Study on simulation model and performance test of locomotive anti-skid valve

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Abstract. When studying the comprehensive performance of the anti-skid valve, the test of the curve characteristic lacks theoretical basis in the brake anti-skid process and the mathematical equations are difficult to derive. Take a domestic locomotive anti-skid valve as the research object. Combine with the physical structure parameters of the anti-skid valve and the working principle of inflating and exhausting. A simulation model of the anti-skid valve was established by using MATLAB/Simulink, which included the motion equation of valve core and diaphragm plate, electromagnetic suction equation of electromagnet and air chamber charging and exhausting equation. The characteristics of the air pressure curve of the anti-skid valve under different control signals were analyzed. In order to verify the correctness of the simulation model, a set of locomotive anti-skid valve comprehensive performance test bench based on STM32 was built for experimental verification. The results show that the simulation results are consistent with the experimental results, meet the test requirements, and verify the accuracy of the model. This model provides a theoretical basis for structural optimization of anti-skid valves and analysis of locomotive braking anti-skid performance.

1. Introduction

The essence of the locomotive anti-skid valve is a pneumatically controlled high-speed solenoid valve. In the anti-skid control system, the anti-skid valve is an important actuator. It mainly receives the anti-skid commands given by the anti-skid brake control system and adjusts the pressure of the brake cylinder to achieve anti-skid control of the locomotive [1]. The comprehensive performance of the anti-skid valve directly affects the braking distance and anti-skid performance of the locomotive, and plays a key role in the driving safety of the locomotive. Therefore, according to TKQ600-DM01 "Anti-skid Valve Test Specification [2]" and TB/T 3009-2011 "Anti-skid device for railway passenger car and powered car train-set [3]", the curve characteristics of anti-slide valve under different test control signals are studied, which provides an important theoretical role for factory inspection and structural improvement of anti-slide valve.

The current simulation study of anti-skid valves is as follows. Li Bangguo and others established a simulation model of anti-skid valve based on AMESim, which verified the air charging characteristics and mitigation characteristics of the anti-skid valve [4]. Lu Qiang and others used AMESim to establish a brake system with the anti-skid valve, and analyzed the brake cylinder pressure curve after applying the anti-skid valve during the braking process [5]. Liao Zhijian and others used AMESim to customize the intake and exhaust valves of the anti-skid valve, and simplified the model of the anti-skid system [6]. Ho-Yeon Kim and Chul-Goo Kang proposed a HILS system with a venting valve and
developed advanced anti-skid logic [7]. The above simulation model focused on the physical system design, and the operating characteristics of the anti-skid valve under different control states were not comprehensive. Therefore, the mathematical model was established by using MATLAB/Simulink, and the simulation was used to analyse the air pressure curve of the anti-skid valve under the test standard control signal, which played an important role in studying the physical structure and comprehensive performance of the individual components of the anti-skid valve. At the same time, in order to verify the accuracy of the simulation, the locomotive anti-skid valve comprehensive performance test bench based on STM32 was built for experimental verification.

2. Anti-skid valve structure and working principle
Take a certain type of anti-skid valve of domestic locomotive as the detection object, and its structural schematic diagram is shown in Figure 1. The valve is provided with three ports, the port D is an input port and is connected to the output of the brake control device; the port C is an output port and is connected to the brake cylinder; the port O is an exhaust port. When the movable iron core VM1 and VM2 do not operate, the diaphragm e is opened, and the gas enters the brake cylinder from the D port through the C port, and the anti-skid valve is in an inflated state (a). When both VM1 and VM2 are activated, the diaphragm h is opened, and the gas in the brake cylinder is discharged from the O port through the C port, and the anti-skid valve is in the exhaust state (b). When VM1 does not operate and VM2 moves, the diaphragm is both closed, and the pressure at port C remains unchanged. At this time, the anti-skid valve is in a pressure-holding state (c).

