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Research Article

Research on the Dynamics of Geological Drilling Rig against Drill Pipe Impact

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With the ever-increasing demand for resources, the task of geological surveys has increased rapidly, and automated geological drilling rigs that can improve the efficiency of geological surveys have become the mainstream research direction. Automated geological drilling rigs can improve construction efficiency, reduce labor intensity, and effectively reduce construction accidents. During the construction of geological survey operations, accidents such as drill breakage, stuck drill, and equipment damage, which are easily caused by drill pipe impact, occur from time to time. Therefore, drill pipe impact dynamics is currently a hot topic in academic research, but there are few studies that combine automated geological drilling rigs with drill pipe impact. This article starts with the research on the hydraulic system of the power head of the automatic drilling rig and studies the dynamics of the damage of the power head of the drilling rig caused by the impact of the drill pipe by means of hydraulic valve torque limit and closed-loop control of the speed. The oil valve group reduces the reverse torsional impact of the drill pipe on the hydraulic motor. At the same time, the requirements for the selection and pressure setting of the relief valve in the buffer filling oil valve group are proposed. The natural frequency of the relief valve is not less than 20 Hz, and the pressure is set to the working pressure 1.25 times. The effects of shock with or without the buffer filling oil valve group and the oil supply line on the power head motor were compared. The research results of this paper can provide theoretical reference and design basis for subsequent development of automated drilling rigs.

1. Introduction

The power head of the drilling rig is the power source to drive the drill pipe, and its ability to resist the impact of the drill pipe is directly related to the safety of equipment and personnel. How to prevent and reduce drill pipe impact, especially the elastic impact of the drill pipe, is a hot research topic of the drill pipe and drill string.

Most of the load of the drill pipe in service is alternating load, which determines that the failure of the drill pipe will be related to fatigue [1]. Pearson [2] found that about 50% of the failure accidents of the drill pipe were caused by fatigue. Statistics in the Persian Gulf region show that there is one fatigue-related failure incident per footage of 1981.2 m accumulated. In the 1980s, Li et al. [3], through statistical analysis of drill pipe failure accidents in major oilfields in northeast, North, and southwest China, believed that 80% of drill pipe failure accidents were related to fatigue. According to statistics by Liu et al. [4] of all kinds of drill tool failure cases in China from 2007 to 2008, 58% of drill tool failure is related to fatigue. Therefore, the effective control of power head torque and rotation speed is an effective means to reduce the alternating load borne by the drill pipe during service.

In this paper, starting from the research on the torque and speed control of the power head of the drilling rig, in-depth research on the safety control of the elastic impact of the drill pipe is conducted. The hydraulic control method is used to effectively control the damage of the power head by the elastic impact of the drill pipe.

2. Closed-Loop Control of Power Head Torque and Speed of Drilling Rigs

2.1. Over Torque Control. The output torque of the drill pipe of the rig has the following equation:

$$T_{\text{pipe}} = i_{\text{sum}} \times T_{\text{motor}}, \quad (1)$$

where T_{pipe} is the drill pipe torque, $N \cdot m$; i_{sum} is the power head reducer total transmission ratio; and T_{motor} is the power head hydraulic motor output torque, $N \cdot m$.

The torque of the power head hydraulic motor has the following equation:

$$T_{\text{motor}} = \frac{V_g \times_{\Delta} p \times \eta_{\text{mh}}}{20 \times \pi}, \quad (2)$$

where V_g is the hydraulic motor displacement, ml/r; Δp is the pressure difference between two inlet of the hydraulic motor, MPa; and η_{mh} is the hydraulic motor machinery-hydraulic efficiency.

Therefore, from the above two equations, the following equation can be obtained:

$$T_{\text{pipe}} = i_{\text{sum}} \times T_{\text{motor}} = i_{\text{sum}} \times \frac{V_g \times_{\Delta} p \times \eta_{\text{mh}}}{20 \times \pi}. \quad (3)$$

According to the above equation, the torque applied on the drill pipe can be controlled by $V_{g},_{\Delta} p, \eta_{\text{mh}}$, and i_{sum} controlling the torque. η_{mh} belongs to system inherent attribute, not easy to control. Actual control mostly uses $V_{g},_{\Delta} p$, and i_{sum} combined transformations to output drill pipe torque. However, in order to avoid excessive load (such as geological stratigraphic changes and so on) from causing excessive torque damage to the drill pipe, the conversion time can only be shortened as soon as possible. In the comparison of $V_{g},_{\Delta} p$, and i_{sum}, the change rate of Δp is better than the other two, which makes it easier to control remotely and automatically.

By controlling LS (load-sensing) feedback pressure and multiway valve power, head motor section safety valve pressure and hydraulic motor buffer valve pressure can, respectively, limit the output pressure of the hydraulic pump and multiway valve power head motor oil supply pressure and hydraulic motor internal pressure and then control the pressure (Δp) of the hydraulic motor.

2.2. Model of Four-Way Valve Controlled Hydraulic Motor. As shown in Figure 1, the simplified equation (4) of total output of the valve controlled hydraulic motor when hydraulic slide valve spool displacement X_v and external load torque T_L are input at the same time.

$$\theta_m = \frac{\left(K_q/D_m\right)X_V - \left(K_{\text{ce}}/D_m^2\right)\left(1 + \left(V_t/4\beta_e K_{\text{ce}}\right)s\right)T_L}{s\left(\left(s^2/\omega_h^2\right) + \left(2\xi_h/\omega_h\right)s + 1\right),} \quad (4)$$

where θ_m is the angular displacement of hydraulic motor output shaft; D_m is the displacement of the hydraulic motor; V_t is the total volume of the two chambers of the hydraulic motor and the connecting pipe; K_q is the flow gain, $K_q = \partial q_L/\partial x_v$; K_{ce} is the total flow rate-pressure coefficient; Be is the volume modulus of working medium; and T_L is any external load moment acting on the motor shaft.

