Finite Element Simulation of TFY-YH Anti-slip Device

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Abstract. Anti-slip device is a kind of parking anti-slip device used in arrival and departure lines, intermediate stations, special lines, loading lines and locomotive depot incoming and outgoing lines. With the development of railway technology, the automation and intelligence level of anti-slip system have been continuously improved, and it has gradually become one of the main means of parking and anti-slip. The dynamic response characteristic of TFY-YH anti-slip system is studied, the finite element simulation model of anti-slip system is constructed, the displacement of piston rod, the time of piston rod in place, the speed of piston rod and the change of oil pressure of accumulator in the process of braking and relieving of anti-slip system are systematically studied, the pressure holding performance of anti-slip system is analyzed, and the method to improve the pressure holding ability is put forward, and the relationship between the stop position of train and the time of braking and relieving in place is obtained, and a method to locate the stop position of train is proposed, which lays a certain theoretical foundation for intelligent control and monitoring of anti-slip system.

Keywords: Hydraulic system; Anti-slip system; Finite element simulation.

1 Introduction

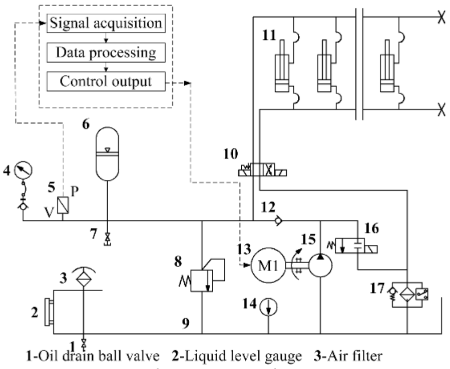
With the rapid development of modern industrial technology, railway equipment is becoming more and more complicated, heavy-duty and intelligent. Anti-slip device is an important technical equipment to prevent locomotives or trains from slipping away due to strong winds and slopes when they stay at rest. The anti-slip device can stop and prevent slip by friction braking on the side of wheels. Compared with traditional parking roofs and anti-slip iron shoes, anti-slip devices have the characteristics of more monitoring means, less labor intensity and less safety risks. At present, the widely used parking anti-slip equipment has a low degree of automation and intelligence, and the theoretical basis for layout and installation is insufficient [1].

In order to further improve the automation and intelligence of parking anti-slip devices, enrich the monitoring means of parking anti-slip devices, and improve the theoretical basis for the installation and layout of anti-slip devices, domestic scientific research institutions have carried out a lot of research on parking anti-slip devices. Fu Bo et al. [2], studied the control system of TDDH parking anti-slip device, put forward the control system and prediction system scheme of anti-slip device, and combined the parking anti-slip device control system with SAM control system, which improved the centralized control and automatic control ability and operation efficiency of the station, reduced manual operation and improved the safety of shunting operation. Wang Zheyao [3], studied the layout mode of parking anti-slip devices, simulated the actual sliding process at the tail of marshalling yard by using the simulation software of parking anti-slip devices at the tail of marshalling yard, obtained the influence of parking anti-slip devices layout mode on anti-slip effect under different working conditions, and put forward suggestions on the layout of parking anti-slip devices, which provided theoretical basis and data reference for the layout scheme of parking anti-slip devices at the tail of marshalling yard. Chen Lichun [4] et al., studied the braking process of TDDH parking anti-slip device, established the finite element simulation model of TDDH anti-slip device, and analyzed the contact stress distribution between the brake rail and the wheel of the anti-slip device, which provided a reference for the accuracy of braking capacity calculation of the anti-slip device. Bao Lukun [5] et al., studied the layout of parking anti-slip devices on the arrival and departure lines of stations, designed the simulation program of parking anti-slip devices on the arrival and departure lines, calculated the action probability of parking anti-slip devices when typical vehicles are in the coverage area of parking anti-slip devices, and obtained the optimal layout scheme of parking anti-slip devices according to the requirements of locomotive stay displacement and related safety distance. Niu Yanhua [6] et al., studied the mechanical behavior of the parking device in the braking process, established a three-dimensional model of the parking device, and obtained the relationship between braking force, braking height and braking gauge through theoretical calculation and analysis. Jia Jifeng [7] et al., studied the braking ability of TTK-92 anti-slip device, established the calculation model of anti-slip device, calculated the main technical parameters and braking ability of anti-slip device, obtained the theoretical calculation method of braking ability of anti-slip device, and optimized the design cycle of anti-slip device. Overall, the current research on anti slip devices is relatively macroscopic and single, and the basic principles and functions of anti slip devices are not fully studied, making it difficult to provide accurate basis for the intelligent control and monitoring of anti slip device systems.

In response to the above issues, this article constructs a finite element simulation model of the anti slip system for the TFY-YH type anti slip device. The system analyzes the piston rod displacement, piston rod in place time, piston rod movement speed, and accumulator oil pressure changes during the braking and release process of the anti slip device system. The relationship between the train stop position and the anti slip device braking and release time is obtained, laying a certain foundation for the automation and intelligence of the anti slip device system.

2 Structure and working principle of TFY-YH anti-slip device system

TFY-YH anti-slip system is mainly composed of hydraulic components such as motor, gear pump, oil cylinder, accumulator, relief valve, solenoid valve, one-way valve, etc. The working principle is shown in Figure 1. Among them, the oil cylinder is the core actuator of the anti-slip device, which is rigidly connected with the brake rail through the cylinder body connector and piston rod connector. According to the working schematic diagram of the anti-slip device system, the arrangement of the oil cylinder is cylinder 1-6 from right to left, and the arrangement structure of the anti-slip device system is shown in Figure 2.



