Design of TBM Cutterhead Jamming Breakout Test Bench Based on Hydro Viscous Clutch

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Abstract: This work proposes and designs a TBM cutterhead Jamming breakout test bench based on Hydro viscous clutch to address the engineering challenges of cutterhead jamming and difficulty in extricating in complex geological conditions of full section tunnel boring machines (TBMs). The study introduces hydro viscous clutch technology, which utilizes its controllable soft start characteristics to achieve flexible adjustment of cutterhead drive torque and transient high torque output function, significantly improving the system's ability to overcome jamming. On this basis, the overall design scheme of the experimental platform was elaborated in detail, with a focus on the selection and design of the mechanical structure, hydro viscous clutch, hydraulic system, and control system hardware. This study provides new driving ideas and experimental verification platforms for TBM cutterhead jamming breakout technology, which has important theoretical significance and engineering application value for improving the construction safety and excavation efficiency of TBM under harsh geological conditions.

Keywords: Tunnel Boring Machine; TBM Cutterhead; TBM Jamming; Jamming Breakout Technology; Hydro Viscous Clutch; Test Bench

1. Introduction

Tunnel Boring Machine (TBM), abbreviated as TBM, is a large-scale industrial tunnel construction equipment that integrates multidisciplinary technologies such as rock mechanics, mechatronics control, optical measurement, and tunnel engineering [1-2]. TBM cutterhead often experiences jamming accident due to insufficient driving torque under extreme geological conditions [3-7]. The starting torque required for a jammed cutterhead is often several times greater than the torque during normal excavation, and the cutterhead system (gearbox, planetary reducer) cannot withstand excessive impact loads during design. Therefore, TBM extricating poses a new challenge to existing cutterhead drive technology and is recognized as one of the industry's difficulties [8-11]. Due to the drawbacks of manual rescue methods in terms of economy, safety, efficiency, etc., TBM usually adopts the method of adding additional installed power at the beginning of design to prevent jamming accidents, and the energy utilization rate urgently needs to be further improved [12-13].

In order to solve the contradiction between instantaneous high torque output and low impact during the jam breakout process of TBM, our research team has introduced hydro viscous transmission technology into the TBM cutterhead drive system. Through the coordinated control of hydro viscous clutch, a double load torque output capacity with a duration of up to 70 seconds has been achieved, effectively meeting the practical engineering jam breakout needs of TBM [14-16], which has important engineering application value. This work will provide a detailed analysis and design of the TBM Cutterhead Jamming breakout Test Rig Based on Hydro Viscous Clutch.

2. Analysis of the Principle of TBM Cutterhead Jamming Breakout

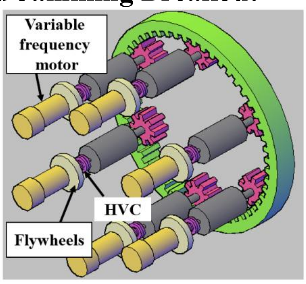

Figure 1. New Drive Scheme of TBM Cutterhead

After comprehensive analysis of several existing TBM cutterhead drive schemes, another design scheme was proposed [17-18]. As shown in Figure 1, the new TBM cutterhead drive system is mainly composed of variable frequency motor, inertia flywheel, hydro viscous clutch, speed reducer, planetary gear reducer. The power transmission route is: variable frequency motor-hydro viscous clutch-reducer-planetary gears-sun gear.

This scheme has the following characteristics: Firsty, it still adopts variable frequency motor, retaining the high efficiency and strong geological adaptability of variable frequency motor. Secondly, the extricating device is not added separately, and the installed power of the original TBM cutterhead driving system is not increased. Considering that TBM extricating is only a transient process, the mechanical energy storage function of inertia flywheel can be used to provide extra energy to achieve extricating. Thirdly, a hydro viscous clutch must be added in the intermediate link between the variable frequency motor and the cutterhead to work cooperatively with the inertia flywheel. When the cutterhead get jammed, the variable frequency motor can be completely separated from the load by controlling the hydro viscous clutch. After the motor starts under light load, it drives the inertia flywheel to accelerate and store energy. Then, by reasonably controlling the closure of the hydro viscous clutch, the flywheel and the motor cooperate to participate in the TBM cutterhead extricating. Fourthly, due to the separation function of the hydro viscous clutch, the variable frequency motor can avoid the interval with poor torque characteristics at low speed, and then participate in driving the cutterhead after the rotating speed reaches a certain value and ensures the torque output capacity, so that the jamming breakout torque will be greatly improved compared with the starting torque at low frequency. Fifthly, the introduction of a flexible transmission link such as a hydro viscous clutch can avoid the system impact caused by the rigid connection of the motor at the moment of start-up, and it is easy to realize the dynamic balance of torque among multi power sources.