Natural frequency of hydraulic power components is as follows:

$$\omega_h = \sqrt{\frac{4\beta_e D_m^2}{V_t J_t}}, \quad (5)$$

where J_t denotes the total inertia of the hydraulic motor and load converted to the hydraulic motor shaft.

Damping ratio of hydraulic power components is as follows:

$$\xi_h = \frac{K_{\text{ce}}}{_mD}\sqrt{\frac{\beta_e J_t}{V_t}}, \quad (6)$$

and the transfer function of the angular displacement of the hydraulic motor shaft to the spool is

$$\frac{\theta_m}{X_V} = \frac{\left(K_q/D_m\right)}{s\left(\left(s^2/\omega_h^2\right) + 2\xi_h/\omega_h s + 1\right)}. \quad (7)$$

The transfer function of the angular displacement of the hydraulic motor shaft to the external load torque is

$$\frac{\theta_m}{T_L} = \frac{-\left(K_{\text{ce}}/D_m^2\right)\left(1 + \left(V_t/4\beta_e K_{\text{ce}}\right)s\right)}{s\left(\left(s^2/\omega_h^2\right) + \left(2\xi_h/\omega_h\right)s + 1\right).} \quad (8)$$

It can be seen from equations (7) and (8) that the change of gain constant K_q/D_m is almost caused by the change of flow gain K_q. The flow gain K_q is the largest when there is no load, and its value will decrease when the load is applied [5]. If the general design rule that the maximum load is $p_L = 2/3\,p_s$ is used, then the gain of the valve will be reduced to 57.7% of the no-load value, and the reduction of gain is detrimental to the stability of the system. When the motor is stationary, the static load torque will cause motor leakage and cause the shaft to continuously rotate. $K_{ce}/D_M{}^2$ should be reduced as much as possible to make the valve control hydraulic motor have higher rigidity.

2.3. Torque and Speed Control Simulation. According to the hydraulic principle of the geological drilling rig, use AMESim software to establish the simulation model of the power head as shown in Figure 2, and set the simulation boundary conditions according to Table 1. The simulation model consists of load-sensing pump, load-sensing multiway valve power head valve (electric proportional valve), load feedback proportional pressure limiting valve (used to limit LS feedback pressure), power head and buffer filling oil valve group, and bottom hole load model and closed-loop control model.

The simulation simulates the sudden increase in drill pipe torque under external load, protects the drill pipe by limiting the output torque under pressure ($_{\Delta} p$), and carries out closed-loop control on the output speed of power head

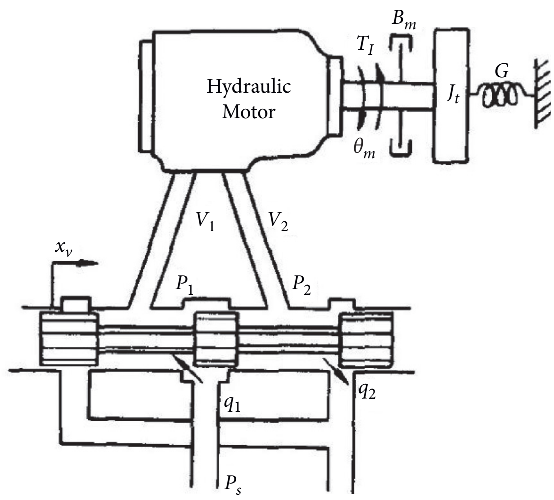

FIGURE 1: Model of the valve controlled hydraulic motor [5].