1-Oil drain ball valve **2**-Liquid level gauge **3**-Air filter
4-Pressure gauge and Pressure gauge adapter **5**-Pressure sensor
6-Accumulator, **7**-Oil drain ball valve **8**-Relief valve **9**-Oil tank
10-Two-position four-way solenoid valve **11**-Hydraulic cylinder
12-One-way hydraulic valve **13**-Motor and Coupling
14-Temperature sensor **15**-Gear pump
16-Two-position two-way solenoid valve **17**-Oil return filter

Fig. 1 Working schematic diagram of TFY-YH anti-slip device system

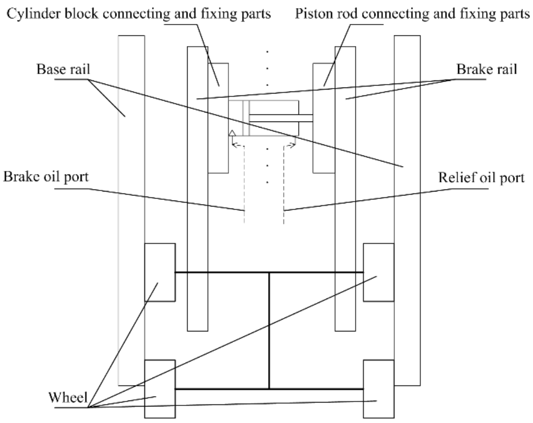


Fig. 2 Schematic diagram of anti-slip device system layout structure

TFY-YH anti-slip system has two working states: braking state and relieving state. According to the whole action process, it can be divided into four stages: starting, braking, pressure maintaining and relieving stage. The working principle of each stage is as follows:

(1) Start-up stage: When the anti-slip system receives the braking (or relief) command, the hydraulic system enters the starting state, the electromagnet of the two-position two-way solenoid valve is electrified, and the solenoid valve is opened. At this time, the motor drives the gear pump to rotate, so that the oil flows through the two-position two-way solenoid valve, and then returns to the oil tank through the oil return filter to reduce the damage of the load to the gear pump. When the gear pump is filled with oil and runs stably, the electromagnet of the two-position two-way solenoid valve loses power, the oil outlet of the gear pump is disconnected from the oil tank, and the anti-slip system enters a state to be braked (or relieved).

(2) Braking stage: When the anti-slip system receives the braking command, the hydraulic system passes through the starting stage first, the electromagnet of the two-position four-way solenoid valve gets electricity, the oil outlet of the gear pump communicates with the oil cylinder braking chamber, the volume of the oil cylinder braking chamber gradually increases, the volume of the oil cylinder relief chamber gradually decreases, and the oil in the relief chamber returns to the oil tank through the oil return filter. At the same time, the oil outlet of the gear pump is connected with the accumulator through a one-way valve to supply oil to the accumulator. When the piston rod of the oil cylinder extends out to push the execution terminal to move in place, the anti-slip system reaches the braking state.

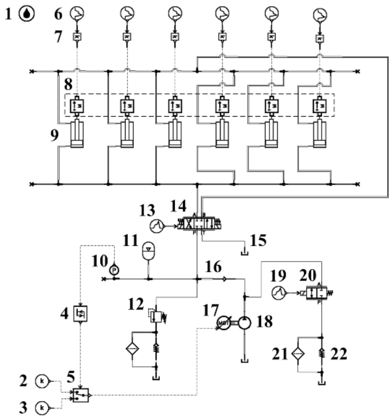
(3) Pressure holding stage: When the piston rod of the oil cylinder pushes the execution terminal to move in place in the braking stage, and the accumulator reaches the set pressure, the two-position four-way solenoid valve loses power, the position of the solenoid valve remains unchanged, the motor stops rotating, and the anti-slip system enters the pressure holding stage. In the pressure holding stage, the oil pressure sensor can monitor the main oil circuit pressure. When the main oil circuit pressure is lower than the set value, the motor starts again to charge the accumulator.

(4) Relieving stage: When the anti-slip system receives the relief command, the hydraulic system passes through the starting stage first, the electromagnet of the two-position four-way solenoid valve gets electricity, the oil outlet of the gear pump communicates with the relief cavity of the oil cylinder, the volume of the relief cavity of the oil cylinder gradually increases, the volume of the brake cavity of the oil cylinder gradually decreases, and the oil in the brake cavity returns to the oil tank through the oil return filter. When the cylinder piston rod is retracted in place, the electromagnet of the two-position four-way solenoid valve loses power, and the position of the solenoid valve remains unchanged.

3 Simulation Modeling of TFY-YH Anti-slip System

According to the working principle of TFY-YH anti-slip system, the finite element simulation model of anti-slip system is built in Simcenter Amesim software by using

Hydraulic Component Design, Mechanical Component Design and Signal Control Component Design, as shown in Figure 3 below.



1-Hydraulic oil module **2**、**3**、**6**、**13**、**19**-Signal source module
4-Trigger module **5**-Switch module **7**-Amplifier module
8- Quality module **9**-Hydraulic cylinder **10**-Pressure sensor
11-Accumulator **12**-Relief valve
14-Two-position four-way solenoid valve **15**-Oil tank
16-One-way hydraulic valve **17**-Motor and Coupling
18-Gear pump **20**-Two-position two-way solenoid valve
21-Oil return filter **22**-Hydraulic control one-way valve

Fig. 3 Simulation model of TFY-YH anti-slip device system

Gear pump is the core component of power output of anti-slip system, and its nominal volume flow and output flow can be characterized as:

$$q_p = V_p \cdot n_p \tag{1}$$

$$q_{out} = q_p \cdot \frac{\rho(p_{up})}{\rho(0)} \tag{2}$$

In the formula, q_p is the nominal volume flow (cc/min) of gear pump, V_p is the displacement of gear pump (cc/rev), n_p is the typical speed of gear pump (rev/min), q_{out} is the output flow of gear pump (cc/min), and $\frac{\rho(p_{up})}{\rho(0)}$ is the density ratio of rated pressure to 0 pressure.

