Design and study of new riser dynamic load compensation device test system

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Abstract—To ensure the safety of the riser, a kind of new riser dynamic load compensation device is used in this paper, which can effectively reduce the dynamic load of the riser. At the same time, according to a certain similarity criterion, the riser dynamic load compensation test system was designed, including heave simulation system, load simulation system and compensation system. The co-simulation model of the system was established by using ADAMS and Simulink, and the effects of accumulator volume, precharge pressure, piston rod area of compensating hydraulic cylinder and throttle time constant on the system performance were analyzed. The simulation results show that the dynamic load compensation device has a certain compensation effect.

1. INTRODUCTION
The South China Sea is a tropical storm vulnerable areas. Frequent typhoon seriously affects the deepwater drilling operation efficiency and security of personnel and equipment [1]. When a typhoon comes, the drilling platform needs to stop operation and deal with the typhoon. A common way to avoid the platform is to recover all the risers to the platform for evacuation, which increases the non-productive time and operation costs [2]. A portion of the riser is recovered to save costs, while the rest is suspended and withdrawn with the drilling platform. At present, there are mainly hard suspension and soft suspension for riser suspension [3]. In the former case, the riser is rigidly connected to the platform chuck. Under the influence of the platform heave movement, accidents such as overloading of the suspension beam or dynamic compression of the riser are easy to occur. The latter uses a tensioner to connect the riser, but the stroke of the tensioner limits it, and the operation is complicated and time-consuming [4]. In this paper, a hydraulic buffer type riser dynamic load compensation device is
designed to compensate the moving load into the riser. A set of the test system is designed. Through the co-simulation analysis of ADAMS and Simulink, the feasibility of the system is verified.

2. PROPOSAL OF DYNAMIC LOAD COMPENSATION DEVICE FOR RISER

The riser dynamic load compensation device belongs to a new type of suspension scheme, mainly composed of an annular cylinder and suspension sub. The scheme has the characteristics of simple structure, lightweight, easy installation and good stability. A schematic diagram of the scheme is shown in Fig. 1.

![Schematic diagram of riser dynamic load compensation device](image)

1 - cylinder piston rod; 2 - cylinder flange; 3 - annular cylinder; 4 - cylinder flange; 5 - suspension sub; 6 – centralizer

Figure 1 Schematic diagram of riser dynamic load compensation device

The specific implementation process is as follows: when entering the platform escape operation, after the riser is disconnected from the bottom LMRP, part of the riser is recovered, and the rest is stuck at the jaw of the chuck. At this time, the annular oil cylinder and hanging sub are lifted to the drilling head. After connecting the hanging sub with the riser, the clamping jaw is loosened to lower the annular oil cylinder. In the circular cylinder flange stuck again, and then quickly can start the device running after connecting the hydraulic pipeline. Centralizer can turn in bop enclosure movement guarantee remain the state of the vertical riser with chuck [5].

3. TEST SYSTEM DESIGN

3.1. Test System Scheme Design

When the floating drilling platform heave with the sea waves, the dynamic load compensation device makes compensation motion through the hydraulic compensation cylinder [6]. When the acceleration of the riser increases to the set value, the acceleration of the riser does not increase through the motion compensation of the hydraulic cylinder, to achieve the purpose of reducing the peak dynamic load of the riser. According to the working principle of the riser dynamic load compensation device, the designed test system scheme is shown in Fig. 2.

The test system is divided into a heave simulation system, a compensation system, and a load simulation system. In the heave simulation system, the heave movement of the floating drilling platform is simulated by controlling the hydraulic cylinder. In the load simulation system, an adjustable load is used to simulate the load applied to the suspension by the riser system. In the compensation system, the compensation function is realized by controlling the annular oil cylinder.
According to the similarity criterion, this paper designs the specific index parameters of the test system concerning the engineering prototype, as shown in Table 1.

### 3.2 Active control principle of piecewise variable damping

To achieve the compensation effect, the piecewise-variable damping active control technology is used to reduce the load's dynamic load. The principle is shown in Fig. 3.

The load moves sinusoidal under the heave simulation system's drive, and the load's dynamic load before compensation is large. Set the upper and lower limits of load acceleration, when the acceleration does not reach the set value, no compensation; when the platform moves upward from zero, the acceleration gradually increases in the negative direction. The compensation starts when the lower limit is reached at $t_1$. The load acceleration is maintained unchanged until $t_2$ by motion compensation of the compensation cylinder. The period compensation principle of $t_3$-$t_4$ is the same.

