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Research of the cavitation in the rear brake chamber of hydraulic rock drill

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Abstract: The hydraulic rock drill is a type of hydraulic impactor that relies on hydraulic power to deliver periodic impact forces externally. As a core piece of equipment in mining operations, the service life of its key internal moving components and supporting seals directly determines the overall operational efficiency of the equipment. This paper focuses on the cavitation erosion issue affecting the rear guide sleeve, which is a primary support component in the rock drill. Utilizing a rock drill impact test bench, experiments were conducted. The results indicate that the negative pressure zone generated during the outward movement of the impact piston from the rear braking chamber is the main cause of cavitation near the rear guide sleeve. By comparing the motion of the impact piston within the braking chamber and the corresponding pressure variations under different clearance conditions, it was found that the formation of the negative pressure zone primarily depends on the penetration depth of the piston into the braking chamber and the maximum pressure within the chamber. Furthermore, by comparing the variations in the impact performance of the rock drill under different clearance conditions, it is proposed that the optimal clearance for this type of rock drill should be within the range of 0.04 to 0.06 mm.

Keywords: Hydraulic rock drill, Cavitation, Clearance

1 Introduction

Hydraulic rock drills serve as critical equipment in modern mining and tunnel excavation [1,2], as shown in Fig 1.1. The stability and continuity of a rock drill's operation directly determine the overall working efficiency of mining equipment. Therefore, the service life of its core internal components is critical, primarily including the impact piston, guide sleeve, and sealing elements.

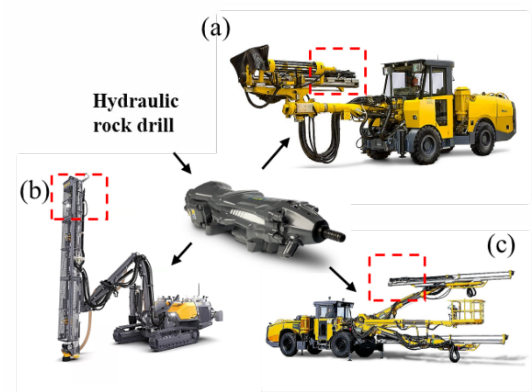

Fig 1.1 Mining machinery equipped with hydraulic rock drill

(a) Underground drilling rig, (b) Surface drilling rig, (c) Drill jumbo

Scholars have conducted in-depth research on the motion of the impact piston through mechanism analysis and hydraulic simulation techniques. Gorodilov [3] proposed a segmented modeling approach based on the assumption of constant inlet pressure and neglecting the impact and spool switching processes. To couple the motion of piston and spool, Hu [4] provided an AMESim-based model and based on this model Yang [5] added a plunger pump model to analyze the effect of inlet flow fluctuations on impact performance. Besides, Guo [6] employed the bond graph method to model the coupled motion, achieving lower complexity compared to AMESim.

Based on the motion analysis of the impact piston, the correlation between piston motion and internal cavitation has received some scholarly attention. For instance, Tian [7] and Jia [8] through experimental analysis, established a relationship between the service life of the front water seal and the impact piston's stroke in a rock drill, finding that seal life decreases as the stroke increases. Separately, Geng [9,10] in a study on the cavitation mechanism in a dual-cushion rock drill, demonstrated that the piston's kinematic parameters upon entering the front cushioning chamber are a critical factor in the formation of cavitation in the front chamber.

While existing research has extensively addressed the front guide sleeve and seals, the rear guide sleeve has received limited attention. To bridge this gap, this study is dedicated to investigating the rear-side problem. By integrating laser displacement and pressure sensors, we capture the key parameters of piston motion and pressure dynamics at the rear cushioning chamber to elucidate the mechanism of cavitation erosion and develop a targeted solution.

2 Problem statement

2.1 Structure and working principle

Dual-control hydraulic rock drills (DHRD) have become the primary type of rock drill equipment in current mining machinery due to their high energy efficiency. Their basic structural composition is illustrated in Fig 2.1. The main body of DHRD contains impact piston (1), guide sleeve (2), and cylinder sleeve (3). It is used to perform the shock motion. The motion is controlled by the spool of direction valve (5), installed on the valve seat (4). There are four chambers formed by part 1 and 3, i.e., front chamber (FC), piston signal chamber (PSC), tank chamber (TC), and rear chamber (RC). Also, there is a signal chamber for direction valve (VSC). The pressure oil is input to power the whole system from the inlet port of direction valve.