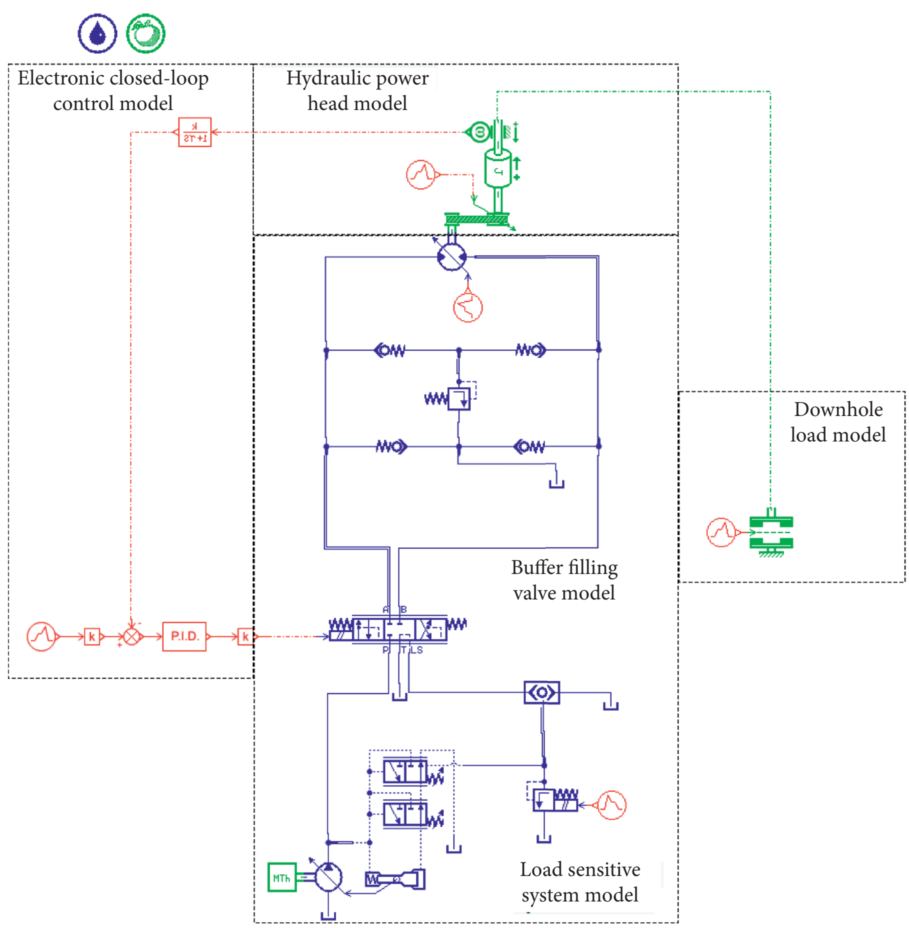

FIGURE 2: Simulation model of power head of the geological drilling rig.

to ensure that the drill pipe speed can be adjusted and controlled.

It can be seen from Figure 3 that the closed-loop control of power head speed achieves the desired goal, and the speed control is stable. At 2 s, the load torque suddenly increases to 5000 Nm; at this time, the speed of the power head with the overtorque control valve drops sharply to stop, and the speed of the power head without the overtorque control valve is not affected, which will inevitably affect the life of the drill pipe. It even caused a broken drill and stuck drill accident.

PID parameters are as follows: $K_P = 500$; $K_I = 1$; and $K_D = 2$.

Table 1: Simulation boundary parameters of the hydraulic system of the geological drilling rig.

No.	Parameter name/unit	The set value	NO.	Parameter name/unit	The set value
1	Maximum pressure of the main pump (MPa)	35	12	Maximum displacement of the power head motor (mL/r) [7]	80.4
2	Engine speed (r/min)	2400	13	Minimum displacement of the power head motor (mL/r)	40
3	Auxiliary pump displacement (mL/r) [6]	32	14	Pressure difference between inlet and outlet of the power head motor (MPa)	24
4	Main pump displacement (mL/r)	74	15	Power head motor 1 stage reduction ratio I_0	2.5
5	Volumetric efficiency of the main pump [6]	0.92	16	Power head 1 gear reduction ratio i_1	15.625
6	Main pump mechanical efficiency [6]	0.95	17	Power head 2 gear reduction ratio i_2	7.8125
7	Volumetric efficiency of the auxiliary pump	0.8	18	Power first 3 gear reduction ratio i_3	4.3925
8	Auxiliary pump mechanical efficiency	0.95	19	Power first 4 gear reduction ratio i_4	2.5
9	Feed cylinder diameter (mm)	90	20	Power head motor volume efficiency [7]	0.95
10	Feed cylinder rod diameter (mm)	70	21	Power head motor mechanical efficiency [7]	0.98
11	Feed cylinder stroke (mm)	3500	22	Mechanical efficiency of power head reducer [8]	0.98

