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Design and Simulation Analysis of Semi-Active Lifting and Sinking Compensation System

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Abstract: The lifting and sinking motion generated by the drilling platform causes large fluctuations in the suspension tension of the large hook, which affects the stability of the drilling pressure acting on the rock at the bottom of the well, greatly reduces the efficiency of drilling and decreases the service life of the drill bit and the drill column under the action of cyclically varying loads, causing an increase in the overall drilling cost. In order to avoid the above situation, a nonlinear model of a semi-active lift-sink compensation system is designed and a method to improve the compensation efficiency of the system is proposed. Firstly, the relevant physical model of the hydraulic system is established, and then its simulation model is built according to the working principle of the semi-active lift and sink compensation system. Then the effects of different factors are added to the simulation model of semi-active lifting and sinking compensation, and finally the effects of different factors on the compensation efficiency of the active lifting and sinking system are compared. The simulation results show that a regular sinusoidal wave with a peak value of 6.5 m and a period of 10 s as the ship's lifting and sinking displacement input will reduce the compensation efficiency of the semi-active lifting and sinking compensation system, while the smaller the stiffness of the wire rope, the better the compensation effect of the drill column and the higher the compensation accuracy. The proposed method can effectively improve the compensation efficiency of semi-active lifting and sinking compensation system.

Keywords: Lifting and sinking compensation, Simulation, Hydraulic cylinder, Semi-active.

1. Introduction

At present, the lifting and sinking compensation devices used on deepwater floating drilling rigs in China need to be imported from abroad at high prices [1], and the relevant design and manufacturing technologies of lifting and sinking compensation devices are not yet mastered in China. Therefore, to carry out in-depth research on the dynamics of deep-water drilling systems, find the laws and mechanical characteristics of the lifting and sinking motion of the drilling column system and the lifting and sinking compensation device in the process of deep-sea drilling, and on this basis, develop a special overhead crane lifting and sinking compensation device for deep-water floating rigs with China's own intellectual property rights, is the technical key and equipment support for China to realize safe and efficient drilling operations in the deep sea.

The patent proposed by Jones and Cherbonnier in 1990 was one of the first active lifting and sinking compensation systems using a microprocessor to control it [2]. Professor Fang Huachan has conducted many in-depth studies on the lift and sink compensation equipment for mother ships in offshore drilling and equipment, and discussed the specific structure and layout form, system working principle and design method of the open-sink compensation device installed on the mother ship of offshore drilling [3]. Wu Baihai, Xiao Tibing et al. proposed an active lifting and sinking compensation system for deep-sea mining equipment in 2003 and 2004, and the results of their own simulation and simulation tests showed that the system is feasible and has high compensation accuracy [4]. Zhou Li et al [5] designed a fuzzy predictive control algorithm for the time lag of the active compensation system, which improved the stability of the system and ensured the anti-disturbance of the system at the same time.

At present, domestic and foreign research on lift-sink compensation systems mainly focuses on the improvement of mechanical structures and control algorithms [6-9], and few consider the nonlinear forces on hydraulic cylinders in modeling. In fact, the nonlinear forces on hydraulic cylinders can make the system response slower and errors increase during use. Nonlinear modeling of hydraulic cylinders can improve the accuracy of the modeling of lift-sink compensation systems and thus obtain simulation results that are closer to reality. This study focuses on modeling the nonlinear frictional force and nonlinear spring force applied to the hydraulic cylinder, based on which a nonlinear modeling method for a semi-active lift-sink compensation system is proposed and the effect on the lift-sink compensation efficiency is analyzed.

2. Hydraulic Cylinder Model

2.1. The kinetic model of hydraulic cylinder.

The hydraulic cylinder is the execution element pestle of the hydraulic system, which transforms the pressure energy of the hydraulic pump output liquid into mechanical energy output, so as to drive the external load, and its dynamics model is shown in Figure 1.

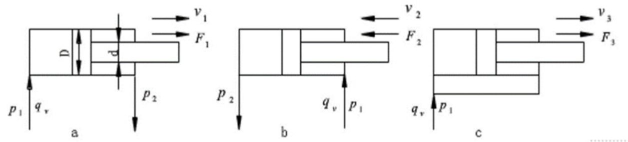

Figure 1. Piston cylinder force analysis

Single-piston rod hydraulic cylinder parameters calculation. As shown in Figure1 a, the expression of the piston rod extension speed is:

$$v_1 = \frac{4q_v}{\pi D^2}$$

The force output from the piston rod is:

$$F_1 = (p_1 - p_2)\frac{\pi D^2}{4} + p_2\frac{\pi d^2}{4}$$

As shown in Figure 1 b, the velocity of the piston rod retreating is:

$$v_2 = \frac{4q_v}{\pi(D^2 - d^2)}$$

The force output from the piston rod is:

$$F_2 = p_1\frac{\pi(D^2 - d^2)}{4} - p_2\frac{\pi D^2}{4}$$

As shown in Figure 1c, the force output from the piston rod is:

$$F_3 = p_1\frac{\pi d^2}{4}$$

The piston extends at the velocity of:

$$v_3 = \frac{4q_v}{\pi d^2}$$

2.2. Force analysis of hydraulic piston cylinder

The pull sub-stiffness of the piston cylinder can be calculated using the calculation methods of fluid mechanics and fluid-solid coupling theory, but the calculation process requires finite element analysis, which is more complicated and time-consuming. Considering the error within 7% between the stiffness calculation method according to compressible fluid and the above finite element calculation method, the active cylinder pull-pressure stiffness is calculated according to the following formula [9].

$$k_a = \frac{EA}{L}$$

Where:

E——Bulk modulus of elasticity of the hydraulic medium, MPa.