The oil cylinder is the core component of the anti-slip system, and its output force and oil pressure changes at the oil inlet of the braking chamber and relief chamber are calculated as follows:

Considering the influence of oil pressure and viscous friction on the piston rod of the oil cylinder, the output force of the oil cylinder can be characterized as:

$$F_{rod} = p_1 A_1 - p_2 A_2 + v \cdot visc \tag{3}$$

$$A_1 = \frac{\pi \cdot d_p^2}{4} \tag{4}$$

$$A_2 = \frac{\pi \cdot (d_p^2 - d_r^2)}{4} \tag{5}$$

In the formula, F_{rod} is the output force of the oil cylinder, p_1 is the inlet pressure of the brake chamber, p_2 is the inlet pressure of the relief chamber, A_1 is the cross-sectional area of the piston acting on the p_1 inlet pressure of the brake chamber, A_2 is the annular area acting on the p_2 inlet pressure of the relief chamber, v is the moving speed of the piston rod, $visc$ is the viscous friction coefficient, d_p is the diameter of the piston and d_r is the diameter of the piston rod.
Considering the influence of internal leakage of oil cylinder on the oil pressure at the inlet of oil brake chamber and relief chamber of oil cylinder, the leakage flow rate of oil cylinder can be characterized as:

$$q_{leak} = (p_1 - p_2) \cdot leak \tag{6}$$

where, $leak$ is the leakage coefficient of the oil cylinder.
The oil volume in the brake chamber can be characterized as:

$$V_1 = A_1 \cdot x_{act} + dead_1 \tag{7}$$

where, x_{act} is the displacement of piston rod and $dead_1$ is the dead space volume of brake chamber.
The inflow flow in the brake chamber is:

$$Q_{in1} = q_1 - q_{leak} - v \cdot A_1 \cdot \frac{\rho(p_1)}{\rho(0)} \tag{8}$$

In the formula, q_1 is the flow rate at the entrance of the brake chamber of the oil cylinder, q_{leak} is the leakage flow rate of the oil cylinder, and $\frac{\rho(p_1)}{\rho(0)}$ is the density ratio of the oil pressure at the entrance of the brake chamber of the oil cylinder to the pressure of 0.
The derivative of the pressure at the inlet of the brake chamber of the oil cylinder with respect to time, i.e. the rate of change of pressure with time, can be obtained through the oil pressure at the inlet of the brake chamber, the oil volume in the brake chamber and the inflow flow in the brake chamber:

$$B_{wall}(p_1) = \frac{1 + w_{comp} \cdot p_1}{w_{comp}} \tag{9}$$

In the formula, $B_{wall}(p_1)$ is the wall modulus under the inlet pressure (p_1) of the brake chamber and w_{comp} is the compliance coefficient of the wall.

$$\frac{1}{B_{eff}(p_1)} = \frac{1}{B_{fluid}(p_1)} + \frac{1}{B_{wall}(p_1)} \tag{10}$$

In the formula, $B_{eff}(p_1)$ is the effective bulk modulus under the inlet pressure (p_1) of the brake chamber and $B_{fluid}(p_1)$ is the bulk modulus of the fluid under the inlet pressure (p_1) of the brake chamber.
The effective volume and effective flow rate of fluid under the inlet pressure (p_1) of brake chamber can be characterized as:

$$V_{eff}(p_1) = V_1 \cdot \left(1 + w_{comp} \cdot p_1\right) \tag{11}$$

$$Q(p_1) = Q_{in1} \cdot \frac{\rho(0)}{\rho(p_1)} \tag{12}$$

To sum up, the change rate of oil inlet pressure of oil cylinder brake chamber with time can be characterized as:

$$\frac{dp_1}{dt} = \frac{B_{eff}(p_1) \cdot Q(p_1)}{V_{eff}(p_1)} \tag{13}$$

The oil volume in the relief chamber can be characterized as:

$$V_2 = A_2 \cdot (stroke - x_{act}) + dead_2 \tag{14}$$

In the formula, $stroke$ is the stroke of the oil cylinder and $dead_2$ is the dead space volume of the relief cavity.

The inflow flow in the relief cavity is:

$$Q_{in2} = q_2 - q_{leak} - v \cdot A_2 \cdot \frac{\rho(p_2)}{\rho(0)} \tag{15}$$

In the formula, q_2 is the flow rate at the entrance of the brake chamber of the oil cylinder and $\frac{\rho(p_2)}{\rho(0)}$ is the density ratio of the oil pressure at the entrance of the relief chamber of the oil cylinder to the pressure of 0.

Similarly, the change rate of the pressure at the inlet of the relief chamber of the oil cylinder with time can be obtained by relieving the oil pressure (p_2) at the inlet of the relief chamber, the oil volume (V_2) in the relief chamber and the inflow flow (Q_{in2}) in the relief chamber, namely:

$$\frac{dp_2}{dt} = \frac{B_{eff}(p_2) \cdot Q(p_2)}{V_{eff}(p_2)} \tag{16}$$

4 Simulation analysis of anti-slip system

Through the analysis and research on the working principle of TFY-YH anti-slip device system in the third part, the equations of oil pressure, oil volume and output pressure of anti-slip device system are obtained, and the above equations are input into the finite element simulation model of anti-slip device, and the key simulation parameters are set according to TFY-YH anti-slip device system, and the finite element simulation is carried out. The key parameters of TFY-YH anti-slip system setting simulation are shown in the following table.