![Figure 1. Schematic diagram of the anti-skid valve structure.](image)

3. Simulation modeling of anti-skid valve

3.1. Mathematical formula for simulation

3.1.1. Equation of motion. When the anti-skid valve was in operation, the moving iron cores VM1 and VM2 were subjected to electromagnetic force, spring force, and air pressure. The equations of motion for VM1 and VM2 were shown in Equations (1) and (2), which formed moving iron core VM1 and VM2 models. At the same time, the left and right diaphragms were subjected to air pressure and return spring force. The equations of motion for the left and right diaphragms were shown in Equations (3) and (4), which formed inlet diaphragm model and exhaust diaphragm model.

\[ m_1\ddot{x}_1 = F_{e1} + P_1A_1 - k_1(x_1 + x_{10}) - c\dot{x}_1 \]  \hspace{1cm} (1)

\[ m_2\ddot{x}_2 = F_{e2} - P_2A_2 - k_2(x_2 + x_{20}) - c\dot{x}_2 \]  \hspace{1cm} (2)

Among them: \( m_1 \) and \( m_2 \) were the mass of the moving iron core; \( F_{e1} \) and \( F_{e2} \) were the electromagnetic forces received by the moving iron core; \( P_1 \) and \( P_2 \) were air pressures; \( k_1 \) and \( k_2 \) was
the spring stiffness; $x_1$ and $x_2$ were the displacement of the moving iron core; $x_{10}$ and $x_{20}$ were the preload of the return spring; $c$ was the motion damping of the spool.

$$m_3\ddot{x}_3 = P_3A_{3r} - k_3(x_3 + x_{30}) - c\dot{x}_3 - P_3A_{3r}$$

$$m_4\ddot{x}_4 = P_4A_{4l} - P_4A_{4r} - c\dot{x}_4$$

Among them: $m_3$ and $m_4$ were the mass of the inlet and outlet diaphragm; $P_l$ and $P_r$ were the pressure of the left and right chambers of the diaphragm; $k_3$ was the spring stiffness; $x_3$ and $x_4$ were the displacement of the diaphragm; $x_{30}$ was the preload of the return spring; $c$ was the motion damping of the spool.

### 3.1.2. Electromagnetic equation

According to Maxwell's electromagnetic force equation, Kirchhoff's law and Biot-Savart's law [8]. The electromagnet electromagnetic force equation was expressed by Equation (5), which formed electromagnetic model.

$$F_e = \frac{B^2S}{2u_0} = \frac{1}{2} \frac{N^2U^2u_0S}{R^2r^2}$$

Among them: $F_e$ was the electromagnet suction; $B$ was the magnetic flux density of the air gap; $r$ was the air gap; $u_0$ was the permeability in vacuum, $u_0 = 4\pi \times 10^{-7}\text{H/m}$; $S$ was the cross-sectional area of the moving iron core; $N$ was the number of turns of the coil; $U$ was the input voltage; $R$ was the coil resistance.

### 3.1.3. Gas path equation

It was assumed that there was no gas leakage inside the anti-skid valve, and the air-filling and exhausting process was regarded as an adiabatic process. According to the mass flow equation and the ideal gas equation of the gas flowing through the orifice, the dynamic differential Equation [9] of the gas chamber charging and discharging process was obtained, which was shown in Equation (6). Besides, this equation formed chamber models, aeration and exhaust model.

$$\frac{dp}{dt} = \begin{cases} 
\frac{kA_p}{V} \left( 2 \frac{p}{k + 1} \right)^{\frac{0.5}{k + 1}} \sqrt{\frac{2kR_T}{k + 1}} \left( 0 \leq p / p_0 < 0.528 \right) \\
\frac{kA_p}{V} \left( \frac{2kR_T}{k - 1} \right) \left( \frac{p}{p_0} \right) \left( \frac{p}{p_0} \right)^{\frac{1}{k}} \left( 0.528 \leq p / p_0 \leq 1 \right) 
\end{cases}$$

Among them: $k$ was the adiabatic coefficient, $k = 1.4$; $R_o$ was the gas constant, $R_o = 287.1\text{J/(kg} \cdot \text{K)}$. $T$ was the absolute temperature of the gas, $T = 313\text{K}$; $V$ was the volume of the gas chamber; $p_0$ was the input pressure; $p$ was the output pressure; $A$ was the area of the flow channel.