It is not difficult to understand that jamming breakout refers to the dynamic process of TBM cutterhead from getting jammed to resuming normal, continuous and stable operation. As shown in Figure 2, the ideal TBM jamming breakout torque curve shall have a certain duration not less than the rising time of cutterhead speed except that the torque is greater than the normal tunneling torque. In addition, the jamming breakout torque should also meet the characteristics of high precision and good stability. The ideal cutterhead speed rise curve should meet the characteristics of natural speed rise, stability and no impact. The cutterhead suddenly rotates and then gets jammed again, or the cutterhead suddenly rises to the working speed under strong impact load, both of which are not really jamming breakout.

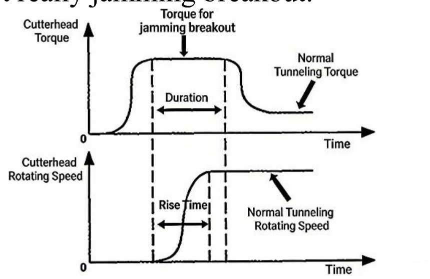

Figure 2. Ideal Variation Curve of Torque and Rotation Speed in TBM Jamming Breakout Process

Therefore, in order to successfully realize TBM jamming breakout, besides providing low-speed and large torque output force, two other factors must be considered: One is torque duration, the second is the impact on the whole system. If the explosive torque is simply output in a short time, without considering the duration, on the one hand, the cutterhead may be jammed again before it reaches the real stable operation state because of the large moment of inertia of the cutterhead. On the other hand, it is likely to bring great impact to TBM cutterhead system. In severe cases, it will cause disastrous consequences to key parts such as sun gear and gearbox of cutterhead system.

Therefore, the main indexes of TBM jamming breakout simulation test bench are as follows:

1.The power grade is 50KW, and the jamming breakout torque is greater than or equal to 2 times of the rated torque.

2.The duration of extricating is more than 1min.

3.Impact degree is lower than $18m/s^3$.

4.The torque control accuracy of hydro viscous clutch shall be within 6%.

The main tasks of TBM cutterhead driving test bench are as follows: It can simulate the working condition of being trapped and locked in TBM tunneling. It can study the control characteristics and torque transmission characteristics of hydro viscous clutch. It can simulate the extricating process of TBM by accurately controlling the oil film of hydro viscous clutch.

3. Design of the Simulated TBM Jamming Breakout Test Bench

The simulated TBM jamming breakout bench is designed for the actual TBM extricating target, and the requirements to be met are as follows:

1.The static and dynamic characteristics of HVC can be tested.

2.It can simulate the load condition when TBM get jammed, and realize the cooperative work of HVC and inertial flywheel to achieve extricating.

3.It can collect the data in the working process, and realize various controls of motor, relief valve and other components.

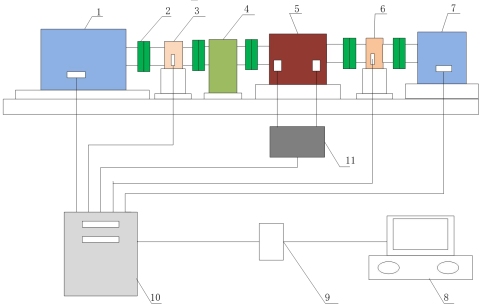

Figure 3. System Diagram of TBM Cutterhead Jamming Breakout Test Bench

1-drive motor; 2-coupling; 3-torque sensor; 4-flywheel set; 5-HVC; 6-torque sensor;7-loading motor; 8-computer; 9-Signalcontrol cabinet; 10-data acquisition card; 11-hydraulic power unit.