Table 1. Key parameters of anti-slip system simulation.

Parameter name	Parameter units	Parameter values
Gear pump	Displacement (ml/r)	8
	Rated speed (r/min)	1500
Motor	Rotating speed (r/min)	1430
Accumulator	Nominal pressure (MPa)	20
	Nominal volume (L)	10
Relief valve	Opening pressure (MPa)	10
	Flow capacity (L/min)	45
Oil cylinder	Piston diameter (mm)	100

Piston rod diameter (mm)	70
Working trip (mm)	150
Piston rod mass (kg)	10

4.1 Dynamic Characteristics of Anti-slip System

The signal inputs for setting signal sources 19 and 13 are shown in figs. 4 and 5, respectively. Among them, the signal source 19 is the input signal of the two-position two-way solenoid valve, which is opened for 2s when the motor is just started, so that the gear pump can be quickly filled with oil and reduce the wear of the gear pump; The signal source 13 is the input signal of the two-position four-way solenoid valve. When the anti-slip system brakes, the two-position four-way solenoid valve opens to the left (-40mA), and when the anti-slip system relieves, the two-position four-way solenoid valve opens to the right (-40mA). The braking and relieving signal is set to 50s. Set the simulation time to 200s and the time interval to 0.1 s.
According to the load force measured by the test, set the signal source 6 (cylinder load curve) as shown in Figure 6.
Set the simulation time of the anti-slip system to 200s, that is, realize the braking and relieving of the anti-slip system twice.

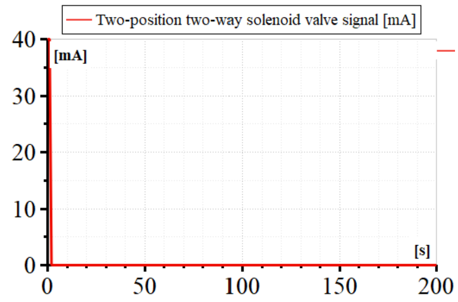


Fig. 4 Signal of two-position two-way solenoid valve

Fig. 5 Signal of two-position four-way solenoid valve

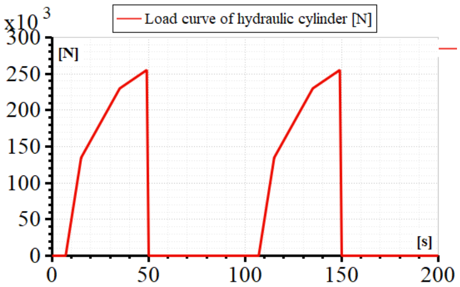


Fig. 6 Cylinder Load Curve

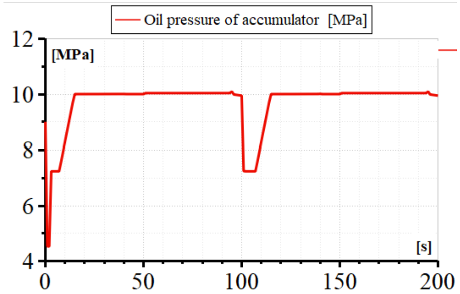


Fig. 7 Accumulator Oil Pressure

As can be seen from Fig. 7, the accumulator in the anti-slip system provides additional supplementary oil pressure for the oil cylinder in two stages (braking) of 0-15 and 100-115s, so that the piston rod of the oil cylinder can start quickly. According to the schematic diagram of TFY-YH anti-slip device system, six oil cylinders are arranged on both sides of the main oil circuit of hydraulic system, and the arrangement interval is far away, so the piston rod of the oil cylinder far away from the main oil circuit moves in place for a long time.

