Experimental Study on Braking Performance of an AT Integrated Hydraulic Retarder of HPT Series

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Abstract. In this paper, the hydraulic working principle of an AT integrated hydraulic retarder of HPT series is introduced, and the braking performance is tested on a bench. The braking characteristics under different solenoid valve currents and different retarder rotational speeds are obtained and analyzed. According to the test results, the optimization method of braking characteristics in high speed section is studied.

Keywords: Hydraulic retarder; Braking characteristics; Bench test.

1. Introduction
The hydraulic retarder produced by four famous hydraulic retarder companies, such as VOITH, ZF, Allison and SCANIA, has been widely used and has formed a series of products applicable to various vehicles. [1] with the introduction of national standards and the deepening of people's attention to traffic safety, hydraulic retarders are more and more widely used in heavy vehicles. [2]
The hydraulic retarder on the vehicle is generally divided into front and rear type according to the layout position. [3] The rear type is widely used because the braking power flow does not need to go through the gear mechanism, the operation is simple and the braking process will not bring load to the gear mechanism. According to the layout form, hydraulic retarders can be divided into independent and integrated type. It can improve the overall reliability of the system by integrating the hydraulic retarder with the transmission which merges the retarder structure and the hydraulic system into the transmission system. And through the optimizing design, the overall layout space can be reduced, then causes the reduction of the transmission system and vehicle weight. [4]
For a given hydraulic retarder, the maximum braking torques under different rotor speeds form the external braking characteristics of the hydraulic retarder. The curve of external braking characteristics of real vehicles can be roughly divided into three sections. The braking torque of low speed section is mainly determined by the speed of the retarder's driving wheel (the rotor), which is the maximum braking capacity of the retarder under the corresponding speed. The braking torque of middle speed section is limited to the maximum braking torque of the design value. The braking torque of high speed segment is limited by the maximum braking power of the retarder allowed by the system, [3] in which the matching of braking demand and heat dissipation of the vehicle shall be comprehensively considered.

2. The Hydraulic Principle Introduction of HPT Series Retarder
For the hydraulic retarder with given circulation circle and cascade system structure, the braking torque it can provide at each speed is mainly determined by the amount of liquid filled in its chamber. All hydraulic retarders have similar working characteristics to centrifugal pumps due to the centrifugal effect of oil caused by the rotation of the driving wheel. Therefore, by adjusting the inlet and outlet
boundary parameters of the retarder, it can indirectly adjust the volume of liquid in the chamber, so as to adjust the braking torque of the retarder within a certain range.

From the hydraulic boundary, hydraulic retarder control methods are generally divided into three types: inlet pressure (flow) control, outlet pressure control, oil replenishment control. [3] HPT series are domestic serialized AT products with independent intellectual property rights. They are equipped with optional hydraulic retarder modules. The installation of hydraulic retarder modules can be realized on the structure simply by replacing the rear cover. Its structural form is a typical integrated rear type, through the transmission housing interface oil channel, the retarder hydraulic circuit is merging into the transmission hydraulic system. This series of retarders adopt the control method of oil replenishment in circular outer ring. During the initial stage of braking, the oil supply to the circular chamber is provided by the main oil line of the transmission under the control of the control valve (it can also be incorporated into the accumulator for quick oil filling). The transmission oil is driven to circulate between the circular chamber of the retarder and the oil cooler (radiator) due to the differential pressure between inner and outer rings caused by centrifugal force. The oil quantity in the chamber is constantly regulated by the control valve to realize the constant control of braking torque of retarder. Its specific hydraulic principle is shown in the following figure.

![Figure 1. Hydraulic schematic diagram of HPT series AT integrated hydraulic retarder.](image)

The hydraulic system in the figure can be roughly divided into three parts: retarder (body), retarder main valve, retarder adjustment control valve. The retarder is located at the end of AT and blocks the oil line between the transmission and the oil cooler (radiator). When the retarder is not working, the main valve plug is in the upper position, and the radiator is used for heat dissipation of the transmission. When the retarder works, the main valve plug is in the lower position, the radiator is used for the retarder heat dissipation, the transmission is cooled by an auxiliary radiator shown in the figure.

The initial oil filling and braking torque control of the retarder is realized by the pressure and flow of main pressure oil and control oil of the transmission hydraulic system. The control valve of retarder adjustment includes proportional solenoid valve, switch valve and regulating valve. After the proportional solenoid valve is energized by corresponding current calculated by the TCU, the switch valve opens to lead the main pressure oil to the regulating valve and the retarder main valve respectively, to regulate the retarder filling through controlling the pressure of the retarder chamber out ring and cause the main valve movement for shifting the cooler circuit. In the working process of retarder, the oil circulates between the retarder body and the radiator to dissipate the heat by the cooler. The amount of liquid filling is indirectly regulated by the pressure of the outer ring of the chamber. The upper end of the regulating valve is the oil pressure of the proportional electromagnetic valve, and the lower end is the chamber pressure of the retarder. The pressure of the retarder chamber is adjusted by the amount of current.
3. Bench Arrangement and Test Process

Based on the hydraulic control principle of the hydraulic retarder, the purpose of this test is to test the braking torque of the retarder at different speeds and different solenoid currents by bench test, and then investigate its control law and braking performance. Since it is difficult to use the hydraulic station to simulate the oil connection between the retarder and the transmission, the retarder is normally installed on the transmission in this test, and the overall bench test is carried out. Table arrangement is shown in figure 2 below. The transmission ends are connected to 500kw motors respectively, and the oil port at the end of the retarder is connected to the oil cooler through the pipeline for heat dissipation of the system.