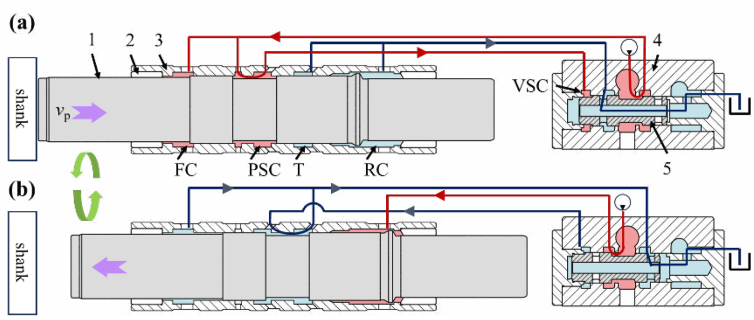

Fig 2.1 Working principle of DHRD

(a) Return stroke state, (b) Impact stroke state

The impact motion of DHRD contains two typical states, i.e., return stroke state (RSS) and impact stroke state (ISS), and the switching of these states above are shown in Fig 2.2. The working principle can be explained as follows: DHRD switches between RSS and ISS repeatedly by the coupling motion of part 1 and 5. The operating state during RSS and ISS can further be divided into acceleration and braking states according to the direction of movement of part 1.

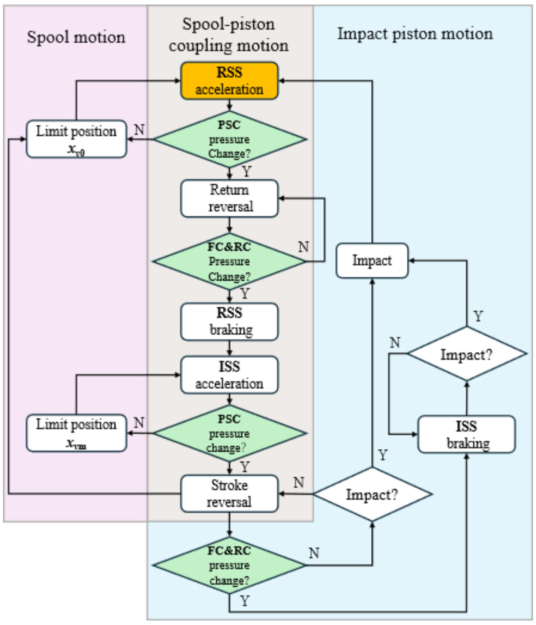

Fig 2.2 Flowchart of working principle of DHRD

2.2 Cavitation in the rear braking chamber

As illustrated in Fig 2.3, when the impact piston moves past a specific position in the cylinder, a sealed oil chamber forms between the piston, the cylinder, and the guide sleeve, and this chamber is the rear braking chamber. The chamber serves as an anti-impact mechanism to protect the piston from striking the rear guide sleeve in the period between RSS braking and ISS acceleration.

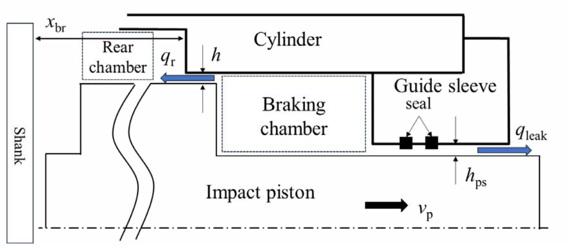

Fig 2.3 Structure of braking chamber

Damage progression in the rear guide sleeves of the same rock drill model reveals a clear time-dependent progression, as shown in Fig 2.4.

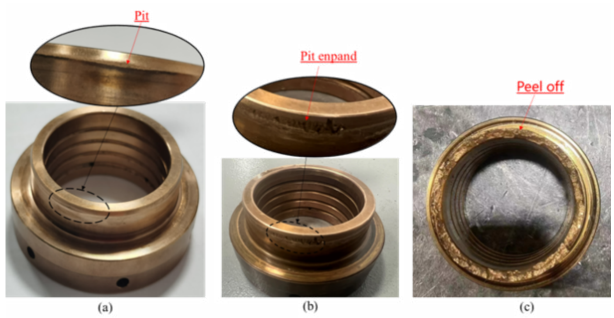

Fig 2.4 Cavitation phenomenon in different stages

(a) Early stage (less than 50 impact hours) (b) Intermediate stage (between 50 and 100 impact hours) (c) Late stage (over 100 impact hours)

Damage to the rear guide sleeve leads to hydraulic oil contamination. The resulting metallic debris poses a significant threat to both the rear seal and the piston surface, substantially reducing the service life of these components. Hence, determining how to optimize the rear braking chamber's structure to mitigate cavitation-induced damage to the rear guide sleeve is a critical issue for improving the service life of rock drills.