### 3.2. Establishment of simulation model

Combined with the above theoretical formula, the mathematical model of the anti-skid valve was established based on MATLAB/Simulink software, as shown in Figure 2.

The control signal consists of high and low level signals of different lengths of time. And the image display shows the air pressure curve under different performance test methods. According to the internal structural parameters of the anti-skid valve, set the parameters of the anti-skid valve model, as shown in Table 1.
Figure 2. Simulation model of the anti-skid valve.

Table 1. Internal structural parameters of anti-skid valve.

| Parameter | Value | Parameter | Value |
|-----------|-------|-----------|-------|
| U/V       | 24    | $A_{3r}, A_{4r} / \text{mm}^2$ | $\pi \cdot 18^2$ |
| N/turns   | 200   | $A_{3l}, A_{4l} / \text{mm}^2$ | $\pi \cdot 10^2$ |
| r/mm      | 0.1   | $m_3, m_2 / \text{kg}$ | 0.003 |
| S/mm$^2$  | $\pi \cdot 1.7^2$ | $m_3, m_4 / \text{kg}$ | 0.007 |
| $k_1, k_2$ /N/m | 600   | $x_1, x_2 / \text{mm}$ | 1.1 |
| $k_3$ /N/m | 1800  | $x_3, x_4 / \text{mm}$ | 0.9 |
| $x_{10}, x_{20}, x_{30}$ /mm | 8     | $A / \text{mm}$ | $\pi \cdot 6^2$ |

4. Experimental verification of comprehensive performance of anti-skid valve

4.1. Analysis of performance test methods

Sealing test: If there is gas outflow when the valve is in the closed state, this is the amount of leakage, and the amount of leakage greatly affects the performance characteristics of the valve. Without the 6L gas storage tank, after the anti-skid valve is aerated to the highest working pressure, stabilize for 1 minute and hold for 1 minute. The change in air pressure of the anti-skid valve within 60s is the amount of leakage when the valve is not loaded with 6L. And it is required to be less than 5kPa. With 6L gas storage tank, after the anti-skid valve is aerated to the highest working pressure, the exhaust gas is exhausted to 200kPa. Then, stabilize for 1 minute and hold for 1 minute. The change in air pressure of the anti-skid valve within 60s is the amount of leakage when the valve is not loaded with 6L. And it is required to be less than 2.5kPa.

Action response time test: This performance affects the reaction speed of the anti-skid action of the anti-skid valve. When the anti-skid valve receives the aeration signal, the time that the pressure value of the output port C rises from 0kPa to 80% of the maximum working pressure is the aeration response time. And it is required to be less than 45ms. When the anti-skid valve receives the exhaust signal, the time that the pressure value of the output port C decreases from the maximum working pressure to 30% of the value is the response time of mitigation. And it is required to be less than 40ms.

Staged aeration and exhaust Performance Test: This property affects the ability of the anti-skid valve to perform repeated actions. The anti-skid valve is inflated for 0.1s, the pressure is maintained for 0.9s, and it is carried out 5 times. After holding for 3s, the pressure value of the brake cylinder indicates the stage inflation performance. And it is required to be more than 150kPa. The anti-skid valve is exhausted for 0.1s, the pressure is maintained for 0.9s, and it is carried out 5 times. After
holding for 4s, the pressure value of the brake cylinder indicates the stage exhaust performance. And it is required to be less than 350kPa.