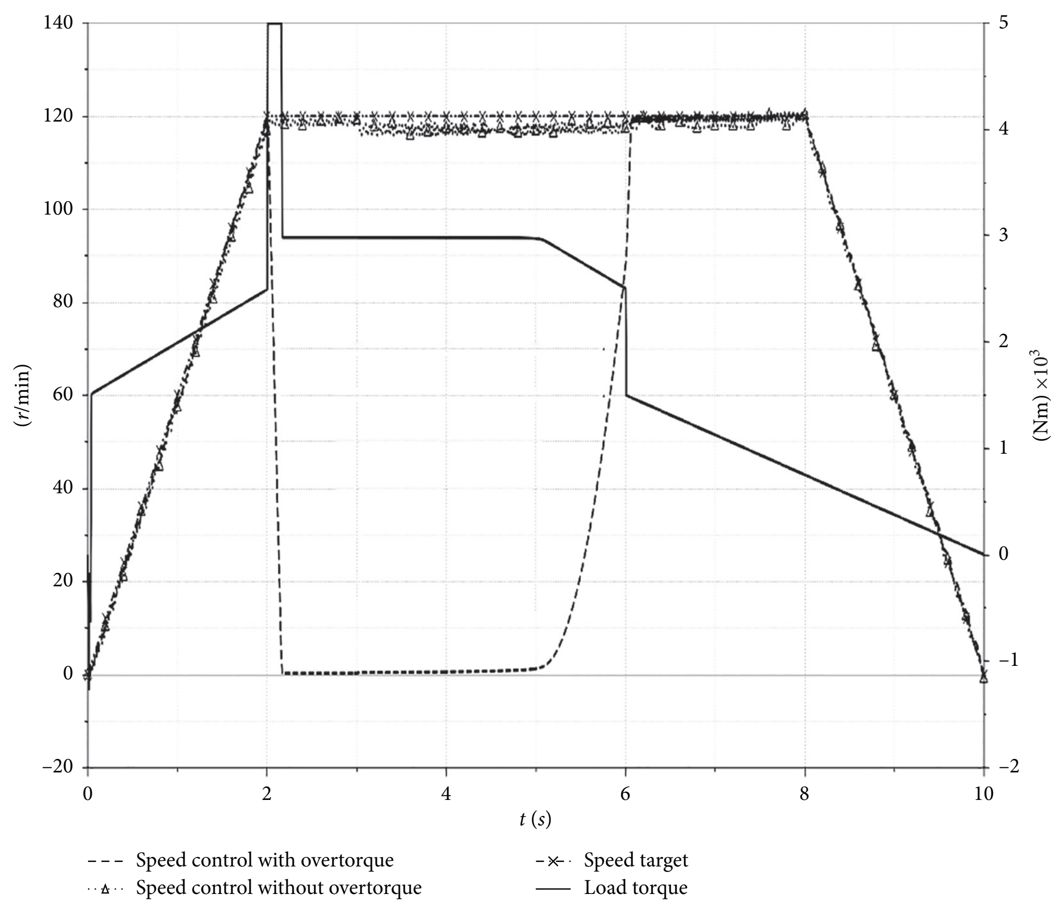

Figure 3: Torque-speed curve of power head output.

3. Safety Control of Elastic Impact of Drill Pipe

In the process of drilling, when the power head stops drilling, the downhole drill pipe will give a reverse torque to the power head. When the bottom hole drill bit is cutting hard rock, the hard rock breaks instantly and the load torque is suddenly released. These two working conditions must be paid attention to in the process of drilling. The elastic impact of the drill pipe is affected by the elastic modulus of the drill pipe, the diameter of the drill pipe, the distance from the power head to the drill bit, the overall bending state of the drill rod, and the size of the system movement pair clearance various influences. With the increase in drilling depth, the elastic impact of the drill pipe cannot be ignored [9].

Therefore, the hydraulic system of the power head of the drilling rig must be equipped with a safety device that resists the elastic impact of the drill pipe to ensure the safety of the power head motor and the drill pipe.

3.1. Elastic Model of Drill Pipe. The dynamic starting stage of mechanical impact process can be divided into three stages: clearance elimination, elastic component deformation, and load starting movement [10]. The drill pipe is treated as an elastic component, and the output end of the power head is treated as a load. The dynamic model is established, as shown in Figure 4. The influence of drill pipe elastic impact on the hydraulic motor of power head should be included in load starting movement.

The motion equation of this process is

$$\begin{cases} J_1\ddot{\theta}_1 + k_1(\theta_1 - \theta_2) = M_1, \\ J_2\ddot{\theta}_2 - k_1(\theta_1 - \theta_2) = M_2, \end{cases} \tag{9}$$

where J_1 is the moment of inertia of the drill bit; J_2 is the moment of inertia of power head; K_1 is the elastic coefficient of the drill pipe; θ_1 is the angular displacement of drill bit; θ_2 is the angular displacement of the output shaft of the power head; M_1 is the equivalent torque of drill bit; and M_2 is the equivalent torque of power head.

Make the relative angular displacement

$$\Delta\theta = \theta_1 - \theta_2. \tag{10}$$

Equation (9) is transformed by calculation and substituted into equation (10) to obtain the relative angular motion equation as follows:

$$\Delta\ddot{\theta} + \omega_n^2\Delta\theta = \frac{M_1(t)}{J_1} - \frac{M_2(t)}{J_2}, \tag{11}$$

where natural frequency is as follows:

$$\omega_n = \sqrt{\frac{k_1(J_1 + J_2)}{J_1 J_2}}. \tag{12}$$

Solution response is as follows:

$$\Delta\theta = -\frac{(M_1 - M_2)J_2}{k(J_1 + J_2)}\cos\omega_n t + \frac{\omega_1}{\omega_n}\sin\omega_n t + \frac{M_1J_2 + M_2J_1}{k(J_1 + J_2)}. \tag{13}$$

Among them, the angular velocity is as follows:

$$\omega_1 = \sqrt{\frac{2M_1\delta}{J_1} + \frac{(2M_1 - M_2)M_2}{k_1}}, \tag{14}$$

where Δ denotes system clearance and $\delta = M_1 t_\delta^2/2J_1$, where t_δ is the time taken by the system to overcome the clearance.

When $M_2 = 0$, the maximum internal torque of the drill pipe is as follows:

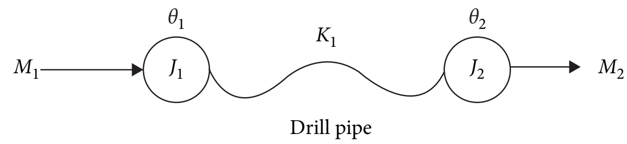

Figure 4: Dynamic model of elastic deformation of the drill pipe [10].

$$M_{12\,\max} = \frac{M_1J_2}{J_1 + J_2}\left[1 + \sqrt{1 + \frac{(J_1 + J_2)k_1t_\delta^2}{J_1J_2}}\right]. \tag{15}$$

When there is no the clearance in the system, $\delta = 0$, we get

$$M_{12\,\max} = \frac{M_1J_2 + M_2J_1}{J_1 + J_2}\left[1 + \sqrt{1 - \frac{J_1(J_1 + J_2)M_2^2}{(M_1J_2 + M_2J_1)^2}}\right], \tag{16}$$

$$M_{12} = \frac{M_1J_2}{J_1 + J_2}. \tag{17}$$

According to equations (15) and (17), the calculated dynamic load factor is as follows:

$$\varphi = \frac{M_{12\,\max}}{M_{12}\left[1 + \sqrt{1 + \left((J_1 + J_2)k_1t_\delta^2/J_1J_2\right)}\right]}. \tag{18}$$

It can be seen from equation (18) that the elastic impact torque of the drill pipe is at least twice the static torque, so this relationship must be considered when setting the impact torque in simulation.