A——area of the end face, m^2.

L——axial length, m.

2.3. Passive compensated hydraulic spring stiffness

Considering the working gas cylinder as part of the accumulator and the gas compression as an isothermal process, the liquid-gas spring stiffness of the passive compensation part can be calculated from the total volume of the accumulator, the average pressure and the piston area according to the gas law and the definition of the liquid-gas spring stiffness as follows:

$$k_g = \frac{A_s^2 P_s}{V_0}$$

Where:

A_s——Accumulator piston area, m^2.

P_s——Average accumulator pressure, MPa.

V_0——Total volume of the accumulator, m^3.

Assumptions. x_1 is the platform rises and falls in displacement. x_2 is the displacement of the piston of the compensation cylinder, and m_1 is the mass of the platform, the m_2 is the mass of the piston and the piston rod, take a small piece of separator at the piston of the compensation hydraulic cylinder, when the platform rises on its force analysis can be known its force: the liquid gas spring recovery force F_1, and its value is $k_1(x_1 - x_2)$, When the platform rises, the storage cylinder gas is compressed, its state for the extension, its direction to the right. Compensate the damping of the hydraulic cylinder liquid f_1 , whose magnitude is $c_1\left(\frac{dx_1}{dt} - \frac{dx_2}{dt}\right)$, its direction is opposite to the direction of piston movement. Piston rod inertia force F_{a1} . Its magnitude is $m_2\frac{d^2x_2}{dt^2}$, and its direction is opposite to the piston motion.

The piston is subjected to a force acting on it by the drilling column F_2 and its direction is downward.

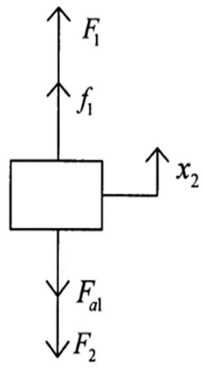

Figure 2. Compensating cylinder piston force analysis

By force analysis, the kinetic equations of the piston and the drilling column are obtained as

$$k_1(x_1 - x_2) + c_1\left(\frac{dx_1}{dt} - \frac{dx_2}{dt}\right) = \frac{d^2x_2}{dt^2}m_2 + F_2$$

Letxs be the strain displacement between the lift-sink compensator and the drill column, and the tension acting on the drill column is F_2 and its value is k_2X_3, through analysis, it is known that the displacement strain of the drilling column should be equal to the difference between the rising displacement of the platform and the displacement of the piston movement:

$$F_2 = k_2X_3$$

$$x_3 = x_2 - x_1$$

2.4. Analysis of the force of the wire rope

According to the size of the drill column under working conditions, from the material mechanics, we can get

$$k_w = \frac{ES}{l}$$

Where:

k_w——Wire rope stiffness, N/m.

E——Modulus of elasticity of the drilling column, Pa.

S——Cross-sectional area of the drilling column, m^2.

l——Wire rope length, m.

2.5. Longitudinal force analysis of drilling column

Drill column stiffness can be determined based on the drill column size and well depth[14]:

$$k_D = \frac{A_D E}{L_D}$$

A_D——Equivalent cross-sectional area of drill string, m^2 ;
L_D——Length of drilling column, m.

Longitudinal vibration model of drilling column under the action of compensation device

Experience in offshore oil development shows that floating rigs drifting more than a few meters from their mooring points will result in fatigue damage or outright breakage of the drill column, causing unnecessary downtime and significant economic losses. For this reason, it is extremely important and necessary to compensate for the longitudinal motion of the drill column. Although the lifting and sinking compensation device greatly improves the working environment of the drill bit in the floating drilling system, the dynamic load caused by the vibration of the drill column has a nonnegligible effect on the effect of the compensation device. Therefore, a study of the longitudinal vibration of the drill column after the installation of the compensation device is a necessary part of improving the efficiency of the lifting and sinking compensation device. In this project, based on the consideration of the structural characteristics of the active-passive combined overhead crane lifting and sinking compensation device and the knowledge related to the dynamics of mechanical system, the vibration model of its drill column is established as shown in Figure 2-2.

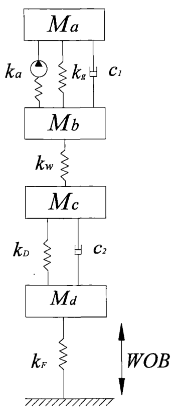

Figure 3. Drill column vibration model

The lifting and sinking motion is a damped vibration caused by wave excitation, and the law is similar to a sine wave.

The platform has 6 degrees of freedom of motion, and the crane compensation device does not consider the reaction force on it.

The main cylinder block, the driven cylinder block and the support structure move together with the derrick and the platform, so the above components are considered as rigid concentrated masses M_a.

The floating crane and the lifting and sinking moving parts are connected by the piston rod of the compensation cylinder, and the piston, piston rod and floating crane are regarded as concentrated masses M_b, The fluid stiffness of the active compensation cylinder k_a ,passive compensation part fluid gas spring stiffness k_g, The fluid viscous damping coefficient c_1.

The tensioned part of the crane above the neutral point of the big hook, top drive and drill column is regarded as the concentrated mass M_C, The rigidity of the floating crane, which is connected with the floating crane by a wire rope, is k_w .Drill column equivalent stiffness coefficient k_D The coefficient of viscous damping in the mud is c_2.