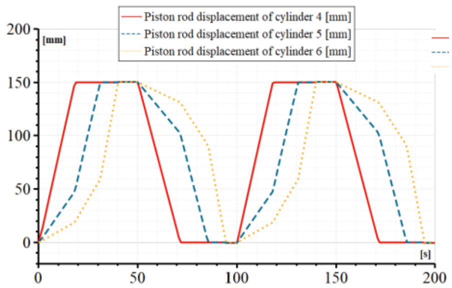


Fig. 8 Displacement of cylinder piston rod

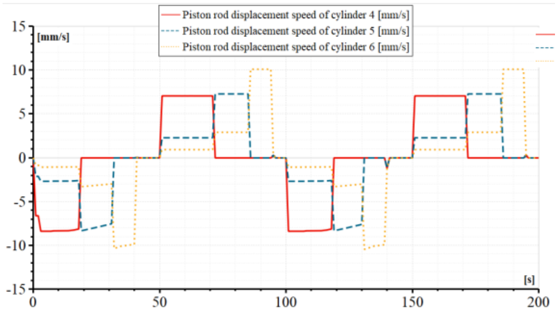


Fig. 9 Displacement velocity of cylinder piston rod

According to the simulation results of Fig. 8 and Fig. 9, it can be seen that when the anti-slip system brakes, the piston rod in place time of the oil cylinder 4 is 19.1 s, and the piston rod displacement speed of the oil cylinder 4 is divided into two stages: the piston rod displacement speed rises rapidly to 8.3 mm/s in the process of 0-19.1 s, and drops rapidly to 0mm/s in the process of 19.1 s-50s; The piston rod of cylinder 5 is in place for 31.9 s. The piston rod displacement speed of cylinder 5 can be divided into three stages: in the process of 0-17.8 s, the piston rod displacement speed climbs slowly to 2.6 mm/s, in the process of 17.8 s-31.9 s, the piston rod displacement speed rises rapidly to 8.3 mm/s and then decreases slowly to 7.53 mm/s, and in the process of 31.9-50s, the piston rod displacement speed drops rapidly to 0mm/s; The piston rod of cylinder 6 is in place for 41.1 s. The piston rod displacement speed of cylinder 6 can be divided into four stages: the piston rod displacement speed slowly climbs to 1mm/s in the process of 0-17.7 s, rises to 3.3 mm/s in the process of 17.7-30.8 s, then slowly decreases to 2.9 mm/s, rapidly rises to 10.3 mm/s in the process of 30.8-41.1 s, then slowly decreases to 9.8 mm/s, and rapidly decreases to 0mm/s in the process of 41.1-50s.

When the anti-slip system is relieved, the piston rod of cylinder 4 is in place for 22.3 s, and the displacement speed of piston rod of cylinder 4 can be divided into two stages: the displacement speed of piston rod rises rapidly to 7mm/s in the process of 50-72.3 s, and drops rapidly to 0mm/s in the process of 72.3-100s; The displacement velocity of the piston rod in cylinder 5 is divided into three stages: the displacement velocity of the piston rod rises to 2.2 mm/s in the process of 50-71.1 s, rapidly rises to 7.3 mm/s in the process of 71.1-85.7 s, and rapidly drops to 0mm/s in the process of 85.7-100s; The displacement velocity of cylinder 6 is divided into four stages: the displacement velocity of cylinder 6 rises to 0.8 mm/s in 50-69.8 s, 2.9 mm/s in 69.8-85.1 s, 10.1 mm/s in 85.1-94.8 s, and 0 mm/s in 94.8-100s.

Through the simulation of the dynamic characteristics of the anti-slip system, the time of the piston rod in place and the change of the displacement velocity of the piston rod during braking and relieving of the anti-slip system are obtained. According to the dynamic characteristics of each oil cylinder of the anti-slip system, the hydraulic control accuracy of a single oil cylinder can be improved, and the overall performance of the anti-slip system can be improved.

4.2 Research on Pressure Retaining Performance of Anti-slip System

In order to improve the pressure holding ability of anti-slip system, TFY-YH anti-slip system adopts feedback regulation mechanism to control the oil pressure of main oil circuit of motor-gear pump-anti-slip system in closed loop. An oil pressure sensor is arranged in the accumulator oil circuit to monitor the oil pressure of the main oil circuit of the anti-slip device system, and the signals collected by the sensor are input to the trigger module. The upper limit and lower limit of the trigger module are set to be 10MPa and 8.5MPa, and the trigger is connected with the switch module. The high-level output value of the switch module is set to be 1430r/min and the low-level output value is 0 r/min, that is, when the main oil circuit pressure of the anti-slip device is less than 8.5 MPa, the motor starts and drives the gear pump to replenish oil for the main oil circuit of the anti-slip device system to maintain the set oil pressure (10MPa). Set the simulation time to 200s and the time interval to 0.1 s.

Under normal circumstances, the leakage amount of oil cylinders is small and the simulation time is long. In order to reduce the calculation force and verify the automatic pressure compensation ability of the system under a short simulation time, the leakage coefficient of any oil cylinder in the anti-slip system is increased and set to 5×10^{-5} L/min/bar. The leakage amount of normal oil cylinder and the leakage amount of abnormal (increased leakage coefficient) oil cylinder are shown in Figure 10 below:

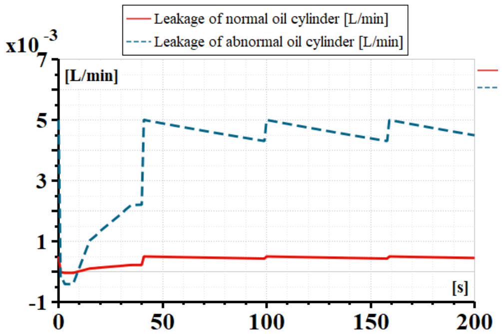


Fig. 10 Comparison of normal leakage cylinder with increased leakage cylinder

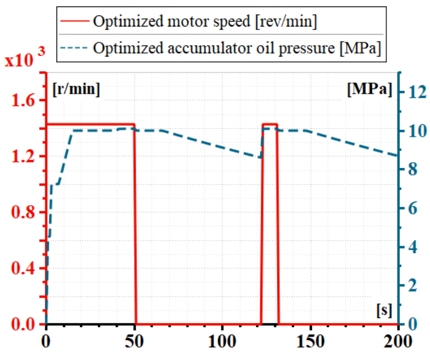


Fig. 11 Curve of oil pressure of accumulator and motor speed in anti-slip system

According to figs. 10 and 11, the anti-slip system starts braking from 0s, and the oil flows into the accumulator and oil cylinder. At 40.1 s, the anti-slip system brakes in place. At this time, the accumulator pressure is stable at 10MPa, the motor stops running, and the anti-slip system enters the braking and pressure maintaining state; Due to the leakage of the oil cylinder, after the motor stops running, the oil pressure of the accumulator drops from 10MPa to below 8.5 MPa in the process of 41.3-97.8 s, and the motor starts automatically in 97.8 s to replenish the oil pressure for the accumulator. In the case of obvious oil cylinder leakage, the motor automatically starts twice within 200s to replenish the oil pressure for the accumulator to maintain the pressure of the main oil circuit of the anti-slip system.