3 Method and materials

3.1 Hydraulic rock drill test bench

As shown in Fig 3.1, a hydraulic rock drill impact test bench (HRDITB) was established to acquire the pressure within the rear braking chamber and the motion of the impact piston.

The HRDITB is equipped with (A) an impacted system serving as the load, (B) pressure sensors mounted on the rock drill to record pressure variations in the rear braking chamber, and (C) a laser displacement sensor to track piston movement.

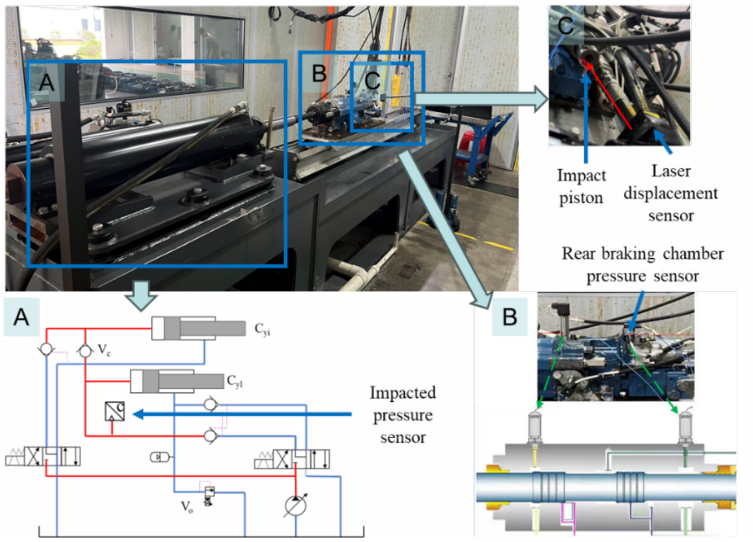

Fig 3.1 Hydraulic rock drill impact test bench

The load from the impacted system is generated by a hydraulic circuit where the impact cylinder (Cyi) is connected to a loading cylinder (Cyl). The pressure in the rodless chamber of Cyl is controlled by an overflow valve (Vo), thereby establishing a stable pressure differential across the intervening check valve. This differential pressure is the definitive load provided by the system. The sensors parameters are shown in Table. 3.1.

Sensor	Model	Range	Accuracy	Sample Rate
Laser displacement sensor	TS803-140/100	140~240 mm	± 0.05% FS	8192Hz
Pressure sensor	YN-63-IV-400	0~400 MPa	± 0.5% FS	8192Hz

Table. 3.1 Sensors used in the HRDITB

3.2 Motion and pressure variation characteristics of the impact piston in the rear braking chamber

Based on the mechanical structure, the position at which the impact piston enters the rear braking chamber is identified as x_{bl} = 58.5 mm. As shown in, the laser displacement data confirm that the piston travels beyond this point. A calculation of the piston's velocity reveals that it decelerates abruptly within the braking chamber. Thus, the braking phase is defined by the piston’s velocity.

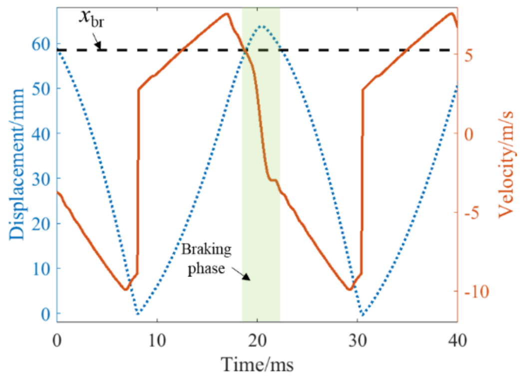

Fig 3.2 Braking phase in the motion of the impact piston

Based on Fig 3.2, the motion of the impact piston within the braking chamber (the braking phase) can be defined. To further analyze the relationship between piston motion and pressure changes during this process, the braking phase is subdivided according to the relative pressure relationship between the braking chamber and the rear chamber. The correlation between the piston's penetration distance into the braking chamber (x_{pb}), as well as the piston velocity, with the chamber pressure are separately shown in Fig 3.3.