3.2. The Role of Buffer Filling Oil Valve Group. According to the parameters in Table 1, the boundary conditions of the simulation model in Figure 2 are set. The power head stops rotating at 5 s, and the drill pipe impact reverse torque occurs at 5.1 s. The maximum output torque of a geological drilling rig is 4000 Nm (drill pipe side). According to equation (18), the elastic impact torque is at least twice the static torque [11]. According to the comprehensive working conditions of the drilling rig, 1.45 times of the maximum output torque is taken as the boundary condition of impact torque; that is, the impact torque is set at 5800 Nm, and the duration is 0.1 s.

Figure 5 shows that the maximum pressure of the hydraulic motor with the buffer filling oil valve group is 33.6 MPa, and the maximum torque is −429.2 Nm. The maximum pressure of the hydraulic motor without the buffer filling oil valve group is 42.8 MPa, and the maximum torque is −545.9 Nm.

According to the performance parameters of the hydraulic motor shown in Figure 6, the hydraulic motor with

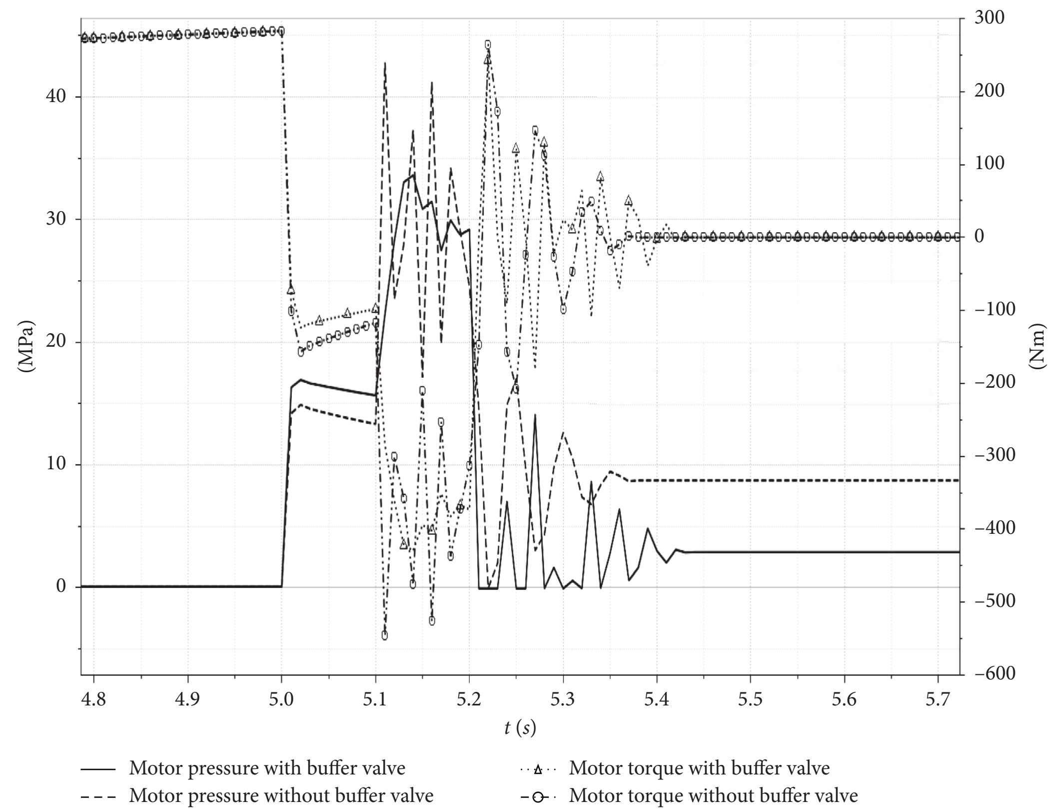

Figure 5: Motor pressure-torque comparison curve with or without buffer filling oil valve set.

Table of values (theoretical values, without considering η_{mh} and η_v; values rounded)

Size			**28**	**32**	**45**	**56**	**63**	**80**	**90**	**107**	**125**	**160**	**180**	**250**	**355**
Displacement	V_g	cm^3	28,1	32,0	45,6	56,1	63,0	80,4	90,0	106,7	125,0	160,4	180,0	250	355
Max. speed	n_{max}	rpm	6300	6300	5600	5000	5000	4500	4500	4000	4000	3600	3600	2500	2240
	$n_{max\ interm.}$ 1)	rpm	6900	6900	6200	5500	5500	5000	5000	4400	4400	4000	4000	–	–
Max. flow	$q_{V\,max}$	L/min	176	201	255	280	315	360	405	427	500	577	648	625	795
Torque constants	T_K	Nm/bar	0,445	0,509	0,725	0,89	1,0	1,27	1,43	1,70	1,99	2,54	2,86	3,98	5,64
Torque, Δp = 400 bar	T	Nm	178	204	290	356	400	508	572	680	796	1016	1144	1393 2)	1976 2)
Case volume		L	0,20	0,20	0,33	0,45	0,45	0,55	0,55	0,8	0,8	1,1	1,1	2,5	3,5
Moment of inertia	J	kgm^2	0,0012	0,0012	0,0024	0,0042	0,0042	0,0072	0,0072	0,0116	0,0116	0,0220	0,0220	0,061	0,102
Weight (approx.)	m	kg	10,5	10,5	15	18	19	23	25	34	36	47	48	82	110