Figure 4. Mechanical Structure of TBM Cutterhead Jamming Breakout Test Bench

The schematic diagram of TBM jamming breakout test bench is shown in Figure 3 and Figure 4, which mainly consists of driving device, hydro viscous clutch, load device, rotational torque sensor, flywheel, hydraulic system and electronic control system. The drive motor 1 provides power for the whole system, and the loading motor 6 is used to simulate the TBM cutterhead load to realize accurate loading of different torque curves; The hydraulic oil supply system 10 provides the required lubrication oil and control oil for the hydro viscous clutch; The rotational speed and torque sensor 4 tests the rotational speed value and torque value of the input shaft of the hydro viscous clutch; The data acquisition card 9 collects the detection data and further uses the computer 7 and Labview program to analyze and process the data and carry out matching control on the parameters in the system.

3.1 Mechanical Structure Design

The test bench is driven by AC asynchronous power dynamometer and loaded by AC asynchronous power dynamometer. AC power dynamometer is a new type of dynamic test equipment based on the speed and torque control technology of four-quadrant AC asynchronous motor. It has stable torque, simple and reliable structure, wide range of torque, speed and power test, and is a common driving device for hydraulic viscous and hydraulic transmission tests.

Other commonly used dynamometers include hydraulic dynamometer, eddy current dynamometer, mechanical friction dynamometer and DC dynamometer. Eddy current dynamometer and hydraulic dynamometer only consume energy and do not provide driving force, so they cannot be used as driving equipment. Mechanical friction dynamometer is often used together with hydraulic dynamometer, which is characterized by simple structure and low cost, and can measure the torque value at zero speed working point. The disadvantages are unstable braking at high speed, large brake wear and high heat generation. The response speed of DC dynamometer is fast, but the disadvantages are large volume and inertia. The AC dynamometer is small in size, small in inertia, easy to control, fast in dynamic response and high in test accuracy, so the AC dynamometer is selected as the loading equipment of this test bench.

AC power dynamometer can work in two states: electric and power generation. The driving electric dynamometer to absorb energy from the DC bus and simulate engine operation using a variable frequency speed controlled electric motor. The loading power dynamometer absorbs the energy of the tested object to simulate TBM load in the form of generator, converts the mechanical energy transmitted by the tested object into electric energy and feeds it back to the dragging power dynamometer through DC bus to realize energy closed loop. This function has obvious performance effect in the durability test season, and because the feedback energy only flows on the DC bus of the power dynamometer and does not return to the power grid, it has no interference to the power grid and will not affect the operation of other instruments and equipment on the power grid.

The electric dynamometer has the following outstanding advantages: 1. It can still provide rated torque when the rotating speed is zero, which is very similar to the working condition when TBM get jammed. 2. Compared with the conventional hydraulic system, it has the advantages of high precision, simple structure and convenient control. 3. The resistance torque of the loading motor can be controlled in real time according to the set functional relationship, which can simulate both TBM load and fan water pump load, and has a wide range of applications. 4. The common bus technology is adopted to realize energy recovery, which greatly saves energy.

According to the use requirements of TBM jamming breakout test bench, the rated power transmitted by hydro viscous clutch is 50kW, the rated speed is 1500 r/min, and the rated torque is 318N·m. According to these original input data, some key parts are selected and designed.