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**Figure 2 Schematic diagram of riser dynamic load compensation device test system**

**Figure 3 Diagram of piecewise variable damping control**
4. EXPERIMENTAL SYSTEM MODELLING AND SIMULATION ANALYSIS

The test system of the riser dynamic load compensation device is a complex system of electromechanical and hydraulic integration. The simulation using a single software may only have a one-sided understanding of the system due to the lack of mutual communication. ADAMS has a powerful kinematics analysis function; SIMULINK can build the hydraulic model of the test system, and simulate the hydraulic and control system. The co-simulation can simulate the movement more truly and improve the accuracy of the simulation [7].

4.1. Test System Modeling

![Mechanical system model](image)

1 - Heave hydraulic cylinder; 2 - suspension device; 3 - suspension sub; 4 - compensation hydraulic cylinder; 5 – load

Figure 4 Mechanical system model

The mechanical system model established based on ADAMS is shown in Fig. 4. This article uses Simulink to build a hydraulic control system, including accumulator, throttle valve, etc. The co-simulation principle is shown in Fig. 5. The displacement, velocity and acceleration of the mechanical system output are used as the input of the hydraulic system. The hydraulic system outputs the driving force as the mechanical system input.

The co-simulation model is shown in Fig. 6. The input variable of the hydraulic system is the acceleration of the load, which compensates for the displacement, velocity and acceleration of the hydraulic cylinder, and the relative displacement, velocity and acceleration of the piston of the hydraulic cylinder. The output variables are the position drive of the hydraulic heave cylinder and the piston drive of the hydraulic compensation cylinder.

![Co-simulation schematic diagram](image)
4.2. Hydraulic Key Component Simulation

4.2.1. Influence of Accumulator Volume

In the case of other parameters unchanged, the accumulator volume is changed to carry out simulation research. When the accumulator volumes are 80L, 100L and 120L, the corresponding pressure fluctuations are 0.35MPa, 0.28MPa and 0.25MPa, respectively, as shown in Fig. 7.

The larger the initial volume of the accumulator, the more conducive to the control of the system, but the larger the volume of the accumulator, the higher the system cost and space requirements. Due to the above factors, the initial volume of the accumulator is set to 100L.

4.2.2. Influence of Accumulator Precharge Pressure

The precharge pressure is set as 70%, 80% and 90% of the lowest pressure of the system. The corresponding pressure fluctuations of 3.39MPa, 3.84MPa and 4.32MPa are 0.312MPa, 0.288MPa and 0.267MPa, respectively, as shown in Fig. 8. From the perspective of system control, the smaller the pressure difference between the two ends of the throttle valve, the better the control. Therefore, the greater the accumulator precharge pressure, the better. Combined with the actual situation, the accumulator precharge pressure is selected as 4.25MPa.
4.2.3. Influence of annular cylinder operating area
Circular cylinder using differential forms for connection, the effective area is the area of the piston rod. The force of the piston rod is the product of the pressure of the hydraulic cylinder and the area of the piston rod. When the force is certain, the smaller the area of the piston rod, the greater the pressure in the rod-free cavity of the hydraulic cylinder. When the platform carries out sinusoidal heave movement, the pressure of the hydraulic cylinder reaches the minimum and maximum at the peak and trough of the wave. The smaller the area of the piston rod is, the greater the pressure difference will be. When the compensation rate set by the system is certain, the range of the upper and lower limits of the set pressure will be larger. The smaller the piston rod action area, the more conducive to control, but in the actual situation, due to the cost and processing accuracy of the reason, the piston rod area is limited, so the selection of the piston rod area of 44mm².

4.2.4. The Influence of Throttle Parameters
The throttle valve is the core component of the hydraulic control system. The time constants of the movement of the valve core of the throttle valve were set as 0.002, 0.004, 0.01 and 0.02, and the results were shown in Fig. 10.
When T is 0.002, the throttle valve can respond quickly and reach a stable state through small-amplitude oscillation. With the increase of T value, the response speed of the valve becomes slower, the amplitude of oscillation becomes larger, and the adjustment time becomes longer. However, the smaller the T value, the higher the requirement for the valve, and the worse the control stability. Considering comprehensively, T is selected as 0.004.

4.3. Analysis of Simulation Results
Dynamic load compensation effects are shown in Fig. 11. The load acceleration is 0.92 m/s² before compensation, and the maximum load acceleration is 0.652 m/s² after PID controller compensation, and the compensation rate is 29.2%.

Figure 11 Acceleration compensation effect comparison

5. CONCLUSION
According to the working principle of the dynamic load compensation device of the riser, a set of dynamic load compensation device test system is designed, including two parts: mechanical system and hydraulic control system. The following conclusions can be drawn through the establishment of the model and joint simulation analysis: The riser dynamic load compensation device combines the advantages of hard and soft suspension devices, and because its working space is above the drill floor, the operation is simple and safe. Through simulation, the optimal parameters of key hydraulic components are determined. The simulation results show that the dynamic load compensation test system can effectively reduce the acceleration of the load within a limited distance, that is, to reduce the dynamic load, which provides a reference and basis for engineering application.

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