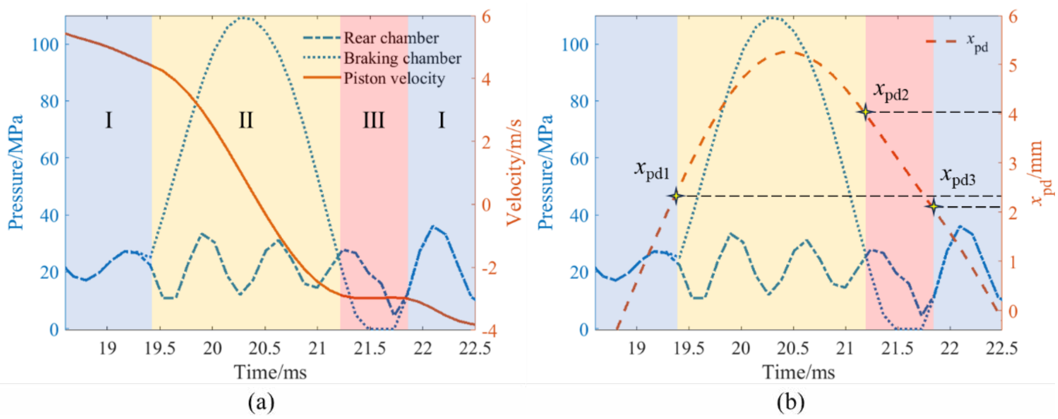

Fig 3.3 Motion and pressure variation in the rear braking chamber

As shown in Fig 3.3, the braking phase can be further divided into three states:

In part I, the braking chamber pressure equals the rear chamber pressure, the piston has entered the rear braking chamber less than 2 mm. During this state, the braking chamber does not form a sealed cavity. For most of the time, the pressure in the braking chamber equals that in the rear chamber. However, as the impact piston approaches the transition point x_{pd1} or x_{pd3}, a minor pressure difference arises between the braking chamber and the rear chamber due to localized throttling effects.

Then in part II, the braking chamber pressure exceeds the rear chamber pressure, the piston has entered the rear braking chamber between 2~5.3 mm. During this phase, the braking chamber is nearly fully sealed and functions approximately as a hydraulic spring on the impact piston, resulting in a linear change in piston velocity.

Part III occurs only when the impact piston exits the braking chamber, typically at a position about 4 mm within it. At this point, the braking chamber remains sealed, and the lack of fluid replenishment causes a sharp drop of the pressure in braking chamber as the piston moves. Consequently, the braking chamber no longer provides the outward thrust on the piston. Since the resistance from the front chamber pressure is relatively low, the piston exhibits a state of uniform motion during this phase.

Vapor cavitation is defined as the phase change resulting from a rapid drop in local pressure below the fluid's saturated vapor pressure. Hence the rapid pressure-drop in part II and the negative pressure in part III are the core elements of the appearance of the cavitation. Therefore, this study employs an experimental approach to investigate the influence of the clearance size on the pressure dynamics within the braking chamber, aiming to identify the optimal clearance value for mitigating cavitation in the current design.

4 Result analysis and discussion

Based on the analysis of the impact piston's motion within the braking chamber and the corresponding pressure variation in the chamber, it can be concluded that cavitation erosion on the rear guide sleeve is associated with the local negative pressure generated during the part III of the piston braking phase, as well as the piston's movement. During this state, the sealing length formed between the piston and the cylinder is insufficient to fully isolate the braking chamber. As a result, high-pressure oil from the rear chamber flows into the braking chamber through this restricted passage via throttling. This flow, combined with the pressure drop induced by the piston's motion, ultimately leads to the occurrence of local cavitation.

To investigate the influence of different clearances on the pressure variation in the braking chamber and the motion of the piston within it, this study designed three sets of experiments with varying clearances for comparative analysis. In the experiment, the clearance between the braking chamber and the impact piston was controlled by adjusting the diameter tolerance of the cylinder's braking chamber. ISO VG 46 hydraulic oil was used in the tests, and the oil temperature was monitored via a temperature sensor installed in the tank. All experiments were conducted with an initial oil temperature of 60°C. Given the short duration of each test, it was assumed that the internal oil temperature of the rock drill remained consistent with the oil temperature in the tank.