The pressurized part of the drill column below the neutral point is considered as concentrated mass M_d, and the contact stiffness factor of the formation k_F.

The drilling pressure is the contact force between the drill bit and the formation, and its variation value can be estimated by the following formula.

$$\Delta WOB = x_b \cdot k_e$$

Where:

x_b——Displacement of the overhead crane relative to the seabed.

k_e——Effective stiffness between the overhead crane and the ground.

$$k_e = \frac{1}{\frac{1}{k_w} + \frac{1}{k_D} + \frac{1}{k_F}}$$

Where:

k_w——Wire rope stiffness between the overhead crane and the travel shop, KN/m.

k_D——Drill column equivalent stiffness, KN/m.

k_F——Floor contact stiffness, KN/m

In the longitudinal vibration model of the drill column under the action of the active-passive joint overhead crane lifting and sinking compensation device, the drill pipe is assumed to be a homogeneous equal diameter elastic rod, and the spacer pipe is considered to be vertically downward, with no bending deformation of the drill column and no friction with the pipe wall.

In addition, the main differences between this model and the vibration model under the action of the rover lifting and sinking compensation device are.

(1) The tensile and compressive stiffness of the active compensation part is considered;

(2) The rigidity of the strata is taken into account, instead of considering the drill collar as a rigid body connected to the strata;

(3) The drilling pressure variation calculation is based on the effective stiffness between the overhead crane and the formation, not just the equivalent stiffness of the drill column.

Derivation of system transfer functions:

According to the longitudinal vibration model of the drill column under the action of dead rope compensation device, the concentrated mass is taken as the separator, and the respective equations of motion are established, so as to solve the transfer function of the system. Suppose the platform lifting and sinking displacement is x_a and the displacement of the concentrated mass of the drilling column is x_b and the displacement of the lower drilling column is x_c.

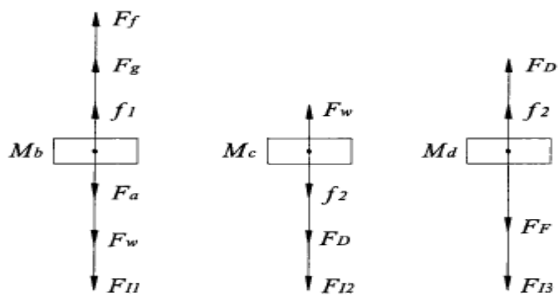

Figure 4. Force analysis

The separators are taken at each concentrated mass, and the forces are analyzed separately, and the meanings of each conform to the following in Figures4:

Concentrated mass M_b, hydro-pneumatic spring recovery force F_g.The size is the product of the liquid-gas spring stiffness coefficient and the absolute deformation, and the direction is the same as the direction of platform movement; the oil damping force f_1. The size of the fluid viscous damping coefficient is the product of the absolute velocity of the fluid in the compensation cylinder, and the direction is the same as the direction of the platform movement; the friction force F_f. The value of the friction force is complex, according to the test, the direction is consistent with the direction of platform movement; active cylinder thrust F_a.The direction is opposite to the direction of platform movement; wire rope force F_f, direction downward; inertia force F_{I1} , direction downward.

Concentrated mass M_C , the elastic recovery force of the drill column F_D the product of the equivalent stiffness coefficient of the drill column and the absolute deformation, in the downward direction; mud damping force f_2, the size of which is the product of the mud viscous damping coefficient and the absolute motion speed of the drill column, in the downward direction; wire rope force F_w, direction upward; inertia force F_{I2}, direction downward.

Concentrated mass M_d , the elastic recovery force of the drill column F_D, the size of which is the product of the equivalent stiffness coefficient of the drill column and the absolute deformation, in the upward direction; mud damping force f_2 , the size of which is the product of the mud viscous damping coefficient and the absolute motion speed of the drill column, in the upward direction; the formation elastic recovery forceF_F direction down; inertia force F_{I3}, direction downward.

Based on the force analysis, the differential equations of the mathematical model of the system are obtained based on Newton's law and the force balance equation as follows[10].

$$M_{\mathrm{b}} \frac{d^2 x_a}{dt^2} = F_w + F_a - F_{\mathrm{f}}$$

$$M_c \frac{d^2 x_b}{dt^2} + c_2 \frac{dx_b}{dt} + k_D x_b = -F_w$$

$$M_d \frac{d^2 x_c}{dt^2} + c_3 \frac{dx_c}{dt} + k_F x_c = -f_2 - F_D$$

Applying the Laplace transform to the above differential equation, it is

$$M_b s^2 X_a(s) = k_w[X_a(s) - X_b(s)]$$

$$M_c s^2 X_b(s) + c_2 s X_b(s) - k_D X_b(s) = -k_w[X_a(s) - X_b(s)]$$

$$M_d s^2 X_c(s) + k_F X_c(s) = -c_2 s X_c(s) - k_D X_b(s)$$

The transfer function model of the longitudinal vibration system of the drill column under the lift-sink compensation device is derived from the Laplace transform equation and modeled in simulink as shown in Figure 5.