The simulation shows that the anti-slip system can make up the pressure of the main oil circuit in time during the braking and pressure maintaining process, so as to ensure the stop and anti-slip effect of the anti-slip system.

Because the oil cylinder of the anti-slip system frequently extends and retracts in the working process, there may be great friction loss between the cylinder block and the piston rod in the oil cylinder, resulting in internal leakage or external leakage of the oil cylinder. The increase of oil cylinder leakage will lead to frequent start of the motor and gear pump, which will easily lead to heating of the motor and gear pump, and then affect the stability of pressure keeping of the anti-slip system.

Therefore, according to the above-mentioned simulation model under the condition of "oil cylinder with increased leakage coefficient", this paper explores how to slow down the leakage speed of the main oil circuit of the anti-slip system and reduce the starting times of the motor. After many simulation experiments, an accumulator is added to the main oil circuit of the anti-slip system (hereinafter referred to as the main oil circuit accumulator), as shown in Figure 12 below. The nominal pressure of the main oil circuit accumulator is set to be 10MPa and the nominal volume is 10L. The simulation results of accumulator oil pressure change and motor starting after adding the main oil circuit accumulator are shown in Figure 13 below.

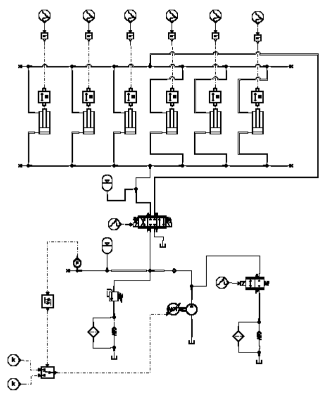


Fig. 12. Position of accumulator in main oil circuit of anti-slip system

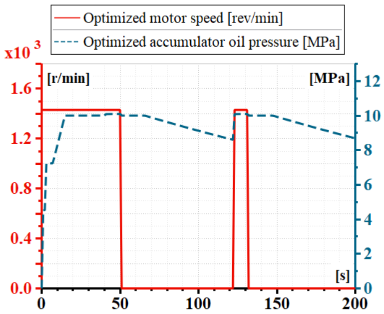


Fig. 13. Curve of oil pressure and motor speed after optimization of accumulator

According to Fig. 13, after the motor stops running, the accumulator drops from 10MPa to below 8.5 MPa in the process of 41.3-122.1 s, and the motor starts automatically in 122.1 s to replenish oil pressure for the accumulator. Under the condition of the same oil cylinder leakage, the leakage speed of the main oil passage of the anti-slip system is reduced by 24.3 s after adding the main oil passage accumulator, compared with that without adding the main oil passage accumulator, and the motor only needs to be started once within 200s.

Through the simulation study of the anti-slip system pressure maintaining performance, the leakage of the oil cylinder, the leakage of the main oil circuit pressure and the starting pressure maintaining of the motor are obtained. According to the simulation data, the hydraulic control accuracy of the anti-slip system can be improved to reduce the loss of the hydraulic system; In addition, through the above simulation research, under the same leakage condition, adding accumulators in the main oil circuit can effectively slow down the pressure leakage of the main oil circuit, reduce the number of motor start-up pressure compensation, reduce the heating of the motor and gear pump due to frequent start-up, and then prolong the service life of the motor and gear pump.

4.3 Influence of different parking areas on braking time and relief time of anti-slip system

Set the oil cylinder sequence from right to left as No.1-No.6 oil cylinder, and the oil cylinder arrangement spacing of the anti-slip system is approximately equal to the train wheelset spacing, and the train traveling direction is from right to left, so as to divide the train stop position into 7 areas, as shown in Figure 14 below:

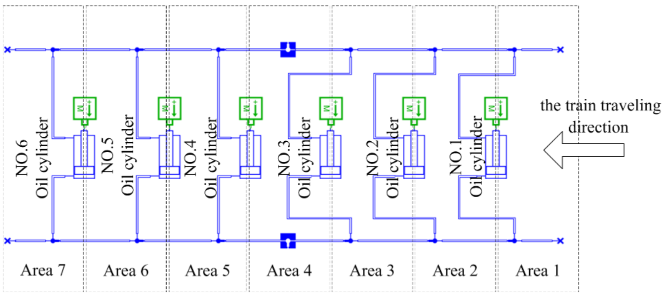


Fig. 14 Cylinder label and train traveling direction

Set the loads of different oil cylinders respectively to simulate the situation that the train stops in different areas. The stop position of the train and the corresponding load oil cylinders are shown in the following table. Set the simulation time to 100s and the time interval to 0.01 s.

Table 2. Stop position of train wheelset and corresponding load cylinder.

Stop area of train	Stop position of train front wheel	Stop position of rear wheel of train	Load cylinder
Area 1	1	-	1
Area 2	2	1	1、2
Area 3	3	2	2、3
Area 4	4	3	3、4
Area 5	5	4	4、5
Area 6	6	5	5、6
Area 7	-	6	6

The train stops in Zone 1, that is, the front wheels of the train stop at Cylinder 1 of the anti-slip system:

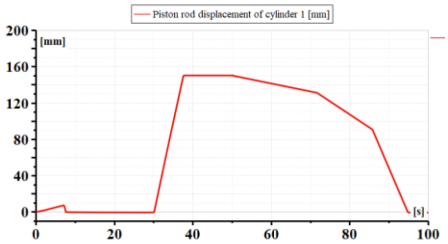


Fig. 15 Piston Rod Displacement of Cylinder 1

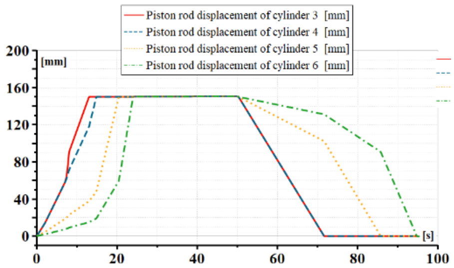


Fig. 16 Piston Rod Displacement of Other Cylinders

As can be seen from Fig. 15, when the front wheel of the train stops at the anti-slip oil cylinder 1 and the anti-slip system brakes, the piston rod of the oil cylinder 1 is in place for 37.69 s; When the anti-slip system is relieved, the piston rod of cylinder 1 is in place for 45.15 s; As can be seen from Fig. 16, the piston rod in place time of other oil cylinders is shown in the following table:

Table 3. Piston rod in place time of other oil cylinders during braking and relieving.

Other oil cylinders	Piston rod in place time (s) during braking	Piston rod in place time (s) during relief
Cylinder 2	24.55	21.68
Cylinder 3	17.05	21.68
Cylinder 4	17.47	35.77
Cylinder 5	26.51	35.77
Cylinder 6	29.97	45.32

Train stops in Zone 2, that is, front wheels stop at Anti-slip System Cylinder 2 and rear wheels stop at Anti-slip System Cylinder 1:

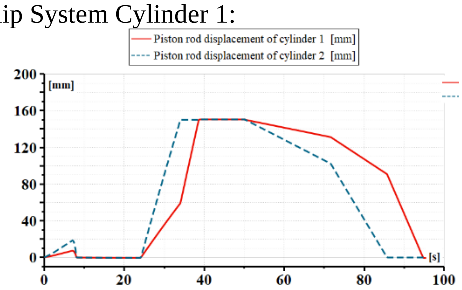


Fig. 17 Piston Rod Displacement of Cylinder 1 and Cylinder 2

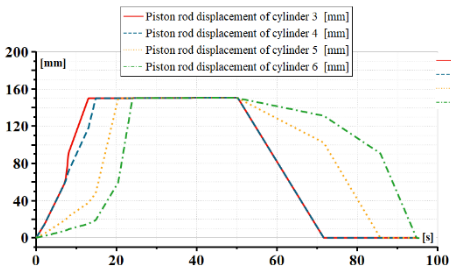


Fig. 18 Piston Rod Displacement of Other Cylinders

As can be seen from Fig. 17, the front wheel of the train stops at the anti-slip system oil cylinder 2, and the rear wheel stops at the anti-slip system oil cylinder 1. When the anti-slip system brakes, the piston rod in place time of the oil cylinder 1 is 38.63 s, and the piston rod in place time of the oil cylinder 2 is 34.19 s; When the anti-slip system is relieved, the piston rod in place time of cylinder 1 is 44.81 s, and the piston rod in place time of cylinder 2 is 35.77 s; As can be seen from Fig. 18, the piston rod in place time of other oil cylinders is shown in the following table:

Table 4. Piston rod in place time of other oil cylinders during braking and relieving.

Other oil cylinders	Piston rod in place time (s) during braking	Piston rod in place time (s) during relief
Cylinder 3	13.08	21.73
Cylinder 4	14.88	21.73
Cylinder 5	20.51	35.78
Cylinder 6	24.03	45.07

Train stops in Zone 3, that is, front wheels stop at anti-slip system cylinder 3 and rear wheels stop at anti-slip device cylinder 2:

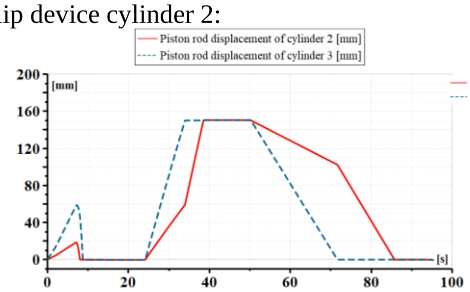


Fig. 19 Piston Rod Displacement of Cylinder 2 and Cylinder 3

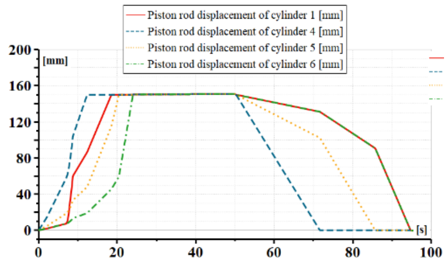


Fig. 20 Piston Rod Displacement of Other Cylinders

As can be seen from Fig. 19, the front wheels of the train stop at the anti-slip system oil cylinder 3, and the rear wheels stop at the anti-slip system oil cylinder 2. When the anti-slip system brakes, the piston rod in place time of the oil cylinder 2 is 38.59 s, and the piston rod in place time of the oil cylinder 3 is 34.03 s; When the anti-slip system is relieved, the piston rod in place time of cylinder 2 is 35.78 s, and the piston rod in place time of cylinder 3 is 21.71 s; As can be seen from Fig. 20, the piston rod in place time of other oil cylinders is shown in the following table:

Table 5. Piston rod in place time of other oil cylinders during braking and relieving.