Figure 6: Performance parameters of a hydraulic motor [8].

size 80 is allowed to have a maximum two-cavity pressure of 40 MPa and a maximum torque of 508 Nm. The hydraulic motor without the buffer filling oil valve group has already been in the condition of overtorque and overpressure, which will cause premature damage of the hydraulic motor.

It can be seen from equation (3) that when the displacement of the hydraulic motor remains unchanged, the output torque of the hydraulic motor can be limited by controlling the motor pressure. Therefore, the installation of a buffer filling oil valve group in the motor of the power head can effectively avoid the damage of the hydraulic motor caused by the reverse torsional impact of the drill pipe during ultra deep drilling.

3.3. Parameter Comparison of Buffer Filling Oil Valve Group

3.3.1. Different Settings of Pressure. Different pressure settings of the relief valve in the buffer filling oil valve group have different response to reverse torsional impact. The simulation settings are as follows: the pressure settings are the actual working pressure (28 MPa), 1.25 times the working pressure (35 MPa), and the maximum working pressure (40 MPa) of the hydraulic motor. The natural frequency of the relief valve is 20 Hz. The power head stopped rotating at 5 s, and the drill pipe impact reverse torque occurs at 5.1 s. The impact torque was 5800 Nm, and the duration was 0.1 s.

It can be seen from Figure 7 that the setting pressure of 40 MPa is overshoot, so the setting pressure is not conducive to the safety of the hydraulic motor [12]. The setting pressure of 28 MPa, which is the same as the actual working pressure, is overshoot and oscillation, which does not exceed the safety pressure of the motor (40 MPa). However, this setting pressure will inevitably affect the normal operation of the power head of the drilling rig, resulting in the reduction and instability of the drill pipe torque output. Therefore, it is more appropriate to set buffer filling oil valve group pressure as 1.25 times of the actual working pressure.

3.3.2. Different Natural Frequencies. Equation of mechanical natural frequency of the relief valve is as follows [13]:

$$\omega_m = \sqrt{\frac{K_e}{m_v}}. \tag{19}$$

In equation (19), $K_e = K_s + 0.43W(p_s - p_R)$, the equivalent spring coefficient (sum of mechanical spring coefficient and hydraulic spring coefficient); W is the main valve area gradient; and m_v is the hydraulic spool mass plus one third of spring mass.

According to the mechanical natural frequency of the relief valve, different brands and different series of the relief valve have different size parameters, so the natural frequency is different. The simulation setting is as follows: the natural frequency of the relief valve is 5 Hz, 10 Hz, 15 Hz, 20 Hz, and 25 Hz, the pressure is set as 35 MPa, the power head stops at 5 s, and the drill pipe impact reverse torque occurs at 5.1 s. The impact torque was 5800 Nm, and the duration was 0.1 s.

It can be seen from Figure 8 that the pressure oscillation amplitude and oscillation period of the relief valve with the natural frequency of 5 Hz are the largest, and the maximum pressure has exceeded the safety pressure of the motor. As the natural frequency increases, the pressure oscillation amplitude and oscillation period gradually decrease. The steady pressure values of natural frequency above 15 Hz (including 15 Hz) are close, and the minimum pressure values of 20 Hz and 25 Hz are greater than the actual working pressure (28 MPa). Therefore, when selecting the relief valve, designers should consider the influence of natural frequency of the relief valve on the performance of the buffer filling oil valve group and select the relief valve with natural frequency above 20 Hz.

3.4. Influence of Hydraulic Motor Filling Oil Pipe. The moment of inertia of the power head of the geological drilling rig is not large compared with that of other engineering drilling rigs, so the problem of oil replenishment of the hydraulic motor of the power head is often ignored in the development and design [14]. However, with the continuous improvement of drilling depth requirements, the reverse torque of drill pipe impact on the hydraulic motor increases, and the cavitation phenomenon of the hydraulic motor occurs frequently [15].

Now, simulate whether the hydraulic motor of power head has a filling oil pipeline and compare the pressure change on one side of the hydraulic motor. The specific settings are as follows: the natural frequency of the relief valve is 20 Hz, the pressure is set as 35 MPa, the power head stops at 5 s, and the drill pipe impact reverse torque occurs at the 5.1 s. The impact torque was 5800 Nm, and the duration was 0.1 s, and the opening pressure of the check valve of filling oil is 0.03 MPa, and the filling oil method is the tank normal pressure (atmospheric pressure) replenishment.

It can be seen from Figure 9 that the pressure on one side of the hydraulic motor without filling oil pipeline is close to –0.1 MPa, and the phenomenon of periodic suction occurs [16]. At this time, the air dissolved in the oil is repeatedly separated out and dissolved into [17], which will cause the cavitation of the hydraulic motor. The pressure at one side of the hydraulic motor with filling oil pipeline only appears –0.03 MPa once, and the rest of the time is positive pressure, and there is no motor suction phenomenon. Therefore, adding the filing oil function in the hydraulic motor of power head can effectively avoid the occurrence of the periodic suction phenomenon of the hydraulic motor, reduce the failure of the hydraulic motor [18], and prolong the service life of the hydraulic motor.