(1) Friction plate design

A pair of friction pairs includes an active friction plate with a lining surface and a dual light plate. The lining surface is sintered from the friction material and fired on the metal surface. Compared with ordinary wet clutch, the friction pair of hydro viscous clutch is mainly in oil film shearing condition, which requires that the friction pair material is easy to form oil film and there must be a sufficiently large and relatively stable dynamic friction coefficient. Copper-based materials have the disadvantages of small dynamic friction coefficient and large difference between dynamic and static friction coefficients, while their advantages are good wear resistance, heat resistance and high power density, which is suitable for high-power applications. However, paper-based material has high dynamic friction coefficient and small difference between dynamic and static friction coefficients. Organic matrix composition can make the friction plate have better wettability to oil film, which is more conducive to the formation of oil film. Therefore, it is an ideal material for medium and small power hydro viscous clutch, so paper-based friction plate is selected. The specific parameters of the friction plate finally determined are as follows: The outer diameter of the friction plate is 313.9mm, the inner diameter is 207.3mm, the total thickness s=3.5mm, the pressure angle α=20°, the modulus m=3 and the number of teeth z=71. The outer diameter of the dual plate is 342mm, the inner diameter is 225mm, the total thickness s=3.5mm, the pressure angle α=20°, the modulus m=3 and the number of teeth z = 112.

(2) Design of compression spring

According to the stress condition of spring and the pressing force demand of friction plate, finally determine the relevant dimensions of spring: diameter 4.5mm, outer diameter 22mm, free length 42mm, number of spring = 24, pre-compression amount 10mm.

(3) Piston design

the piston outer diameter is determined according to the following formula

$$n = \frac{6M_0}{\pi f K \psi p_0 d_2^{\,3}(1-C^3)} \tag{1}$$

Where: n—number of friction pairs, M_0—Rated torque, K—reduction factor of compression force, P_0—Actual pressure of friction plate, P_0=0.4-0.5Mpa, f—dynamic friction coefficient, f=0.07-0.08, ψ—ratio of actual area to nominal area, ψ=0.6-0.7, d_2—outer diameter of friction lining, C—ratio of inner and outer diameters of friction lining, C=0.7-0.75.Take n=8, K=0.96, P_0=0.4 -0.5Mpa, ψ=0.6, C=0.75, f=0.07, by calculation, d_2=0.244mm.

3.2 Hydraulic System Design

HVC hydraulic system is divided into lubricating oil circuit and control oil circuit. There are three requirements for the lubrication system: firstly, ensure the oil supply of the friction pair and form enough cooling oil film. Secondly, the flow rate of lubricating oil can take away the heat generated by the slip of friction plates; Third, the pressure of lubricating oil shall not exceed the allowable pressure of other components in the oil circuit. There are three requirements for the control system: first, within the pressure control range of the relief valve, the driving and driven friction plates can be ensured to operate synchronously. Secondly, the minimum cylinder pressure is as small as possible, and the cylinder pressure can be stepless adjusted. Thirdly, the control oil pump flow needs to be within the working flow range of the proportional relief valve.

Based on the above requirements, the lubricating oil circuit adopts low pressure and large flow oil supply, the control oil circuit adopts stepless adjustable small flow pressure, and the hydraulic system adopts double pumps to supply oil and work independently. The system schematic diagram is shown in Figure 5. The left oil circuit is the lubricating oil circuit, and the lubricating oil enters the clutch through the driving shaft, while the right oil circuit is the control oil circuit, and the control oil enters the clutch through the driven shaft. function arranged in this way ensures that the lubricating oil has enough centrifugal force and can fully soak the friction plate. The speed regulating valve 5 and the relief valve 24 of the lubricating oil passage form an inlet throttle speed control circuit, which can change the oil supply of the lubricating oil passage to change the oil temperature for subsequent observation of the relationship between the oil temperature and the system performance. There are two methods to adjust the oil pressure in the control oil circuit. One is direct adjustment by proportional relief valve 15. Considering that the proportional relief valve 15 requires high cleanliness of oil, any impurities and abrasive debris may cause faults, a fine filter 16 is arranged in the circuit. Secondly, the speed of motor 22 is adjusted by frequency converter to change the control oil pressure. The two methods can be used at the same time or work independently, which can make full use of the advantages of fast response of relief valve and high reliability of motor speed regulation. In order to prevent the oil temperature of the system from being too high, an independent cooling circuit 23 is arranged, which consists of a motor, a pump and a cooler, so as to avoid serious deformation of friction plates and damage to other hydraulic components caused by long-term operation of the system.