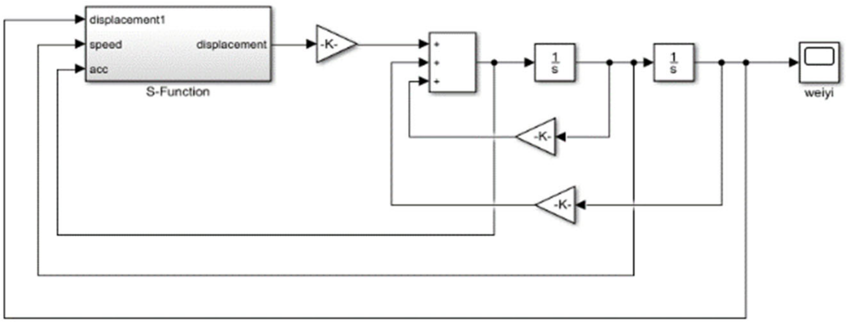

Figure 5. Matlab--simulink model drilling column equivalent stiffness

2.6. Accumulator pressure and volume calculations.

Considering the gas compression of the accumulator as an isothermal process,thgaa hfn -gas spr sess,thgas spring stiffness of the passive compensation part can be calculated from thetotal volume of the accumulator, the average pressure and the piston area.

$$k_g = \frac{A_s^2 P_s}{V_0}$$

Where:

A_s——Accumulator piston area (m^2);

P_s——Average pressure of accumulator (MPa);

Vo——The total volume of the accumulator (m^3).

The static force on the driven cylinder is required to keep the load (drill column and drill post) up and the force required is

$$F_{cyl} = \frac{F_w}{2} = \frac{F_L}{2}$$

Where:

F_{cyl}——The force acting on the accumulator by the passive cylinder, N

F_w——The role of the load force on the wire rope, N

Due to the pressure of the nitrogen tank on the mold, a force is generated with a magnitude of:

$$F_N = (F_{CL} + F_{GC}) \frac{A_{CR}}{A_A} - M_{AA}$$

$$F_N = (F_{CL} + F_{GC}) \frac{A_{CR}}{A_A} - F_{GA}$$

$$P_N = \frac{F_N}{A_A}$$

Where:

F_{CL}——Load force acting on the master cylinder by the wire, N (varies withflow)

F_{GC}——Gravity force on the driven cylinder, N

A_{CR}——Accumulator piston rod area, mm^2

A_A——Accumulator piston area, mm^2

M_{AA}——Gravity force on the accumulator, N

The sum of the forces acting on the piston rod of the driven cylinder is expressedas follows:

$$F_c = M_{CR}a_c = -f_{cl}v_c + F_{AA} + F_{heave} - F_{df} - F_{sf} - F_{DeltaP}$$

Where:

Fc——all external forces on the driven cylinder, N

M_{CR}——accumulator piston rod mass, N

a_c——Accumulator piston rod acceleration, m/s

F_{AA}——Accumulator force on the driven cylinder, N

F_{heave}——Force acting on the piston rod by the lifting and sinking motion of theplatform, N

F_{df}——passive cylinder dynamic friction, N

F_{sf}——static friction of the driven cylinder, N

F_{DeltaP}——Force generated by the pressure drop in the pipe and valve, N

The equation of motion of the accumulator (neglecting nonlinear friction) is [11]:

$$\begin{cases} F_{ACA} = M_{AA}a_a = -B_p v_a + A_{AA}P_1 - A_{AA}P_2 + F_{cyl} + \Delta F_N \\ \dfrac{V_{10} + A_{AA}x_a}{\beta}\dot{P}_1 = -A_{AA}v_a + Q \\ \dfrac{V_{20} + A_{AA}x_a}{\beta}\dot{P}_2 = -A_{AA}v_a + Q \end{cases}$$

$$F_{cyl} = F_C\frac{A_{CR}}{A_A} - F_{Deltap}$$

Where.

F_{ACA}——Accumulator subjected to external force and, N

M_{AA}——mass of the accumulator as a whole, N

a_a ——Acceleration of the accumulator part with the platform motion, m/s^2

B_p——Viscous friction coefficient inside the accumulator

A_{AA}——Accumulator interface cross-sectional area, mm^2

Va——accumulator inflow/outflow velocity. m/s

P1、P2——Accumulator piston chamber and rod chamber initial pressure, Mpa

V_{10} 、 V_{20}——Accumulator piston chamber and rod chamber initial volume, m^3

Xa—— the distance to the center of the piston in the active accumulator, ml

Q——Accumulator's frontal flow rate, L/s

F_{cyl}——Force exerted by the piston rod of the driven liquid cylinder, N

Due to the lifting and sinking motion of the platform, there are two processes of filling and deflating the accumulator, and by the law of conservation of energy there are.

$$PV^{\gamma} = P_0V_0^{\gamma}$$

where the pressure fluctuation of the accumulator can be expressed as

$$\Delta P_N = \left(\frac{V_0}{V_0 - A_{AA}(x_a - x_{a0})}\right)^{\gamma} P_N - P_N$$

$$\Delta F_N = A_A \Delta P_N$$

Stiffness factor k_1: $k_1 = \dfrac{A_2^2 p_0}{V_0 - A_2 x}$

Where:

A_2——working area of auxiliary gas cylinder, m^2;

p_0 ——initial working pressure of the gas in the accumulator, Mpa;

x——the displacement produced by the piston, m;

In summary, the parameters of the compensation cylinder and accumulator areshown in the table.

2.7. PID control principle

Conventional PID control controller is one of the most basic and widely used controllers, which has the advantages of simple algorithm, good stability and reliability.The regulation law of conventional PID controller is very effective for quite a number of control objects, especially for linear constant systems, and the quality of its regulation process depends on the adjustment of each parameter of the PiD controller PID is a linear controller, which controls according to the given deviation and the amount of its deviation change:

The form of the continuous equation of the PID control law [13]:

$$u(t) = K_p(e(t)) + \frac{\int_0^t e(t)dt}{T_i} + T_d\frac{dr(t)}{d_t}$$

The PID control law is discretized into the form of a differential equation:

$$u(k) = K_p(e(k)) + K_I\sum_{j=0}^{k} e(j)T + K_D(e(k) - e(k-1))$$

Where:

u——Controller output signal;

e——deviation signal:

T_i——integration time constant.