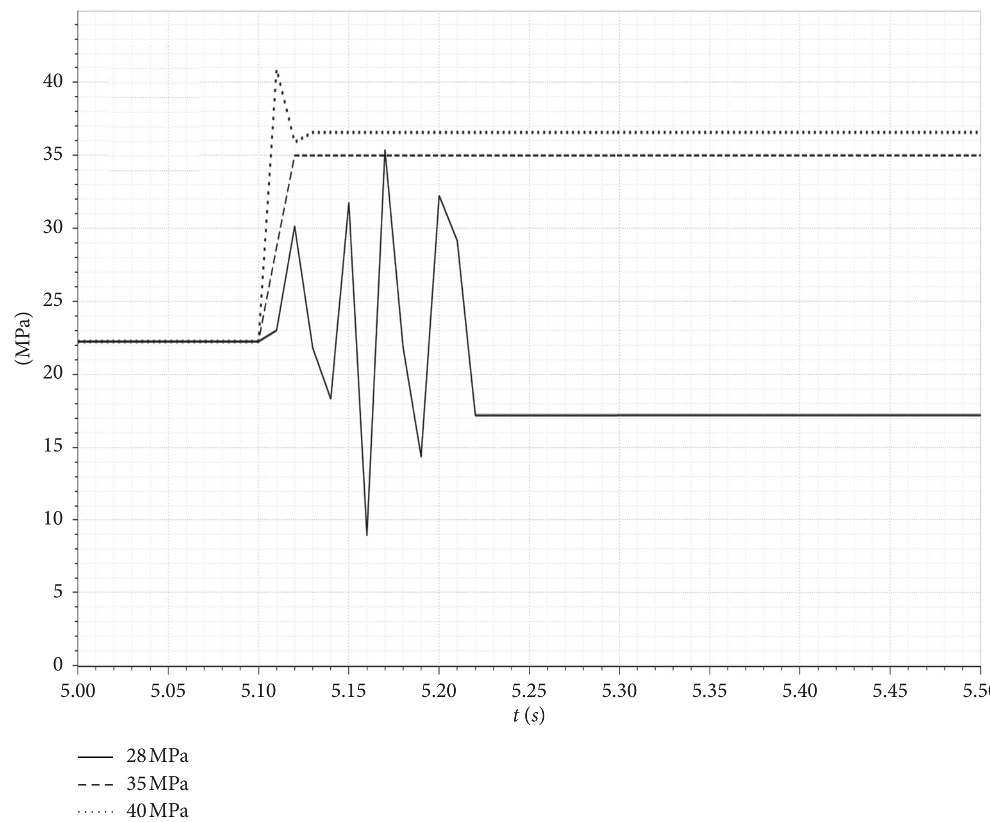

Figure 7: Different setting pressures for the relief valve.

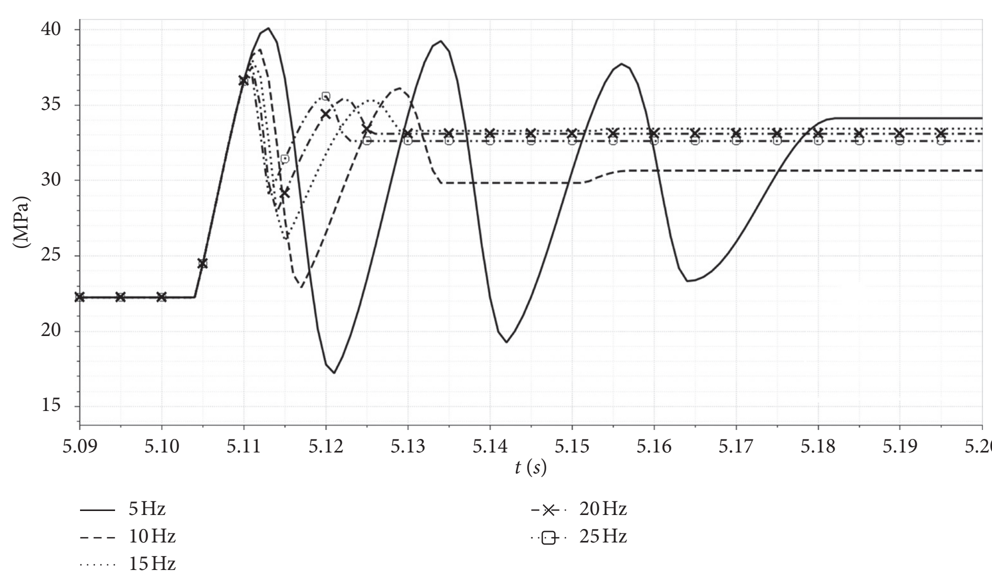

Figure 8: Different natural frequencies of relief valves.

