

Dynamic Analysis of Key Parameters of Anti-Drag Pressure Controller

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ABSTRACT

Gravity-induced contact between the drill string and the wellbore during directional drilling of long horizontal wells results in significant weight-on-bit (WOB) loss. To address this issue, this paper builds upon previous research and proposes an anti-sticking controller for directional wells. The working principle of the controller is introduced, and experimental studies are conducted to determine the effect of driving speed on the tool state under different flow rates. A coupled dynamic model of the drill string system under horizontal directional drilling conditions is established to analyze its dynamic characteristics under different parameter conditions. The results indicate that increasing the rotary table speed and pressure drop enhances the angular velocities of the drill bit, bottom hole assembly (BHA), controller, and drill pipe, reaching stability at 30 s. The phase trajectories of the drill bit, BHA, and controller do not form closed limit cycles, and their variations exhibit similar trends. Meanwhile, the drill pipe stabilizes more quickly. In contrast, increasing WOB reduces the angular velocities of the drill bit and other components, prolongs the drill pipe's stabilization time, and prevents stick-slip vibration. Additionally, the study evaluates the effects of different installation positions of the anti-stick-slip controller and finds that placing it in front of the drill collar achieves optimal performance. These findings provide valuable guidance for the installation and application of similar tools.

INTRODUCTION

With the increasingly complex and diverse types

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of oil and gas reservoirs, exploration and development work has entered a complicated stage, which poses new challenges to oil and gas exploitation (Tian et al., 2022). In order to increase mining efficiency, directional well construction technology such as horizontal well, extended reach well and branch well has been widely used (Tian et al., 2023). In horizontal well drilling, the drill string is difficult to provide WOB, and drilling friction can offset some of the drilling friction, making it difficult to transfer the drill string to the bit (Yu et al., 2022), at the same time, the stress of drill pipe during drilling is very complicated (Peng et al., 2022). The friction encountered during drilling is significant, increasing the likelihood of inducing the stick-slip phenomenon within the drill string. This phenomenon manifests as lateral vibration and torsional deformation in the drill string system. To address these issues, significant advancements have been made in the study of drill string dynamics under horizontal well conditions. During horizontal well drilling, providing adequate weight on bit (WOB) poses challenges as drilling friction partially offsets the WOB, impeding its transmission to the drill bit (Tian et al., 2021). This challenge is particularly pronounced during sliding drilling, where the upper drill string remains stationary, allowing static friction between the drill string and the well wall to accumulate. Once this friction reaches a critical point, it results in backing pressure within the drill string. This backing pressure phenomenon accelerates the wear of drilling tools, significantly hampering directional drilling speed (Tuo et al., 2019).

Furthermore, in the drilling process, the coupling between the axial and torsional directions of the drill string provides a more accurate representation of its actual behavior (Idir et al., 2022). The two-degree-of-freedom mass-spring-damping model is commonly employed to derive second-order ordinary differential equations for stability analysis in axial-torsional coupling dynamics during drilling (Tian et al., 2020). Christoforou et al. (2003) assert that axial, transverse, and torsional vibrations of drill strings are mutually dependent, arising from interactions between oil/formations, wellbores, and various geometric and dynamic nonlinearities. Aarsnes et al. (2019) consider the drill-rock

interaction law, establish a distributed axial-torsional drill string model, and analyze the self-excited axial and torsional vibration characteristics induced by the regeneration effect. They observe that the axial loop gain is influenced by drill bit and rock parameters, increasing the stable region of torsional dynamics and mechanical drilling speed under constant drilling pressure but simultaneously exacerbating axial instability. Addressing the coupled vibration phenomenon of the drill string, Mohammad Javad Moharrami et al. (2021) propose an effective finite element modeling approach for stick-slip vibration of the entire drill string. They account for the nonlinear effects of the drill string and ensemble nonlinear axial-torsional coupling, constructing a five-degree-of-freedom comprehensive parameter model. The model's performance is validated by comparing stick-slip results with field experiments.

Of course, researchers are also working on tools. The hydraulic oscillator proves to be an effective solution for mitigating excessive drag during sliding drilling (Liu et al., 2017). Its drag reduction mechanism relies on the periodic axial vibration induced by variations in flow area between the moving valve plate and the static valve plate. This variation alters the frictional state between the drill string and the wellbore by causing axial displacement of the drill string (Tian et al., 2016). Expanding upon the traditional hydraulic oscillator, Mu et al. (2022) devised a hydraulic oscillator incorporating an additional balanced piston. This design enhancement not only enhances tool longevity but also ensures the efficacy of vibration. In a separate study, Tian et al. (2023) investigated dual-piston hydraulic oscillators and determined that increasing the input flow rate could elevate the axial vibration frequency, thereby enhancing the anti-friction effect.

