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Research on simulation and test of braking performance of ultra-deep mine hoist

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Abstract. With the utilization of the deep resources of the earth, it has become a new trend to develop a ultra-deep mine hoist which can meet the conditions of long distance, high speed and heavy load. The research on its braking performance is directly related to the safe operation. This paper is based on a ultra-deep mine hoist simulation test platform. The simulation model of the braking system is established in AMESim software by analyzing the hydraulic working principle. Finally, the field running data of the ultra-deep mine hoist simulation test platform are used to test and verify. The results of simulation and test show that some parameters' changing rules and impacts such as the greater the spring preloading, the shorter the depressurization time and the response time in the case of no more than a certain value (8.5MPa in this paper). It provides guidance for the research and application of ultra-deep mine hoist brake system in the future.

1. Introduction

The earth's shallow resources have been gradually exhausted with continuous exploitation and the quantity and quality of them have been gradually reduced, so that the exploitation of underground mineral resources continues to expand deeper. According to statistics, next ten to thirty years, a large number of coal will be exploited at a depth of more than one kilometer in the underground [1].In china, the depth of the mine has been excavated on average to 1200m, and the maximum depth will be 1800m.It is expected that the demand for ultra-deep lifting equipment of China's metal and coal mines will exceed sixty sets, and the total market value is estimated to be over 2 billion dollars within ten years [2,3].

The mine hoist is the most important mechanical equipment in the mining system. It is developing from the traditional mine hoist to the ultra-deep mine hoist. The brake system is the top priority for the whole ultra-deep mine hoist system, and it is the most important and the last safety barrier in the operation. The hoist often need service braking under normal conditions and safety braking in the event of an emergency, otherwise, it will cause a major accident, influencing the smooth production and the safety and life of miners [4].For this reason, the project group is developing large scale lifting equipment for ultra-deep mines which can meet the requirements of lifting height over 1500m, lifting speed over 18m/s, and terminal load over 240t. Compared with traditional mine hoists, these new features of long distance, high-speed and heavy load make the stable and safe operation of the braking system of the ultra-deep mine hoist face a severe challenge. In order to carry out the basic research on

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the safe operation of the ultra-deep mine hoist, the project group set up the simulation test platform for the ultra-deep mine hoist in the State Key Laboratory of Mining Heavy Equipment in CITIC Heavy Industries Co, LTD, Luoyang, China. According to the similarity principle of the system and the requirements of the ultra-deep mine hoist, the simulation test platform adopts multi-rope winding type with the derrick height is 47.95m, the depth of foundation pit is 10m, and the winding drum diameter is 800mm.The whole system mainly includes derrick, main shaft device, reducer, brake system and so on. Among them, the brake system is composed of a disc brake and a hydraulic station.

In this paper, the research on simulation and test of braking performance of ultra-deep mine hoist have been completed. Through analyzing the principle and process of braking of the hoist brake system, modeling in AMESim, making the field experiment on the simulation test platform, it is found that the effect of different brake clearance and different first-stage oil pressure on braking performance by combining simulation and experimental results. It provides guidance for the research of maintenance, overhaul, fault diagnosis and crisis prevention for the hoist brake system in ultra-deep mine, which has the practical significance, academic and engineering value.

2. The brake system of the mine hoist

The disc brake is the actuator of the brake system, which produces the brake torque. Hydraulic station, the operation control system, provides oil pressure. The combination of the two can achieve the service braking, safety braking function.

The simplified hydraulic principle of the simulation test platform is shown in Fig. 1 The electro-hydraulic proportional relief valve is used as the core component in order to adjust the pressure. The power is supplied by the UPS, that is, the system can still be braked in the power failure state.

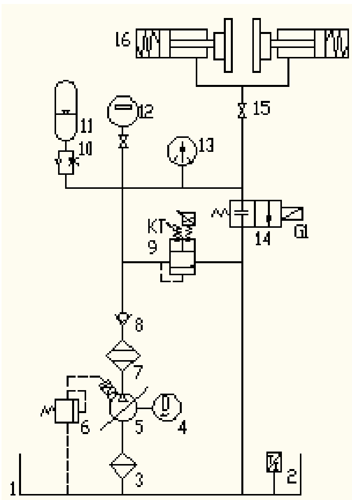

Figure 1. The simplified hydraulic principle of the brake system.