T_d——Differential time constant:

Kp——proportional gain.

K_i——integral gain

K_D——Differential gain.

T——sampling period.

K——sampling serial number.

PID regulation is composed of three types of regulation: proportional, integral and differential, and their respective roles are as follows:

Proportional regulation P: is proportional to the deviation of the response system, once the system deviation, proportional regulation immediately produce regulation to reduce the deviation. A large proportional role can speed up the adjustment and reduce the error, but too large a proportion makes the system less stable, and even causes the system instability.

Integral regulation effect I: makes the system eliminate the steady-state error and improve the non-differential degree. The addition of integral regulation makes the system less stable and the dynamic response slower.

The differential regulation action Dreflects the rate of change of the deviation

signal of the system and can anticipate the trend of the deviation change, so it can produce an overshoot control effect, which has been eliminated by the differential regulation action before the deviation is formed, and can improve the dynamic performance of the system. In the case of suitable selection of differential time, overshoot can be reduced and regulation time can be reduced. The differential action has an amplifying effect on noise disturbance, so too strong addition of differential regulation is not good for the system to resist disturbance. In addition, the differential response is the rate of change, and when the input does not change, the differential action output is zero. The differential action cannot be used alone and needs to be combined with two other regulation laws to form a PID controller.

3. Simulation Modeling of Semi-active Lift and Sink Compensation

Compound system sinking compensation compensation effect through the hydraulic pump and accumulator part of the mutual cooperation, to achieve the reciprocating motion of the hydraulic piston rod, to achieve the effect of compensation sinking. The rod cavity of the hydraulic cylinder is connected to the accumulator and the hydraulic pump, and the accumulator and the hydraulic pump provide hydraulic oil pressure to the rod cavity to bear the load of the system. When the rig moves upward with the wave, the load of the system becomes larger, and the force of the wire rope on the overhead crane becomes larger, driving the overhead crane downward to offset the lifting and sinking movement of the crane. As the crane moves down, the oil in the passive compensation chamber of the lift and sink cylinder is driven to the accumulator, which reduces the volume of the gas inside and increases the pressure inside to bear the load of the enlarged crane. The joint simulation model based on position feedback is shown in Fig. 6, and the mathematical model of position feedback in matlab is shown in Fig. 7.

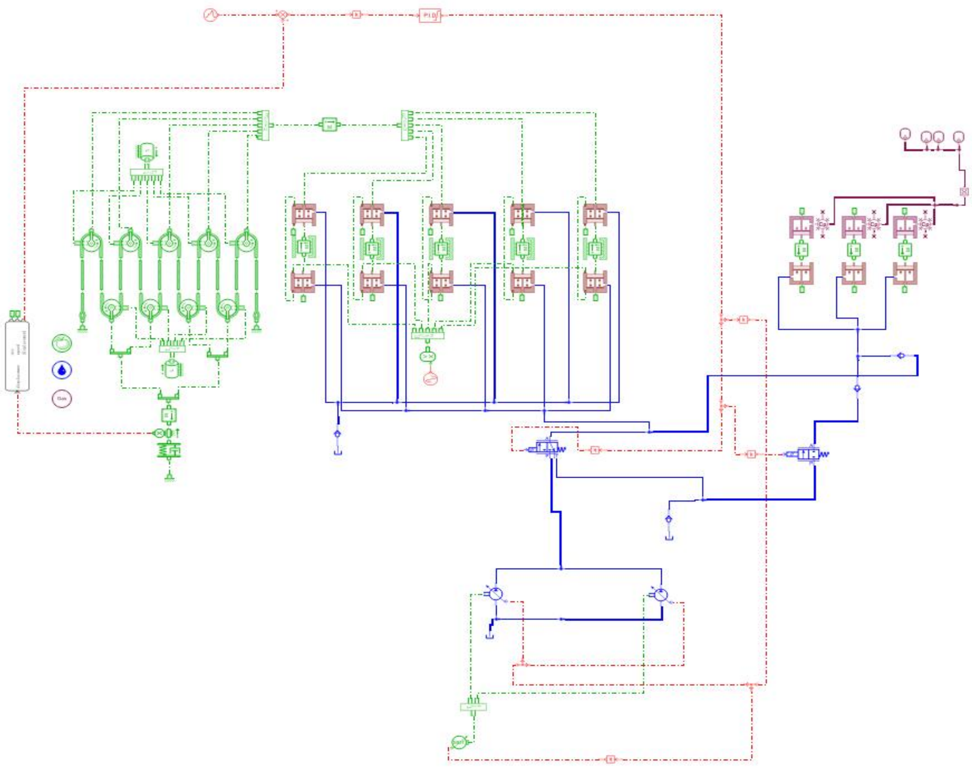

Figure 6. Position feadback model

Through the AMEsim software simulation, the signal module will move the piston rod speed, acceleration, displacement or pressure and the platform signal superimposed as the input of PID control, after the control module calculates the output to the three-way four-way valve, do dlosed-loop PID operation, control the direction of the reversing valve, to achieve the purpose of regulating the expansion of the active cylinder, its working principle is the same as the joint simulation.