More recent research has further expanded on drill string dynamics and vibration suppression under horizontal well conditions. For instance, Guo et al. (2025) developed a nonlinear finite-element reduced-order model and implemented model predictive control (MPC) to actively mitigate stick-slip vibration while accounting for WOB variation. Similarly, Wang et al. (2023) systematically analyzed the effects of surface torque, WOB, and rotary speed on stick-slip and proposed optimization strategies to suppress torsional oscillation. He et al. (2025) conducted experimental and simulation studies of different drill pipe-formation combinations and revealed how hydraulic oscillators affect drag reduction efficiency. Additionally, Ma et al. (2025) introduced a semi-analytical friction drag prediction model validated by field data from directional wells, offering a simplified but robust means for real-time estimation. These recent works contribute to a deeper understanding of coupled vibrations, frictional resistance, and downhole control strategies under evolving horizontal well conditions.

The existing theory of dynamic forces in horizontal wells lays the groundwork for preventing backing pressure and minimizing drill string stick-slip. In recent years, innovative designs and tool combinations have emerged to address these challenges. Taking drilling tools as an example, a range of tools aimed at reducing friction and drag, as well as enhancing speed and efficiency, have been developed and implemented successfully (Tian et al., 2022). Simultaneously, given the prevalence of severe stick-slip phenomena in long-horizontal drill string sections, which increase the occurrence of downhole accidents or risks, there has been continuous development of tools aimed at reducing stick-slip or ensuring downhole safety (Tian et al., 2018). Regarding drill bits, a variety of new designs have been introduced, including those equipped with dental wheels and PDC compounds, optimized or hybridized versions of traditional drill bits, and drill bits featuring composite impact effects. These innovations have seen widespread adoption and application (Saksala et al., 2018).

The distinctive feature of the ADPC lies in its ability to enable both composite drilling and directional drilling. During directional drilling, the drill string rotates while the BHA slides. The ADPC can improve directional drilling efficiency, significantly reduce costs, and exhibit high safety with low accident handling expenses. This paper presents the structural features and working mechanism of an Anti-drag pressure controller. A dynamic model is developed to analyze the key parameters of the Anti-drag pressure controller, including the rotary speed, WOB, and pressure drop across the rotor ends. The variations in rotary speed and pressure drop at the rotor ends affect the driving torque of the Anti-drag pressure controller, while changes in WOB result in an inverse torque in the drill string system. By analyzing the dynamic results of each module when keeping other parameters constant, a comparative analysis is conducted to investigate the impact of key parameter variations on the dynamics of the Anti-drag pressure controller. The research findings provide valuable insights for practical applications of Anti-drag pressure controllers.

WORKING MECHANISM AND EXPERIMENTAL TEST OF ADPC

Working mechanism

The structure of is shown in Fig. 1. When the upper drill string rotates at a certain speed, the rotor rotates along with the drive shaft, drawing a specific volume of drilling fluid into the cavity between the rotor and stator, while the remaining fluid flows through the central channel of the anti-sticking controller's rotor. During rotation, the high- and low-pressure chambers continuously alternate. When

the upper rotation speed reaches a certain threshold, the stator and housing are driven to rotate by the upper drive under hydraulic force, enabling full rotation drilling of the lower BHA driven by the upper section. When the upper speed falls below the threshold, the hydraulic force is insufficient to drive the stator and housing, causing the lower BHA to operate in sliding drilling mode. The differential pressure controller regulates the pressure drop across

the rotor ends through differential nozzles, thereby controlling the driving torque. The drilling fluid inside the anti-sticking controller converges at the lower connector and flows entirely into the lower drilling assembly, resulting in no fluid loss inside the tool. The rotary sealing device maintains the pressure differential between the mandrel cavity fluid and the external fluid, while allowing the upper and lower mandrels to rotate independently.