Rapid aeration and exhaust performance testing: This performance affects the ability of the anti-skid valve to quickly eliminate slip and resume braking. The time that the brake cylinder pressure value rises from 0 to 90% of the maximum working pressure is the rapid aeration time. And it is required to be less than 1.3s. The anti-skid valve exhausts for 1.2 s, and after holding for 3 s, the brake cylinder pressure value indicates the rapid exhaust performance. And it is required to be less than 80kPa.

Full mitigation performance and emptying performance test: This performance affects the ability of the anti-skid valve to restore braking and mitigation. After the full mitigation process of exhausting gas for 2s and holding for 5s, the pressure of the brake cylinder indicates full mitigation performance. And it is required to be less than 20kPa. When the brake cylinder is exhausted through the anti-skid valve for 10s, the pressure of the brake cylinder and the time decreasing from the maximum working pressure to 50kPa indicate the emptying performance. And the pressure is required to be less than 2.5kPa, the time is required to be less than 3.2s.

The above brake cylinders are replaced by 6L air tanks in order to simulate the actual load conditions when the locomotive brakes.

4.2. Overall design of the test bench

1-Gas source 2-Filter and pressure reducing valve 3-40L gas storage tank 4,5-Pressure regulating valve 6,7,8,9-Air control valve 6',7',8',9'-Two-position two-way solenoid valve 10-Silencer 11,12,13-Air pressure sensor 14-6L load storage tank

**Figure 3.** Overall design of the test bench.

The comprehensive performance test bench for locomotive anti-skid valve based on STM32 was designed, as shown in Figure 3. The detection system adopted STM32 as the main controller, including test air circuit, signal conditioning circuit, power drive circuit and touch screen. The gas path of the detection system was composed of a pressure regulating valve 4, pneumatic valves 6, 7, 8, 9, 10, air pressure sensors 11, 12, 13 and 6L load gas storage tank 14. The signal conditioning circuit was responsible for converting the current signal 4~20mA of the sensor pressure into a voltage signal of 0~3.3V, and the converted voltage signal was collected by the internal AD converter of the main controller. The anti-skid valve had a rated power of 8W and an operating voltage of DC24V. Therefore, the power drive circuit with a maximum current of 3A could be used in practical applications. When the system started normally, the test item was selected through the touch screen, and the test parameters and start signal were transmitted to the STM32 through the RS232 serial port. On the basis of the above test method, the STM32 emitted a series of signals, then the solenoid valves and the anti-skid valve acted according to the test procedure. At the same time, the air pressure sensor
continuously collected the air pressure and sent them to the touch screen. The touch screen drew curves, displayed test results and saved them in the SD card via FATFS.

5. Results and analysis

5.1. Sealing test
The anti-skid valve was tested by the direct pressure method. The initial pressure of the anti-skid valve was 500.6kPa and the end pressure was 498.4kPa. The pressure dropped by 2.2kPa and less than 5kPa, meeting the requirements for sealing test without using a 6L gas storage tank. Similarly, the initial pressure of the anti-skid valve was 200kPa, and the termination pressure was 199.3kPa. The pressure dropped by 0.7kPa and less than 2.5kPa, meeting the requirements for sealing with a 6L gas storage tank.

5.2. Action response time test analysis
Figure 4 showed the test results of measuring the response time of the anti-skid valve. The simulation curve was similar to the test curve. It could be seen from the Figure 4 that the aeration response time was measured to be 42.66ms, and the simulation result was 43.60ms, which were less than 45ms; the exhaust response time was measured to be 36.66ms, and the simulation result was 35.20ms, which were less than 40ms. Meet the action response time requirements. Among them, there was slight jitter during the inflation of the air chamber, because the air volume of the anti-skid valve chamber was small, and the high-pressure hose connected to the test air circuit was bent or crossed to cause pressure instability.

