflation response curve, it can be found that the second deflation step time is longer than the other two times. Because the first step deflation air pressure drops too low. The change in friction between the valve body and the valve rod and the force exerted by the rear spool spring on the valve body becomes larger, making the
deflation process longer and slower. The deflation process is slightly slower than the inflation process because the structure of the brake valve determines that the flow area of the intake valve is greater than the flow area of the exhaust valve.

![Image](ICAMMT 2019 IOP Conf. Series: Materials Science and Engineering 631 (2019) 042030 doi:10.1088/1757-899X/631/4/042030)

Figure 5. hardware-in-the-loop platform structure.

In order to verify the accuracy of the model, we simulated and tested the open-loop characteristics of the system separately. It can be seen from Figure 6 that the self-built brake system model simulation predicts the continuous step pressure variation characteristics of the brake system.

![Image](ICAMMT 2019 IOP Conf. Series: Materials Science and Engineering 631 (2019) 042030 doi:10.1088/1757-899X/631/4/042030)

Figure 6. multi-step test and simulation comparison.

3. Air pressure control method

The dead zone is small whose characteristics of the brake valve are mainly caused by rubber plasticity on the rear valve rod. With the change of air pressure, the hysteresis characteristics of the system change greatly, and it is difficult to compensate for the hysteresis part. In this paper, the PID control algorithm and the fuzzy PID control algorithm which are insensitive to system parameters are used for model simulation and test to control the air pressure of the pneumatic brake system.

3.1. PID control algorithm

The output of the continuous PID controller is shown in the equation (4).

\[ X(t) = K_p \cdot e + K_i \cdot \int e dt + K_d \cdot \frac{de}{dt} \]  

(4)

In the equation: \( X \) is the displacement of the linear motor. \( K_p \) is the proportional coefficient, \( K_i \) is the integral coefficient, \( K_d \) is the differential coefficient, \( e \) is the error between expected displacement and actual displacement.

3.2. Fuzzy PID control algorithm

For the PID control algorithm, the three coefficients \( K_p, K_i \) and \( K_d \) cannot be changed once they are determined. When the controlled object changes with the environment or load, the control effect will become worse or even become unstable. In this paper, the fuzzy controller is used to set the three coefficients \( K_p, K_i \) and \( K_d \) online.

The input-output membership function is selected as a Gaussian membership function. PB and NB are Z-shaped and S-shaped membership functions respectively. The error \( e \) input domain is [-1,1], the
The input domain is [-1, 1], the output domain is [0, 3], and the output domain is [0, 2], and the output domain is [0, 1]. Defuzzification method adopts the maximum membership averaging method.

![Fuzzy PID control system](image)

**Figure 7.** Fuzzy PID control system.

The pressure of the gas source pressure is set as the deviation reference pressure $E_i$, and the actual error pressure is $e_i$. Normalize fuzzy input to $e = e_i / E_i$. Similarly, the fuzzy input $e_c$ is also normalized. The average of the input domain of -1 to 1 is divided into 7 parts, respectively defined as NB, NM, …., PB. Then the input membership function as shown in Figure 8.

![Membership function](image)

**Figure 8.** $e$ and $e_c$ membership function.

Fuzzy rules can be obtained as shown in Figure 9.

![Fuzzy rules](image)

**Figure 9.** Fuzzy rules.

When $|e|$ and $|e_c|$ are medium-sized, a smaller $\Delta K_p$ the value should be taken to suppress overshoot. While $|e|$ is larger, $\Delta K_p$ should be increased and $\Delta K_I$ should be taken 0, to improve the speed of system response. When $|e|$ is smaller, $\Delta K_D$ could be increased to improve the steady-state performance and enhance the anti-jamming capability of the system.

### 4. TEST RESULTS

Square wave signal and sine wave signal are used for testing respectively to observe the actual air pressure response of the system. The test list and result are shown in Table 1 and Figure 10, respectively.
Figure 10. tracking test result.

Table 1. Air pressure control method verification test.

| Test               | Description                                                                 |
|--------------------|----------------------------------------------------------------------------|
| Square wave tracking | Offset pressure 0.25MPa, amplitude 0.15MPa, frequency 0.2Hz                |
| Sine wave tracking   | Offset pressure 0.3MPa, amplitude 0.2MPa, frequency 0.5Hz                  |

Table 2. Tracking effect comparison.

| Test                | Inflating stage | Deflating stage |
|---------------------|-----------------|-----------------|
|                     | Response time   | Setting time    | Overshoot | Response time |
| Fuzzy PID           | 0.56 s          | 0.70 s          | 3.07%     | 0.79 s        |
| PID                 | 0.79 s          | 1.04 s          | 2.99%     | 1.15 s        |
|                     | Peak time delay | Mean-root-square error | |
| Fuzzy PID           | 0.284s          | 0.0041          |
| PID                 | 0.398s          | 0.0067          |

The test charts can be obtained as shown in Figure 10 and Table 2. It can be found that both algorithms can better track square wave signals. The response time and setting time of fuzzy PID control are much smaller than that of ordinary PID control, but the overshoot is almost the same. This shows that the system characteristics of the fuzzy PID controller are significantly better than ordinary PID controller. From the inflation stage and the deflation stage, the charging and discharging characteristics of the system are not consistent. The deflation speed is significantly slower than the inflation rate and the response time is longer, which is related to the structure of the pneumatic valve. Obviously, the PID coefficient that is over-adjusted during the inflation phase will not cause overshoot during the deflation phase. The sine wave signals tracking experiment shows that the fuzzy PID controller has a smaller steady-state error than the PID controller and the peak lag time is much smaller. This shows that the phase frequency characteristic of the fuzzy PID controller is better than the ordinary PID controller.

5. Conclusion

It is difficult to control the air pressure of the non-linearly controlled electronic pneumatic brake system. After the brake system is properly simplified, a high precision brake wheel cylinder pressure model is built. The PID controller and fuzzy PID controller are designed to control the air pressure. And the hardware-in-the-loop simulation platform is used to verify the effectiveness of the two control methods. Experiments show that the Matlab/Simulink model has a good agreement with the actual system. Both controllers are effective in controlling the air pressure to follow the desired pressure. The control effect of the fuzzy PID controller is better than that of the PID controller.

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