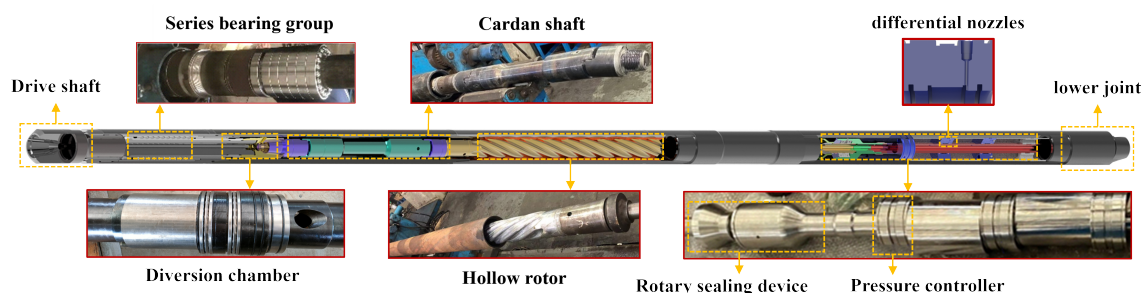


Fig. 1 Schematic diagram of ADPC structure

Experimental test of ADPC

The assembled test tool is installed on the ground experimental test bench, with the upper end connected to the drill string as the fluid inlet, simulating real downhole drilling conditions. The lower end is connected to the power drill and drill bit as the fluid outlet. During the experiment, the Anti-drag pressure controller is horizontally mounted on the bench. In this experiment, clear water is used instead of circulating fluid. A water pump pumps the clear water from the water tank into the inlet of the Anti-drag pressure controller at a certain pressure and flow rate. The water then flows out through the outlet and returns to the water tank via the return pipeline. The specific installation process is shown in Fig. 2.

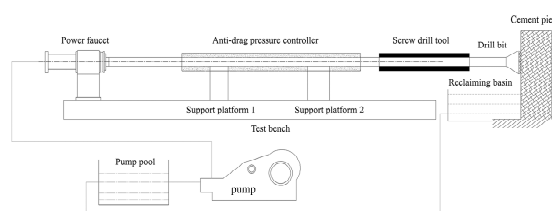


Fig. 2 Schematic of the experimental test

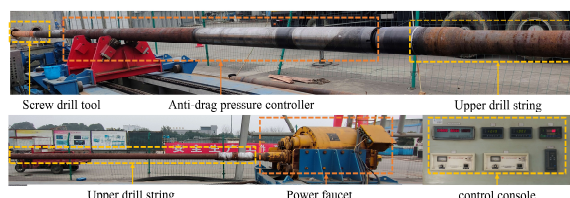


Fig. 3 Ground experimental test

As shown in Fig. 3, the ADPC experimental testing is conducted using a horizontal test bench for downhole tools, and the testing drill string is

connected according to the experimental plan. The front end of the power swivel consists of three drill pipes, ADPC, an adapter and a PDC drill bit. During the test, the power swivel rotates and drives the test drill string to rotate. The parameters of the experimental equipment are shown in Table. 1.

Table. 1 Experimental equipment parameter

Parameter name	Numerical value	Parameter name	Numerical value
Maximum static load (kN)	1350	Power (kW)	969
Rated speed (rpm)	150	Pumping ($biunt / min$)	120
Rated torque ($N \cdot m$)	20000	Stroke (mm)	304.8
Rated power (kW)	370	Maximum bore (mm)	180
Maximum speed (rpm)	250	Stress (Mpa)	18.7
Working speed (rpm)	10–60	Displacement (L / s)	46.54

During the experiment, the flow rate was set to 18 L/s, and the top drive speed was gradually increased from 5 rpm to 15 rpm. A tachometer was used to monitor the shell's rotational behavior. At 13 rpm, the shell remained stationary; further increasing the speed led to forward rotation. Similarly, at a flow rate of 23 L/s, the shell continued to rotate in reverse at 13 rpm and reached a static state at 18 rpm. Further increasing the speed induced forward rotation. Following the same procedure, the equilibrium (static) speeds for flow rates of 28 L/s and 33 L/s were determined. Fig. 4 shows the distribution of the ADPC rotational state with respect to the top drive speed (5-15rpm, 13-20rpm, 18-30rpm, 22-40rpm) under different flow rates (18 L/s, 23 L/s, 28 L/s, and 33 L/s). The horizontal axis represents the upper drive speed (rpm), while the vertical axis classifies the housing rotation state into reverse rotation, static state, and forward rotation. The results indicate that as the flow rate increases, the threshold speed required to stabilize the housing also increases. This is because a higher flow rate generates greater counter torque on the lower BHA, requiring a larger

driving torque to maintain balance, thus necessitating a higher upper drive speed.

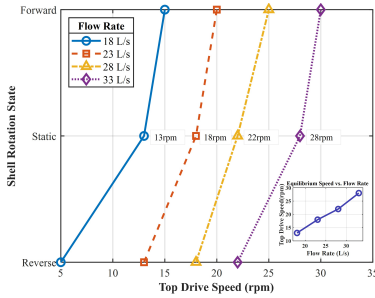


Fig. 4 Effect of Top Drive Speed on Shell Rotation at Various Flow Rates

DRILL STRING DYNAMIC MODELS WITH ADPC

In the directional drilling process, the drilling process is complicated, so it is important to conduct dynamic analysis of the drilling system containing the ADPC.