1-hydraulic tank 2-thermometer 3-oil filter 4-motor 5-plunger pump 6-pressure regulating valve 7-oil filter 8-one-way valve 9-electro-hydraulic proportional relief valve 10-one-way throttle valve 11-accumulator 12-pressure transmitter 13-pressure gauge 14-electromagnetic directional valve 15-ball valve 16-disc brake cylinder

2.1. service braking

The electro-hydraulic proportional relief valve controls the system pressure to achieve the purpose of adjusting the braking torque of the disc brake continuously. When the electro-hydraulic proportional relief valve 9 is on and the 14 is off, hydraulic oil goes from the 15 into the disc brake, the brake is slack state. When the 9 is off and the 14 is off, hydraulic oil goes back the tank 1 through the 9, it achieves the service braking.

2.2. *safety braking*
If accidents occur, the motor 4 is cut off to stop the oil supply and the UPS functions. When the oil pressure reaches a certain value set by the valve 9, the hydraulic oil returns to the tank 1 through the 9 and at this time the system belongs to first-stage braking state. The valve 14 functions after a certain time delay, the hydraulic oil returns to the tank 1 with the system completely unloading and all the brakes functioning. It makes the mine hoist stop at second-stage braking state [5].

3. Simulation on braking performance
The above hydraulic principle is modeled and simulated in AMESim. The simulation parameters of the model are shown in Table 1, and the hydraulic model is shown in Fig.2.

Table 1. The simulation parameters of the model.

parameter	value	parameter	value
Positive pressure	25KN	Spring stiffness	3125N/mm
Maximum oil pressure	5.48MPa	Spring preloading	8mm
Piston area	62.6cm^2	Accumulator pressure	1.9MPa
Rod diameter	21mm	Thickness of pipe wall	2mm
Motor speed	910r/min	Pipe diameter	10mm
Dead zone of oil cavity	27cm^3		

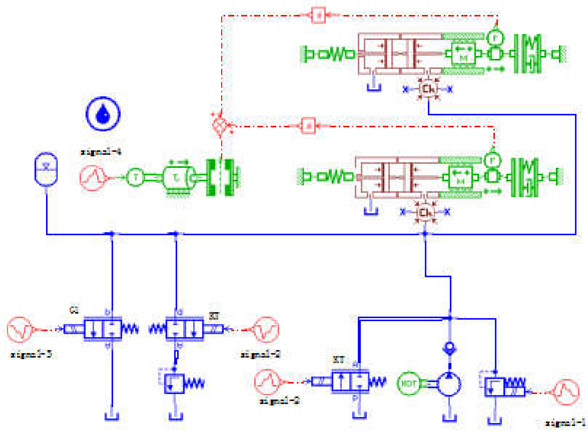

Figure 2. The hydraulic model in AMESim.

As the brake system works for a long time, the wear of the brake shoe increases, resulting in the increase of brake shoe clearance and the decrease of the spring preloading. The change of brake shoe clearance can show the change of spring preloading. When the spring preloading is changed to 7.5mm, 8mm, 8.5mm, 9mm, 10mm respectively while other parameters remain unchanged, it is gotten that the influence of the change of spring preloading on the braking performance. Some parameters in the simulation process are set including the maximum spring force 25KN, the relief valve's regulated oil pressure 2.45MPa, the simulation time 12s and the step length 0.001s.

As shown in Fig. 3 in the 8.2s-8.5s period of the braking process, by comparing the oil pressure curves of the spring preloadings of 7.5mm, 8mm and 8.5mm, it can be found that the response speed is faster with the increase of the spring preloading and the initial time to decrease significantly is 8.414s, 8.375s and 8.319s.This shows that the greater the spring preloading, the shorter the depressurization time and the response time in the braking process.