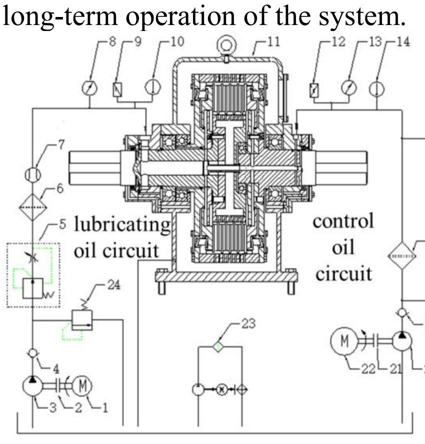

Figure 5. Schematic Diagram of HVC Hydraulic System

1,22-oil pump motor. 2,21-coupling. 3-lubricating oil pump. 4,19-check valve. 5-speed regulating valve. 6, 16-fine filter. 7-flowmeter, temperature sensor. 8, 13-pressure gauge. 9, 12-pressure sensor. 10, 14-Thermometer 11-Hydraulic viscous clutch. 15-Proportional relief valve. 17-Accumulator. 18-Stop valve. 20-Control oil pump. 23-Independent cooling circuit. 24-Direct-acting relief valve

The following is the design and selection of key components.

(1) control oil pump and motor

Considering the advantages of reliable operation, long service life, simple structure, strong oil pollution resistance and self-priming ability of gear pump, and the system belongs to low-pressure system, the control oil pump adopts gear pump. The minimum pressure of the control oil needs to overcome the compression force at the initial position of the spring, and the maximum pressure needs to overcome the compression force at the position of the maximum compression amount of the spring.

$$p = \frac{F}{\pi(r_2^2 - r_1^2)} \tag{2}$$

Where: p-oil cylinder pressure (Pa), F -friction surface pressing force, the minimum pressing force is calculated to be about 10154N, the maximum pressing force is 12616N, r_1-inner radius of a oil cylinder, r_1=80mm, r_2-outer radius of a oil cylinde r_2=112mm. p=0.53-0.64MPa, As the flow is not required too much, gear pump BB-B25 shall be selected and 2.2 kw motor shall be selected for matching.

(2) lubricating oil pump and motor

Gear pump is also selected for lubricating oil pump. The flow of lubricating oil pump shall meet the heat dissipation requirements, and its flow

$$Q = \frac{P_s}{c\Delta t\rho} \tag{3}$$

Where: Q-flow rate(m^3/s), c-specific heat capacity of lubricating oil, c=1674.8J/(kg·°C), ρ-density of oil, ρ=880kg/m^3, Δt-temperature rise of lubricating oil after passing through friction plate gap, Δt=11°C, P_s-maximum power loss, P_s=7.4kW, Q=27.4L/min, gear pump BB-B50(rated pressure of 2.5MPa) was selected and 4 kw motor shall be selected for matching.

(3) Proportional relief valve

According to the pressure range and flow requirements of the control oil circuit, the proportional relief valve is selected as the secondary cartridge valve of Sun hydraulics Company. The pilot stage is a direct-acting electro-hydraulic proportional relief valve RBAP-XDN with a pressure range of 3.5-50bar, and the main stage is a pilot-controlled manual balance slide valve RVBA-LBN with a pressure range of 5-105bar and a flow capacity of 30L/min.

3.3 Design of Electronic Control System

The signal transmission system of the electronic control system of hydro viscous clutch test bench is shown in Figure 6. Sensor signals to be collected include pressure signal, temperature signal, flow signal and torque signal. Analog signals to be controlled mainly include proportional relief valve voltage. Signals to be communicated with frequency converter mainly include motor speed torque, start and stop of motor and cooler fan, etc.

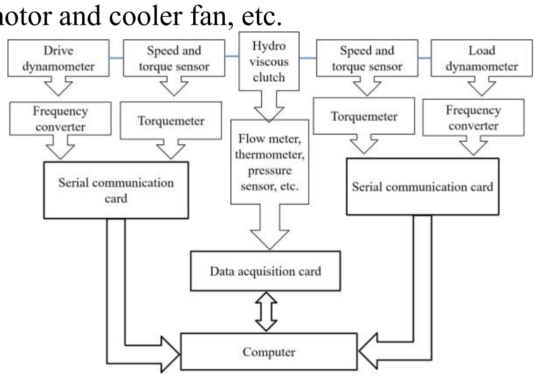

Figure 6. Schematic Diagram of Signal Transmission System of Test Bench

The electronic control system is divided into two parts: hardware and software. The function of the hardware part is signal transmission of analog quantity and digital quantity, while the function of the software part is graphic display, subsequent processing and data storage. At the same time, it should realize the function of executing test bench operation according to user instructions.