5.3. Staged aeration and exhaust performance test analysis
Figures 5 and 6 were the test results of measuring the staged aeration and exhaust performance of the anti-skid valve. The simulation curve was basically consistent with the test curve. After the staged aeration process, the pressure of the 6L gas storage tank was measured to be 187.901kPa, and the simulation result was 191.7kPa, which were higher than 150kPa. After the staged exhaust process, the pressure of the 6L gas storage tank was measured to be 321.1181kPa, and the simulation result was 315.5kPa, which were lower than 350kPa. The results met the test requirements.

5.4. Rapid aeration and exhaust performance test analysis
Figures 7 and 8 were the test results for measuring the rapid aeration and exhaust performance of the anti-skid valve. The simulation curve was basically consistent with the test curve. After the rapid aeration process, the rapid aeration time was measured as 1.20s, and the simulation result was 1.174s, which were less than 1.3s; after the rapid exhaust process, the pressure of the 6L gas storage tank of
the simulated brake cylinder was measured to be 40.588kPa, and the simulation result was 41.291kPa, which were less than 80kPa. The results met the test requirements.

5.5. Total mitigation and emptying performance test analysis

Figures 9 and 10 showed the test results of measuring the full mitigation and emptying performance of the anti-skid valve. The simulation curve was basically consistent with the test curve. After the full mitigation process, the pressure of the 6L gas storage tank of the simulated brake cylinder was measured to be 16.684kPa, and the simulation result was 16.666kPa, which were lower than 20kPa. After the emptying process, the pressure of the 6L gas storage tank of the simulated brake cylinder was measured to be 1.024kPa, and the simulation result was 0.158kPa, which were less than 2.5kPa, and the time from the pressure of 500kPa to 50kPa was measured to be 1.26s, and the simulation result was 1.21s, which were less than 3.2 s. The test results met the requirements.

Figure 5. Staged aeration performance test chart.

Figure 6. Staged exhaust performance test chart.

Figure 7. Rapid aeration performance test chart.

Figure 8. Rapid exhaust performance test chart.

Figure 9. Full mitigation performance test chart.

Figure 10. Emptying performance test chart.
6. Conclusions
The dynamic simulation model of the anti-skid valve was established by using MATLAB/Simulink. The characteristics of the anti-skid valve under different control signals were analysed by simulation, and the accuracy of the model was verified by the test bench test. The results showed that the simulation curve of the anti-skid valve performance was basically consistent with the test curve. This model could provide theoretical support for the study of locomotive braking system.

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References
[1] Liu RC 2018 The Motor Group Microcomputer Controls the Work of the Skater Digital Technology and Application 1 9-10
[2] TKQ600-DM01 2011 Anti-skid Valve Test Specification Locomotive & Car Research Institute
[3] TB/T3009-2011 2011 Anti-skid device for railway passenger car and powered car train-set Ministry of Railways of the People's Republic of China
[4] Li BG and FAN RW 2011 Modeling and Simulation Analysis of Anti-skid Valve International Railway Locomotive & Car 5 128-131
[5] Lu Q and Yang MC 2011 Modeling and Simulation Methodology of Pneumatic Braking System in Metro Vehicles Using AMESim Hydraulics Pneumatics & Seals 10 31
[6] Liao ZJ and Liu ZP 2012 Custom modeling and simulation for anti-skid braking system of EMUs based on AMESim Electric Locomotives & Mass Transit Vehicles 4 29-32
[7] Kim HY and Kang CG 2009 Real-time Simulation for Dynamic Characteristics of Mechanical Braking of the Korean Tilting Train Transactions of the Korean Society of Mechanical Engineers 11 1294-1299
[8] Mei and Liu JL 2012 Calculation of Electromagnet Attractive Force and Simulation Analysis Micromotors 6 6-9
[9] Fan WJ and Zhang JX 2016 Simulation and Design of Dual-channel Gas-liquid Press Machine Transactions of the Chinese Society for Agricultural Machinery 1 391-396