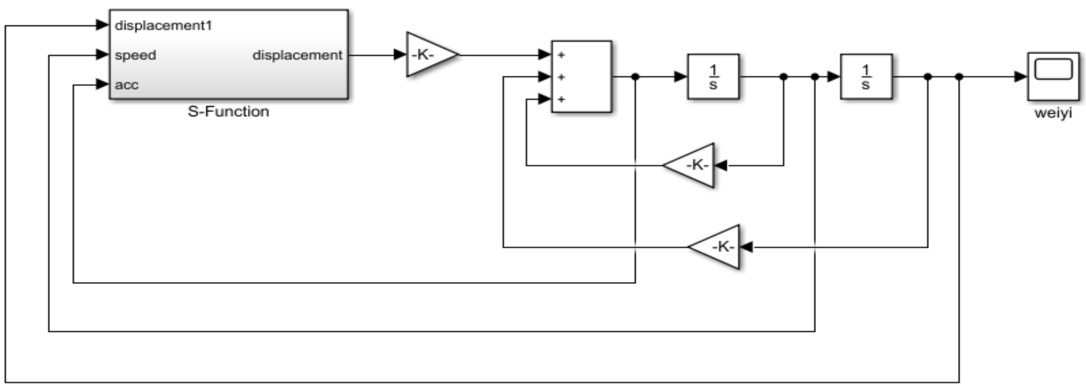

Figure 7. Mathematical model of longitudinal vibration position feadback of drillingcolumn.

4. Simulation Results and Analysis

According to the experimental requirements, the length of drilling column is 3500m, the outer diameter of drilling column is 168.3mm, the weight of drilling tool is about 450t, the mass of top drive is 35t, the modulus of elasticity of drilling column material is 2.06*10^5MPa, the density of drilling fluid is1.2-1.8 $kg \cdot m^3$. The contact stiffnessof the formation is 1000Kn/m.

Amplitude and period of ascending and descending motion

The lifting and sinking motion of the platform by the action of waves and currents is a random vibration whose displacement is a function of time, and usually the law can only be found by probability statistics. In order to facilitate the verification, the platform motion law can be expressed by a sine function [11][12]:

$$H_i = A\,sin(\,wt)$$

In the formula:

H_i ——the lifting and sinking motion curve of the floating platform.

A——wave height, m.

According to the experimental requirements, the wave height is 6.8m/4.8m/2.8m, respectively, and the period is 10s

(1) The influence of environmental factors"

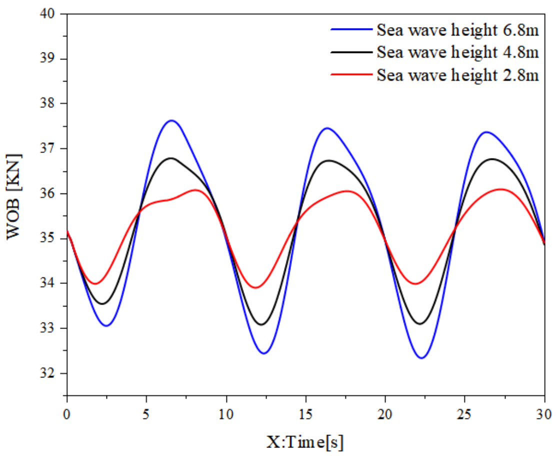

Figure 8. Effect of different sea state on drilling pressure fluctuation

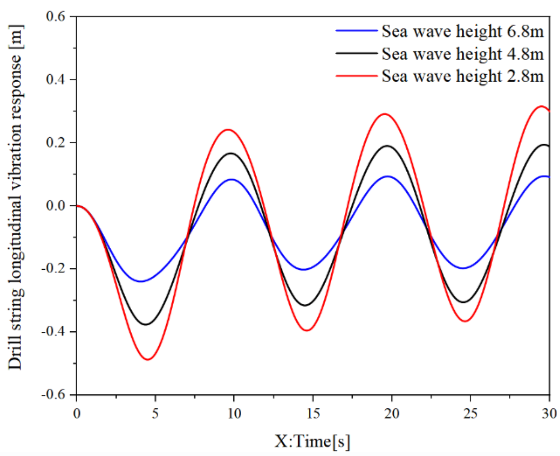

Figure 9. Effect of different sea states on compensation accuracy

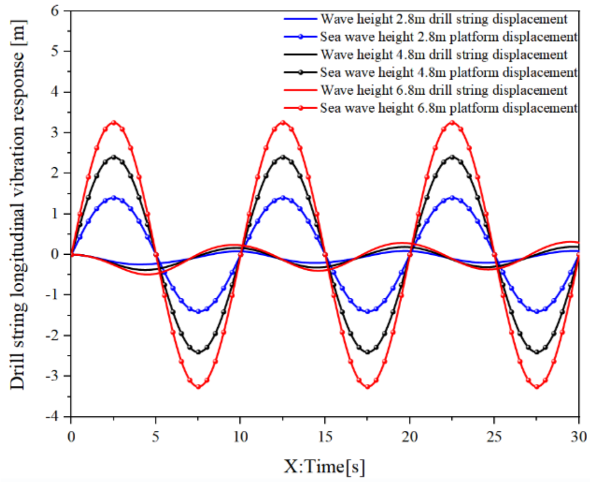

Figure 10. Comparson of drilling column and platform displacement under different sea conditions