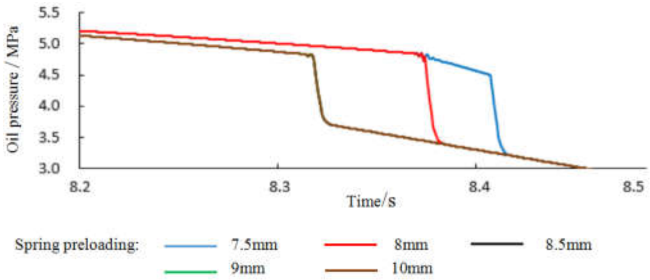

Figure 3. Oil pressure curves under different spring preloadings in simulation.

On the other hand, it can be seen from Fig.3 that the oil pressure fluctuates greatly during the depressurization process and the fluctuation value ΔP is 1.78MPa, 1.56MPa, 1.32MPa respectively with the increase of the spring preloading. So the greater the spring preloading, the smaller the oil pressure fluctuation value and the better the stability of the brake system. But the oil pressure curves of the spring preloading's of 8.5mm, 9mm and 10mm are almost overlapped, that is, when the spring preloading exceeds a certain value, the braking system always is in a state of brake so that the spring preloading has no effect on the oil pressure and the braking performance.

To obtain the influence on braking performance under different first-stage braking pressures in safety braking process, the electro-hydraulic proportional relief valve 9 is adjusted to 1.86MPa, 2.45MPa and 3.03MPa respectively. It can be seen from Fig.4 that curves of the oil pressure are almost steady near set value of first-stage braking pressures, namely the change of the first-stage braking pressure has little effect on the following performance. The oil pressure fluctuation value ΔP is 2.84MPa, 2.25MPa, 1.67MPa respectively with the increase of the first-stage braking pressure, so the greater the first-stage braking pressure, the smaller the oil pressure fluctuation value and the better the stability of the brake system. Similar to the above method, the influence on the brake shoe clearances under different first-stage braking pressures is analyzed and the result is that the brake shoe clearance curves are almost overlapped, that is, it has little effect on the brake synchronism as shown in Fig. 5

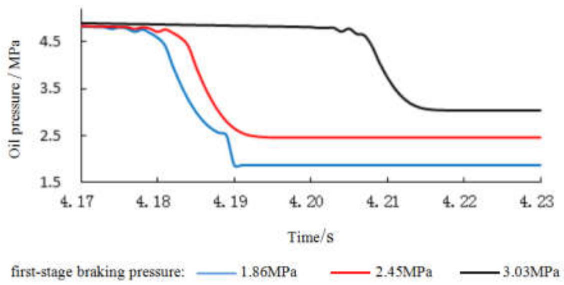

Figure 4. Oil pressure curves under different first-stage braking pressures in simulation.

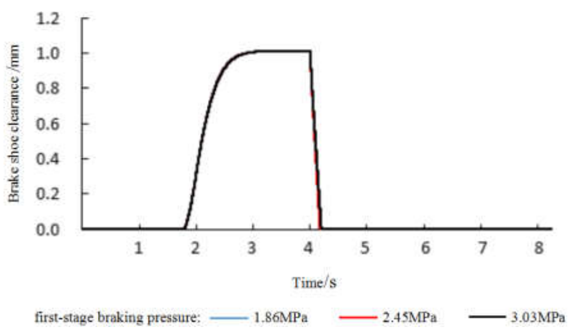

Figure 5. Brake shoe clearance curves under different first-stage braking pressures in simulation.