The corresponding peak pressure (p_{brm}) in the braking chamber, the maximum penetration distance of the piston into the braking chamber (x_{pdm}), and time from piston entry into to exit from the braking chamber (t_{br}) for each clearance h are presented in Tab 4.1.

h /mm	p_{brm}/MPa	x_{brm}/mm	t_{br}/ms
0.02	114.1	1.6	1.7
0.04	120.5	5.25	3.8
0.083	108.6	5.17	3.3

Tab 4.1 Peak braking chamber pressure and critical impact piston motion parameters under different clearances

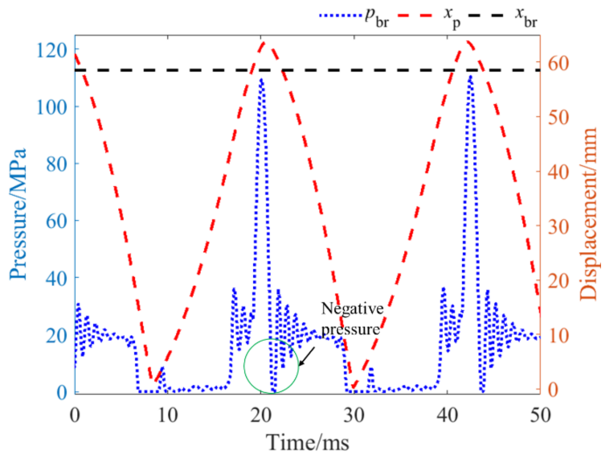

Fig 4.1 Clearance h = 0.083 mm, pressure and impact piston motion variations in the braking chamber

As shown in Fig 4.1, the original structural clearance h = 0.083 mm. Under this clearance, a high-pressure zone exceeding 100 MPa is formed in the braking chamber due to compression by the penetrating impact piston. Since the sealing length formed between the piston and the cylinder at this stage exceeds 5 mm, insufficient oil replenishment occurs in the braking chamber during the piston's outward movement, resulting in a negative pressure zone. This zone corresponds to the part III in Fig 3.3 and is the primary cause of cavitation in the braking chamber region.

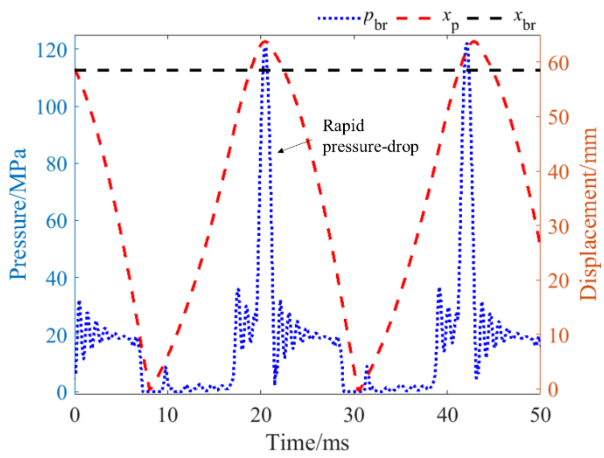

Fig 4.2 Clearance h = 0.04 mm, pressure and impact piston motion variations in the

braking chamber

As shown in Fig 4.2, the reduction in clearance accelerates the high-pressure buildup process in the

rear braking chamber, increase

ng the maximum peak pressure by approximately 12 MPa. However, the change in x_{brm} is minimal, so the pressure loss in the braking chamber caused by the outward motion of the impact piston remains essentially the same as when the h is 0.083 mm. Nonetheless, due to the increased p_{brm}, it does not enter a negative pressure state.

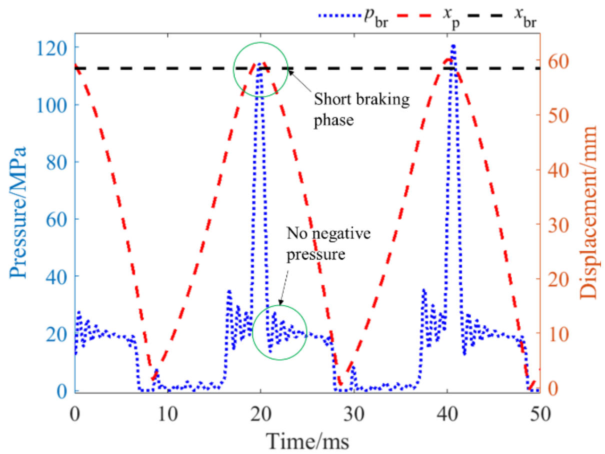

Fig 4.3 Clearance h = 0.02 mm, pressure and impact piston motion variations in the braking chamber