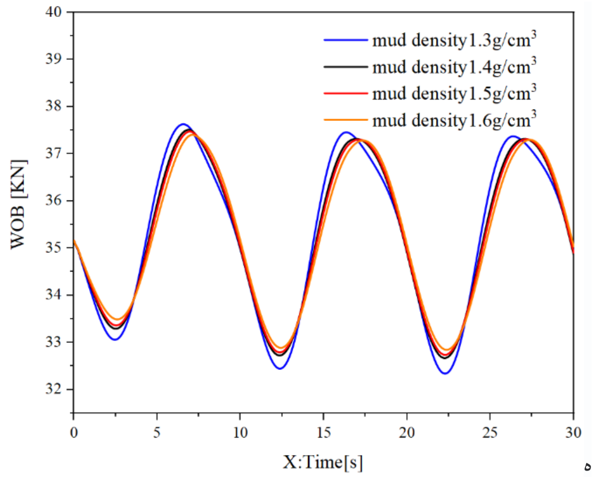

Figure 11. Effect of mud density on drilling pressure fluctuation Figure

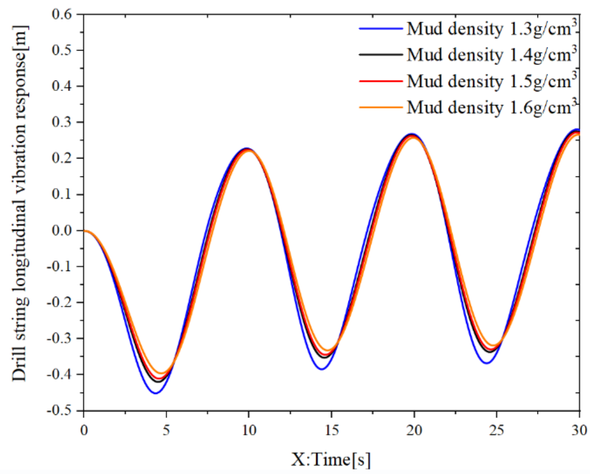

Figure 12. Effect of mud dansity on compansation accuracy

(2) The influence of internal system parametersI

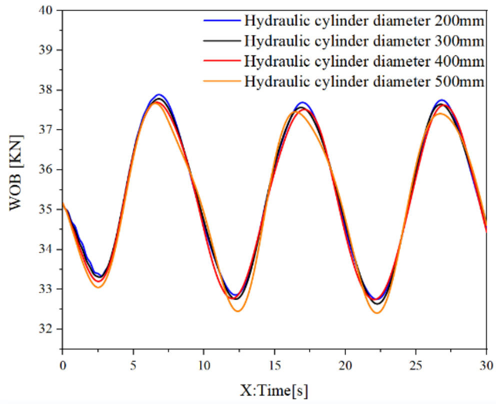

Figure 13. Effect of eylinder diamatar on drilling pressure fluctuation

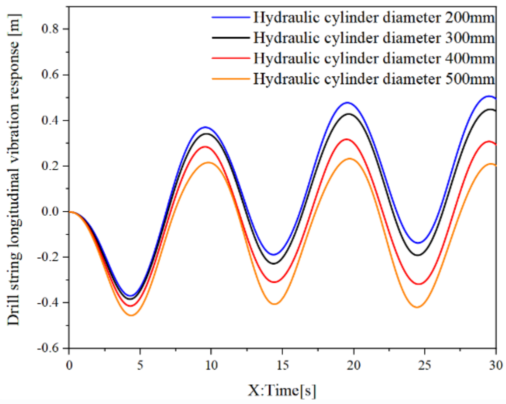

Figure 14. Effect of cylinder diameter on compensation accuracy

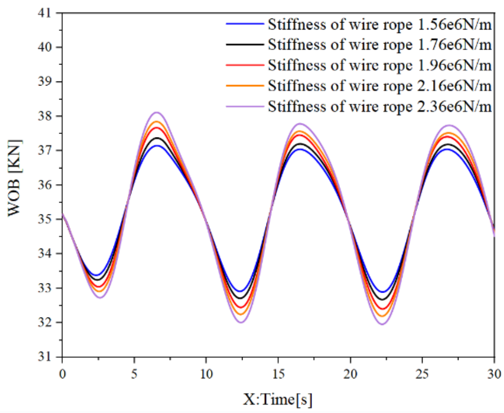

Figure 15. The effect of wire rope stiffness on drilling pressure fluctuation

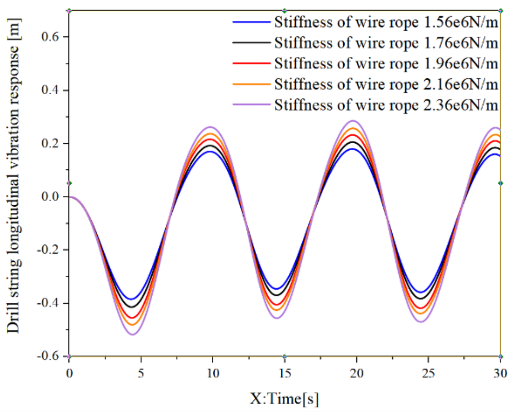

Figure 16. The effect of wire rope stiffness on compensation accuracy

(3) The influence of different control methods

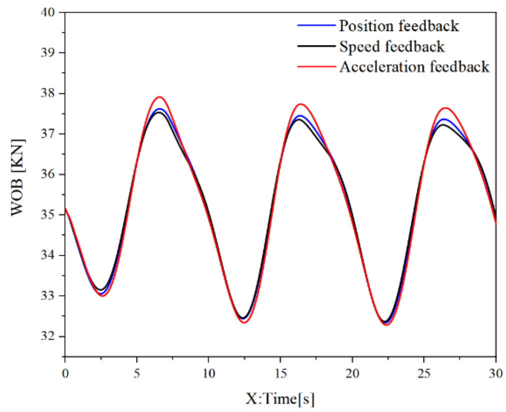

Figure 17. Effect of different position control on drilling pressure fluctuation

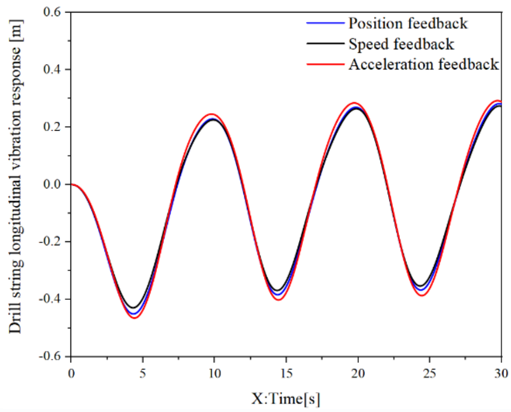

Figure 18. Effect of different position control on compensation accuracy

Effect of different position feedback methods on independent compensation systems:

The lifting and compensation compound system utilizes traditional PID control, which allows the feedback of the drilling column moving speed acceleration and displacement into the controller, and the normal operation of the hydraulic circuit by changing the parameters of P, I and D to achieve the control of the hydraulic cylinder piston movement. Different control methods, with different degrees of control of the hydraulic circuit, lead to different results in the longitudinal movement of the drill column. Choosing a wave period of 10s and a lift of 6.8m, the displacement curves of the drill column with displacement feedback, velocity feedback and acceleration feedback of the independent compensation system were obtained by the joint simulation model as shown in Figure 18, and the compensation efficiency of the three position feedback methods were calculated as follows.

Displacement feedback compensation efficiency $e = 1-\frac{\Delta x}{x} = 90.01$:

Speed feedback compensation efficiency $e = 1-\frac{\Delta x}{x} = 89.6$

Acceleration feedback compensation efficiency $e = 1-\frac{\Delta x}{x} = 84.7$

Then the compensation accuracy of lifting and sinking of displacement feedback is about 90.01%, the compensation accuracy of lifting and sinking of velocity feedback is about

89.6%, and the compensation accuracy of lifting and sinking of acceleration feedback is about 84.7%, then the compensation accuracy of velocity feedback is the highest and the compensation effect is the best in the composite system of lifting and compensation based on position feedback. Figure 17 shows the moving speed of the drill column of the composite system of lifting and compensation based on position feedback, and the drilling pressure fluctuation of the three feedback methods is compared, and the maximum fluctuation amplitude is 5.02KN, which meets the design requirements.