The requirements for the electronic control system of the test bench are shown in Table 1. The electric control cabinet of the test bench is divided into power signal cabinet and control signal cabinet.

The power signal cabinet provides power for the electric equipment, and it is necessary to ensure that the electrical system has automatic protection function in case of abnormal conditions. The physical diagram of the power cabinet is shown in Figure 7, which is mainly composed of two dynamometers and frequency converters of oil pump motors, IGBT elements, relays, air switches, circuit breakers, resistance boxes, etc.

The control signal cabinet is used to collect sensor signals and control the on-off of power lines. The control cabinet is mainly composed of industrial computer, operation button, regulated power supply, data acquisition card, connection terminal and RS232-USB converter, etc. The physical diagrams of its front panel and rear panel are shown in Figure 8.

Table 1. Monitoring Requirements of Hydro Viscousclutch Test Bench

Monitoring point	Quantity	Parameter
Power motor	1	50KW
Power motor	1	75KW
Power motor	2	2.2KW
Power motor	3	4KW
Fan	2	4KW
Analog input	5	0-10V
Analog output	2	0-10V
Analog input	8	24V, 0V
Analog output	7	24V, 0V

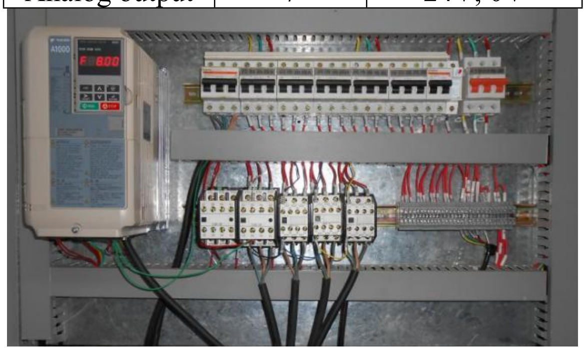

Figure 7. Physical Drawing of Power Cabinet

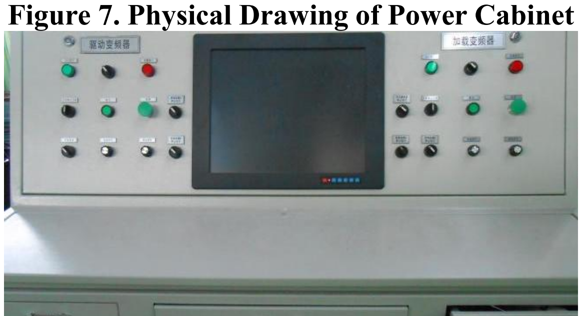

Figure 8. Physical Diagram of Signal Cabinet

4. Conclusion

This work proposes a TBM cutterhead extricating scheme based on hydro viscous clutch to address the issue of TBM cutterhead jamming. The composition and various parts of the simulated TBM cutterhead jamming breakout test bench are selected and designed, including the selection of the mechanical structure of the test bench and the software and hardware design of the electronic control system. The test bench has the following characteristics: 1. No additional installed power for TBM cutterhead drive. 2. It retains the characteristics of high efficiency and strong stratum adaptability of variable frequency motor. 3. Using the mechanical energy storage function of inertia flywheel and the coupling control of hydraulic viscous clutch, the jamming breakout torque can be greatly improved. 4. The oil film shearing transition of the hydraulic viscous clutch can be used to avoid rigid impact on the system during startup. In the future, various TBM cutterhead extricating simulation performance experiments can be carried out on this test bench, which lays an experimental foundation for further proposing TBM cutterhead jamming breakout technology based on hydro viscous clutch.

Acknowledgments

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