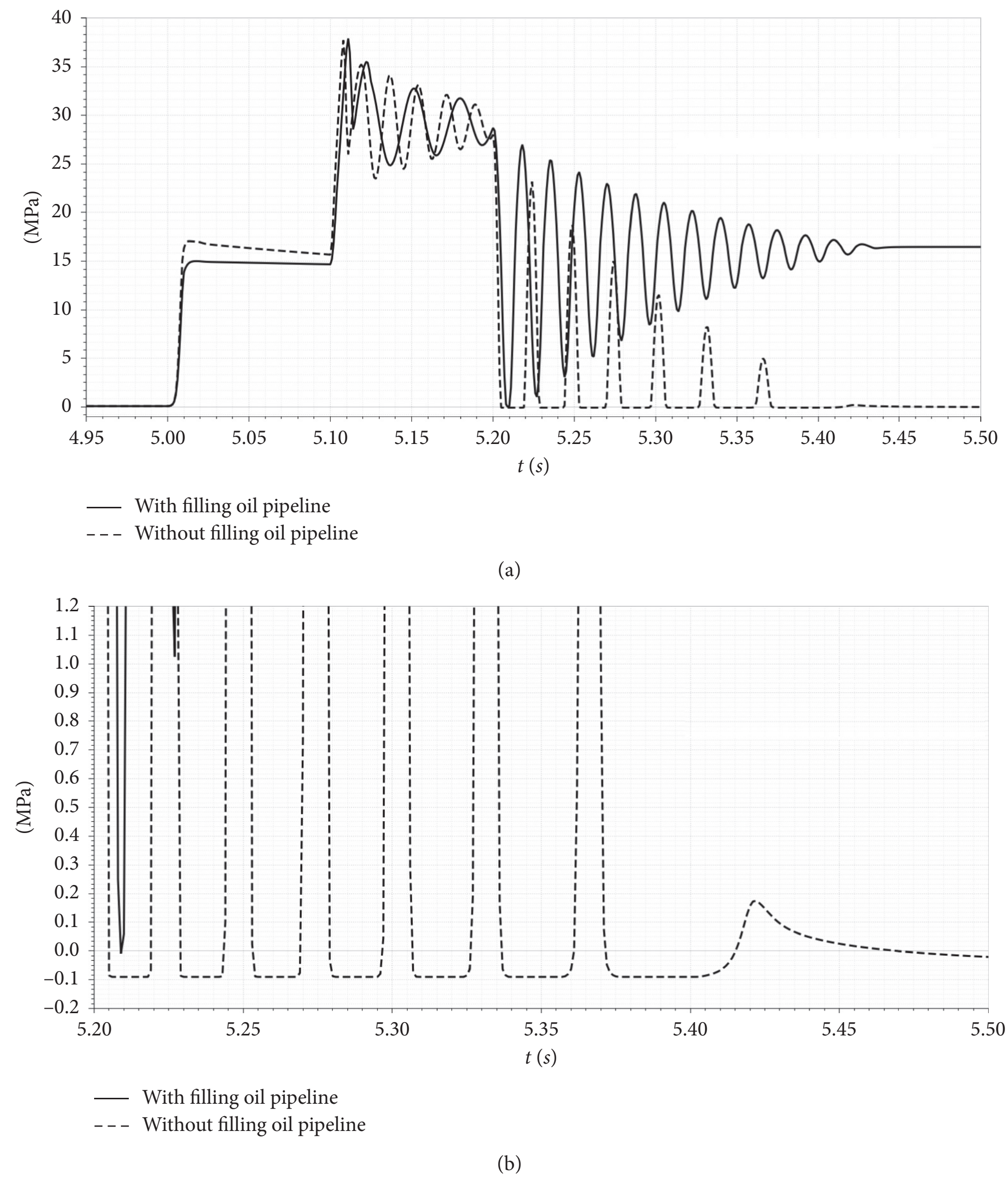

FIGURE 9: Comparison curve of the hydraulic motor of power head with or without filing oil pipeline: (a) original curve; (b) magnification curve.

4. Conclusion

The automatic geological drilling rig is upgraded intelligently on the basis of the original drilling rigs. It is equipped with the electric hydraulic valve and sensor and forms a closed-loop control with the controller. Through the preset controller program, the intelligent control and operation of the working state of the drilling rig can be realized, and different working modes can be switched to adapt to the drilling speed of different geological conditions. At the same time, the manual operation mode is retained, and the state protection of the drilling rig is set through the program so as to avoid drilling accidents caused by the operator's fault and improve the automation level of the geological drilling rig.

In this paper, the control characteristics of the hydraulic system of the automatic geological drilling rig in the process of antidrill pipe impact are studied, and the following conclusions can be drawn:

(1) The hydraulic power head drill rig can limit the torque of the drill pipe by limiting the pressure difference between the two cavities of the hydraulic motor so as to ensure the drill pipe safety in the process of exploration drilling. Using the load-sensing system to limit LS feedback pressure and PID closed-loop control can effectively control drill pipe torque and speed.

(2) The elastic impact model of the drill pipe is established to guide the setting of simulation boundary conditions. The buffer filling oil valve group can effectively deal with the reverse torque impact of the drill pipe and ensure that the hydraulic motor does not have overpressure and overtorque.

(3) The relief valve in the buffer filling oil valve group shall be selected with natural frequency above 20 Hz, and the set pressure shall be 1.25 times of the rated working pressure of the system.

(4) The oil replenishment function can effectively avoid the occurrence of periodic suction of the hydraulic motor, reduce the failure of the hydraulic motor, and extend the service life of the hydraulic motor.

Data Availability

The data in the tables used to support the findings of this study are included within the article. The data in the figures used to support the findings of this study are available from the corresponding author (lzjcumtb@126.com) upon request.

Conflicts of Interest

The authors declare that they have no conflicts of interest.

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