The wave amplitude and different feedback signals have an effect on the compensation effect of the lifting and sinking compensation system. In order to study the change law between them, different wave amplitudes and feedback signals are selected for joint simulation analysis, and the results are shown in Figure 10. From Fig.9, it can be seen that under the condition of the same feedback signal, the compensation effect of the lifting and sinking compensation system gradually becomes worse as the wave amplitude increases; under the condition of the same wave amplitude, different feedback signals correspond to different compensation effects. In order to compare the effect of different operating water depths on the compensation effect, we need to calculate the fluctuation of drilling pressure corresponding to the three operating water depths, and then determine the effect of different operating water depths on the compensation effect by comparing the fluctuation of their drilling pressure.

Figure 9 shows the longitudinal vibration response of the drill column corresponding to different wave heights, and the analysis results are as follows. When the wave height is 6.8m, the fluctuation of the bottom of its drill column is0.95, then its compensation effect is90.01%.

When the wave height is 4.8m, the bottom fluctuation of the drill column is0.44, then the compensation effect is 92.4%. When the wave height is 2.8m, the fluctuation of the bottom of the drill column is 0.21, then the compensation effect is 93.9%.

We also compared the difference between position feedback, the results are shown in Figure 17-18. Which finally resulted in the best compensation effect of 84.3% for velocity feedback.

Simulation results and analysis.

As shown in Fig. 12, by comparing the influence of different mud densities, it is concluded that mud densities have little influence on bottomhole pressure fluctuation. As shown in Fig. 13-16, the smaller the straight passage of the cylinder, the smaller the stiffness of the wire rope, the smaller the vibration amplitude of the drill string, and the smaller the fluctuation of bottomhole pressure.

5. Conclusion

After comparative analysis, it is concluded that the design scheme of lifting and compensation composite system is reasonable, which improves the accuracy of control and compensation, and it is concluded by comparison that the stiffness of wire rope, liquid cylinder diameter and sea condition environment have greater influence on the compensation effect, while the feedback mode and mud density (with or without spacer operation) have less influence on the compensation effect.

Comparing displacement feedback, velocity feedback, and acceleration feedback at a wave height of 6.5m yields a higher accuracy of 84.3% for velocity feedback and a relatively low accuracy of 81.7% for acceleration feedback In the case of different wave heights, the comparison wave heights are 6.5m, 4.8m and 2.8m respectively: as the wave height decreases, the frequency of drilling column vibration will also decrease and the compensation accuracy will become higher, so we should try to choose the operation when the sea condition environment is better.

The smaller the stiffness of the wire rope, the better the compensation effect of the drilling column, the higher the compensation accuracy, but the stiffness is too small wire rope is easy to break, so the selected stiffness should be combined with safety issues.

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