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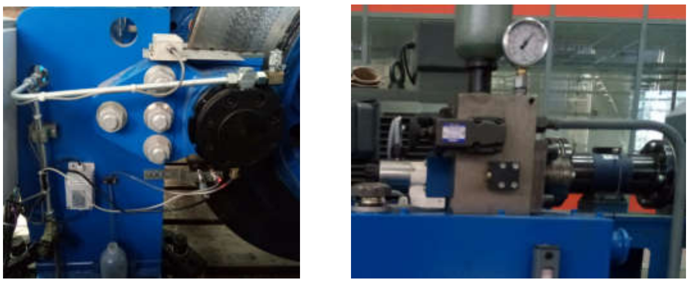
4. Experiment

In order to verify the reliability of the model and simulation of the brake system and the accuracy of the changing trend of the braking performance in this paper, several field experiments are carried out on the ultra-deep mine hoist simulation test platform set up by the project group. The field practical braking system is shown in Fig. 6



(a) disc brake (b) hydraulic station

Figure 6. The field practical braking system on the ultra-deep mine hoist simulation test platform.

The change of brake shoe clearance can show the change of spring preloading, so the clearance 0.5mm corresponds to the spring preloading 8.5mm,the clearance 1mm corresponds to the spring preloading 8mm,the clearance 1.5mm corresponds to the spring preloading 7.5mm.In the test, signal acquisition frequency is 100Hz and the influence of the change of the brake shoe clearance on the braking performance is obtained by manually adjusting the clearance of the brake shoe and keeping other parameters unchanged. As shown in Fig. 7 the depressurization time is longer with the increase of the brake shoe clearance. The greater the brake shoe clearance, the greater the oil pressure fluctuation value and the worse the stability of the brake system. These change rules coincide with the simulation results.

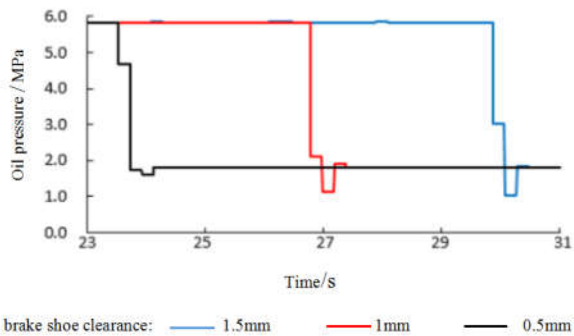


Figure 7. Oil pressure curves under different brake shoe clearances in test.

To test the safety braking under different first-stage braking pressures, the first-stage braking pressure is set to 1.86MPa,2.45MPa and 3.03MPa respectively by adjusting the electro-hydraulic proportional relief valve 9. The oil pressure curves drop to the corresponding set values and keep a certain holding time as shown in Fig.8, so the first-stage braking pressure has little effect on the following performance. Besides, the greater the first-stage braking pressure, the smaller the oil pressure fluctuation value and the better the stability of the brake system. It is similar to the simulation results.

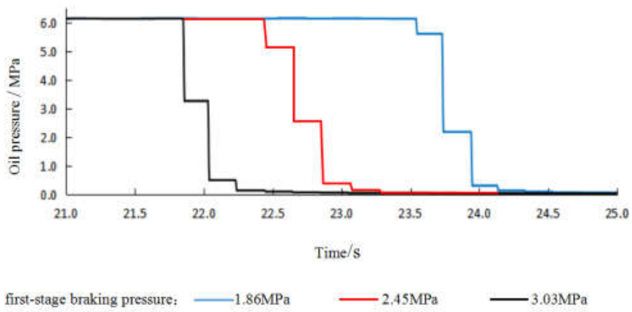


Figure 8. Oil pressure curves under different first-stage braking pressures in test.

5. Conclusion

The simulation and experimental results show that the response speed is faster with the increase of the spring preloading and the greater the spring preloading, the shorter the depressurization time and the response time in the braking process under normal conditions. But when the spring preloading exceeds a certain value (8.5MPa in this paper), the braking system always is in a state of brake so that the spring preloading has no effect on the oil pressure and the braking performance. The greater the first-stage braking pressure, the smaller the oil pressure fluctuation value and the better the stability of the brake system. But the change of the first-stage braking pressure has little effect on the following performance. The influence on the brake shoe clearances under different first-stage braking pressures is analyzed and the result is that the curves are almost overlapped, t hat is, it has little effect on the brake synchronism. These change rules can provide guidance for the research and application of ultra-deep mine hoist brake system in the future.

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