As shown in Fig 4.3, further reducing the clearance to 0.02 mm significantly decreases the sealing length required for the braking chamber to build high pressure. In this case, x_{brm} is only 1.6 mm, which is far smaller than the value under larger clearance conditions. Evidently, the increase in p_{brm} and the shortening of x_{brm} resulting from the reduced clearance jointly diminish the likelihood of the braking chamber entering a negative pressure range. Consequently, under this clearance condition, the rear braking chamber will be free from cavitation risk. However, as the tolerance for the guide sleeve is 0.015 mm, if the clearance h is set too close to this tolerance, it may impair the guiding function of the guide sleeve and significantly increase the probability of wear between the cylinder and the piston. Therefore, the actual selected clearance value should be greater than twice this tolerance.

Based on the above analysis, it can be determined that the ideal clearance value should fall within the range of 0.03 to 0.083 mm. To explore the optimal parameters within this range, the impact performance of each group of rock drills was compared, and the results are presented in Tab 4.2 and Fig 4.4.

h	Stroke/mm	Impact energy/J	Impact frequency/Hz	Impact power/kW
0.04	60.1	361	49.4	17.8
0.06	63.5	327	46.8	15.3
0.083	63.6	331	46.3	15.3

Tab 4.2 Hydraulic rock drill impact performance under different clearances

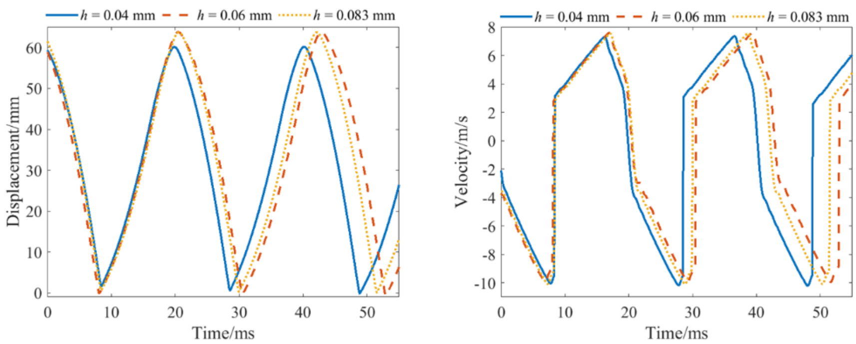

Fig 4.4 Hydraulic rock drill impact performance under different clearances

Based on Fig 4.1~Fig 4.4 and Tab 4.2, when the clearance value is between 0.06 and 0.08 mm, reducing the clearance only increases p_{brm} and lowers the risk of the braking chamber entering a negative pressure state during the piston's outward motion. However, when the clearance is reduced to 0.04 mm, this reduction significantly affects the sealing length required for the rear braking chamber to close, shortening x_{brm} and thereby reducing the total travel of the impact piston and increasing the impact frequency. At the same time, due to the reduction in energy loss within the braking chamber, the impact energy per piston stroke is also enhanced, leading to an overall improvement in the impact power of the rock drill.

5 Conclusion

This study, addressing the cavitation erosion issue in the rear braking chamber of a rock drill's cylinder, systematically investigated the influence of different clearance fits on the chamber's pressure dynamics in response to piston motion. The following principal conclusions were drawn:

(1) Cavitation in the braking chamber of a rock drill primarily occurs during the outward motion of the impact piston from the chamber. When the piston is approximately 4 mm away from exit the braking chamber, the braking chamber and the rear chamber gradually become connected. Due to the extremely low pressure in the braking chamber at this stage, the high-pressure oil from the rear chamber rushes into the braking chamber at high speed, leading to cavitation in the oil. This, in turn, causes vapor erosion on the adjacent guide sleeve.

(2) The clearance primarily influences the x_{brm} by affecting the rate of pressure buildup within the chamber. When the x_{brm} is less than 2 mm, the pressure drop in the braking chamber induced by the piston during the impact stroke is insufficient to create a negative pressure zone, thus preventing oil cavitation. However, if the clearance approaches the tolerance of the guide sleeve, it can impair the sleeve’s guiding function, leading to wear between the piston and the cylinder and reducing the piston's service life. Therefore, based on a comprehensive analysis of the experimental results, the optimal clearance for the model studied in this paper is approximately 0.04 mm.

6 Reference

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