Other oil cylinders	Piston rod in place time (s) during braking	Piston rod in place time (s) during relief
Cylinder 1	18.82	44.82
Cylinder 4	12.42	21.81
Cylinder 5	20.48	35.97

Cylinder 6	24.03	44.82

Train stops in Zone 4, that is, front wheels stop at anti-slip system cylinder 4 and rear wheels stop at anti-slip device cylinder 3:

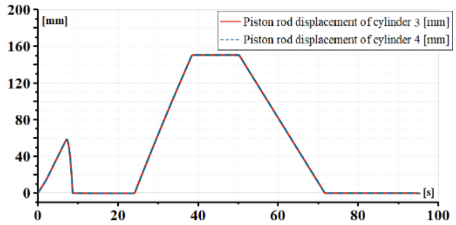


Fig. 21 Piston Rod Displacement of Cylinder 3 and Cylinder 4

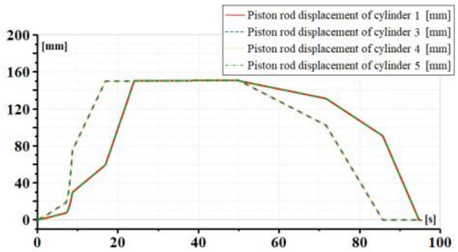


Fig. 22 Piston Rod Displacement of Other Cylinders

As can be seen from Fig. 21, the front wheels of the train stop at the anti-slip system oil cylinder 4, and the rear wheels stop at the anti-slip system oil cylinder 3. When the anti-slip system brakes, the piston rods of the oil cylinders 3 and 4 are in place for 38.49 s; When the anti-slip system is relieved, the piston rod in place time of cylinder 3 and cylinder 4 is 21.63 s; As can be seen from Fig. 22, the piston rod in place time of other oil cylinders is shown in the following table:

Table 6. Piston rod in place time of other oil cylinders during braking and relieving.

Other oil cylinders	Piston rod in place time (s) during braking	Piston rod in place time (s) during relief
Cylinder 1	24.09	44.91
Cylinder 2	16.92	35.78
Cylinder 5	16.92	35.78
Cylinder 6	24.09	44.91

Because the oil cylinders of the anti-slip system are symmetrically arranged, and the main brake oil circuit and the main relief oil circuit of the anti-slip system are arranged in the central symmetrical position, when the train stops in Zone 5, Zone 6 and Zone 7, the piston rod in place time of the corresponding oil cylinders is similar to that of other oil cylinders in Zone 3, Zone 2 and Zone 1, respectively, but the output oil cylinders are different.

According to the above train stop situation, the longest time of piston rod in place during braking and the longest time of piston rod in place during relieving are respectively taken as the braking time and relieving time of anti-slip system, and then the relationship between parking area and braking and relieving time of anti-slip system is obtained, as shown in Figure 23 below.

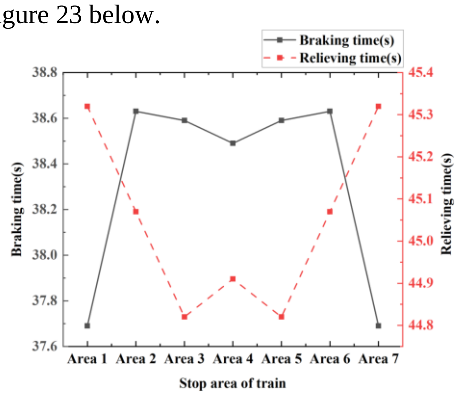


Fig. 23 Relationship between parking area and braking and relief time of anti-slip system

It can be seen from the figure that the train stops in Zone 1 and Zone 7 for the shortest braking time (37.69 s), and stops in Zone 2 and Zone 6 for the longest braking time (38.63 s). The longest mitigation time (45.32 s) is in Zone 1 and Zone 7, and the shortest mitigation time (44.82 s) is in Zone 3 and Zone 5. Because different stop positions of the train correspond to different brake and relief time of the anti-slip system, according to the relationship diagram between parking area and brake and relief time of the anti-slip system, under the condition of unknown train position, the stop position of the train can be determined by collecting brake and relief time of the anti-slip system and other oil cylinder time, which provides a new scheme for online monitoring of the anti-slip system.

5 Conclusion

This paper takes the TFY-YH anti-slip system as the research object, constructs the finite element simulation model of the TFY-YH anti-slip system, analyzes the dynamic characteristics of the anti-slip system, the pressure holding ability of the system and the influence of different stop positions of the train on the anti-slip system, which lays a theoretical foundation for improving the accurate control and high efficiency and energy saving of the anti-slip system. The research conclusions are as follows:

(1) Through the simulation of the dynamic characteristics of the anti-slip system, the time of the piston rod in place and the change of the displacement speed of the piston rod during the braking and relieving of the anti-slip system are obtained. According to the dynamic characteristics of each cylinder of the anti-slip system, the hydraulic pressure of a single cylinder can be adjusted and controlled to improve the control accuracy and overall performance of the anti-slip system.

(2) Through the simulation analysis of the anti-slip system, the leakage of the oil cylinder of the anti-slip system is obtained, and the leakage of the oil cylinder will lead to the pressure drop of the main oil circuit of the anti-slip system. The automatic pressure keeping of the anti-slip system during braking is realized by the closed-loop control method of "motor-gear pump-main oil circuit oil pressure of the anti-slip system"; By adding the accumulator in the main oil circuit, the problem that the pressure in the main oil circuit of the anti-slip system drops too fast can be effectively alleviated, and then the problem that the motor and gear pump heat up due to frequent starting can be reduced, and the service life of the motor and gear pump in the anti-slip system can be prolonged.

(3) When the train stops at different positions of the anti-slip system, the braking and relieving time of the anti-slip system is different and has certain rules. According to the braking and relieving time of each oil cylinder in the anti-slip system, the stopping area of the train can be judged, which improves the monitoring ability of the anti-slip system to a certain extent, provides a new method for the positioning of the train and provides a certain theoretical support for the automation and intelligence of the anti-slip system.

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