![Figure 2. Photos of the test bench.](image)

In the test, the input motor is set to speed mode, and the output motor is set to torque mode and specified as 0. Start the input motor to 800 RPM, change the transmission into the corresponding gear (test with retarder, the test in the low speed section adopts fifth gear, high-speed section using sixth gear). Then change the motor mode of the output end to speed, and set its value according to the transmission ratio and the speed of the input motor. Then the input motor mode is changed to torque and set to 0. After the motor mode is changed, the torque converter of the transmission is operated to make it locked. In this way, the output motor can drive the retarder and assume its braking power. Meanwhile, a small amount of power is transferred to the front oil pump through the gear mechanism to make the whole hydraulic system work normally. After stabilization, the braking performance test process of the retarder can be started.

At each speed point, first record the system idle torque $M_0$ when the current of the solenoid is 0, namely the torque when the retarder is not working. And then record the braking torque $M_2$ under different solenoid currents at each speed point, the difference between the two is the required braking torque $M$.

4. Test Data Processing and Analysis

According to the test data, the following braking torque characteristic curve is obtained. As can be seen from the curve in the figure, the basic trend of braking torque characteristic curve is similar at each speed point. As the current value of the solenoid valve (i.e., the oil-filled control pressure) goes from low to high, the growth rate of its braking torque basically shows a trend of increasing first and then decreasing. That is, the growth rate of braking torque in the low current section increases gradually, while that in the high current section decreases gradually. According to the braking characteristic law of the retarding device, the braking torque is proportional to the filling rate at the same speed, and the test results indirectly reflect the non-linear relationship between the filling pressure (that is, the current of the solenoid valve) and the filling rate.
Figure 3. Brake torque characteristic curves at each speed point.

It is clear from the curve cluster that with the increase of the speed, the braking characteristic curve moves to the right gradually, that is, the braking torque increases with increasing lag. This is due to the fact that the retarder adopts the control of oil replenishment by circulating circle outer ring. As the speed grows, the centrifugal pressure in the outer ring gradually increases, and oil filling becomes more and more "difficult". The growth rate of liquid filling gradually decreases with the increase of current.

The maximum braking torque of the initial design is about 2,700Nm, and the curve can be divided into three groups: the speed section below 1,200rpm, which cannot reach the torque value even under the maximum electric current. The curve end shows a gentle trend, and the speed of this section is the constraint factor of the braking characteristics. With the increase of the current in the middle speed section between 1200rpm~1600rpm, the braking torque can reach or even exceed the design maximum value, and the curve end still shows an obvious increasing trend. The braking torque cannot reach the maximum value of 2700Nm in the speed section above 1800rpm, and the liquid filling rate of this section is the constraint factor of its braking characteristics.

According to the above-mentioned external braking characteristics of the real vehicle, the curve is roughly divided into three sections. The low speed section is determined by the speed of the retarder, which is the maximum braking capacity of the retarder under the corresponding speed. The middle speed section is the control line of the maximum braking torque, and is the maximum braking torque of the retarder required by the system. The high speed segment is limited by the maximum braking power of the retarder allowed by the system. In this paper, the middle speed section of the retarder is limited by the maximum braking torque of 2700Nm, and the high speed section is limited by the maximum braking power of 440KW. According to the design requirements and the test data, the following characteristic curve is obtained.

Figure 4. Test and design brake external characteristic curve.
Because the braking torque or braking power can reach the maximum value of the design at each point of the mid-speed section, it has a good coincidence with the design curve. With the increase of rotational speed, oil filling becomes more and more "difficult", so the actual maximum braking power is lower than the designed maximum braking power. According to the braking characteristics of the retarder, the braking torque is proportional to the second power of the speed, so under the condition of full oil filling, it can completely meet the design maximum requirements in the high speed section. Combined with the hydraulic system and control principle of the retarder, the maximum oil filling pressure is limited by the control oil pressure and the size of control valve plug control end and feedback end, which can be modified to make the braking performance of each point in the high speed section reach the maximum design value.

5. Conclusion
In this paper, the hydraulic working principle of an AT integrated hydraulic retarder of HPT series is introduced and its braking performance is tested on a bench. According to the test data, the characteristic curve clusters of braking torque at each speed point are obtained. Combined with the general design requirements, the test data were processed to obtain the design and test braking characteristics, and give the solution to optimize the performance of the retarder at high speed section through the analysis of the hydraulic principle and of the difference between the test values and design values. Through the test and analysis, it is concluded that the retarder generally meets the requirements of the highest index, and the braking performance of the high speed section can be optimized and improved.

References
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