PDM drill are an important component of the underwater drill combination, and the driving torque required for drill rocks is mainly provided by the working torque produced by the PDM drill. The calculation formula is as follows:

$$T_{PDM} = \frac{1}{2\pi} \Delta p Q \eta_m \eta_v \quad (1)$$

Where, Δp is the pressure drop at both ends of the PDM drill during operation, Q is volume; η_m is mechanical efficiency; η_v is volumetric efficiency.

Comprehensive friction coefficient of PDM, wellbore and drill string pressure characteristics, curved angle torque calculation formula is:

$$T_f = u_1 W_{ob} \sin \alpha \cdot R_{PDM} \quad (2)$$

Where, T_f is corner torque; u_1 is friction coefficient between bending PDM and wellbore; W_{ob} is static bit weight; is drill tool angle; R_{PDM} is tool radius.

Separate the drill pillar system into n concentrated mass blocks, assuming that any drill mass block sliding relative to the well wall is rotated at Ω_i , And the corresponding velocities after collision are V_t , V_n , V^* represents threshold. When $\sqrt{V_t^2 + V_n^2} - V^* \leq 0$, it is assumed that the drilling column is in continuous contact with the well wall. When $\sqrt{V_t^2 + V_n^2} - V^* \geq 0$, the contact state is the collision contact. When the drilling column is in constant continuous contact with the well wall, without taking into account the drill fluid floating force, it is:

$$\begin{cases} F_{Ni} = m_i g \\ T_{fi} = \mu_i F_{Ni} R_i \end{cases} \quad (3)$$

Where, m_i is the quality of the first pillar node; μ_i is Friction coefficient of drilling column nodes and well walls; F_{Ni} is drilling column contact pressure; T_{fi} is the friction torque of the drill pillar nodes; R_i is the radius of the nodes of the pillar.

When the drill column is in contact with the well wall for collision [23], the crimping torque of the drilling column node is:

$$T_{fi} = \mu_i \frac{m_i (1-k) \sqrt{V_i'^2 + \Omega_i'^2}}{\Delta t} R_i \quad (4)$$

In addition, the drilling head also collides and rubs the rock. This paper uses the appropriate Karnopp friction model for engineering practice. This model defines a tiny boundary layer $D_v = 1 \times 10^{-6} \text{ rad/s}$. When the speed of the drill head is in this boundary layer, the drilling head remains stationary, for the static friction torque. When the driven drill torque is greater than that of the static friction, and when the angle of the Drill Head is larger than that border layer it moves for the active friction. The specific expression of the Karnopp model is:

$$T_{fb}(\dot{\phi}_b) = \begin{cases} T_{eb}(\dot{\phi}_b), & \dot{\phi}_b < D_v \text{ and } |T_{eb}| < T_{sb} \\ T_{sb} \text{sgn}(T_{eb}(\dot{\phi}_b)), & \dot{\phi}_b < D_v \text{ and } |T_{eb}| \geq T_{sb} \\ T_{cb}(\dot{\phi}_b) \text{sgn}(\dot{\phi}_b), & \dot{\phi}_b \geq D_v \end{cases} \quad (5)$$

Where, T_{fb} is frictional torque caused by drilling fluid; $\dot{\phi}_b$ is the angular velocity of the bit; $\text{sgn}(\dot{\phi}_b)$ is sign judgment function; T_{eb} is the driving torque applied to the bit; T_{sb} is maximum static friction torque between bit and formation; T_{cb} is coulomb friction torque. T_{eb} is the driving torque applied to the bit, T_{sb} is maximum static friction torque, T_{cb} is coulomb friction torque:

$$\begin{cases} T_{eb} = c_{rb}(\dot{\phi}_c - \dot{\phi}_b) + k_{rb}(\phi_c - \phi_b) - c_{r(b+1)} \dot{\phi}_b \\ T_{sb} = R_b W_{ob} \mu_{sb} \\ T_{cb} = R_b W_{ob} \mu_{cb} \end{cases} \quad (6)$$

Where, R_b is bit radius; μ_{sb} is coefficient of static friction; μ_{cb} is coulomb friction coefficient.

The dynamic model of the drill string system with ADPC is established, as shown in Fig. 5. The upper top drive, vertical well, and inclined wellbore are simplified. The viscous damping moment in this section (T_{at}) is considered. The model's upper end is equivalent to the upper drill string (rotary table) with a moment of inertia of J_{at} and a driving torque of T_{ud} . The drill string is discretized into n drill string nodes, with each node having a moment of inertia of J_i ($i = 1, 2, \dots, n$). The equivalent moment of inertia of the bit is J_c , and the equivalent moment of inertia of

the ADPC is J_d . The drill string assembly includes measurement instruments, PDM drill collars, stabilizers, etc, with a moment of inertia of J_B and the driving torque of the PDM is T_{PDM} . Considering the interaction between the drill bit and the formation, the viscous damping moment is represented by T_b .

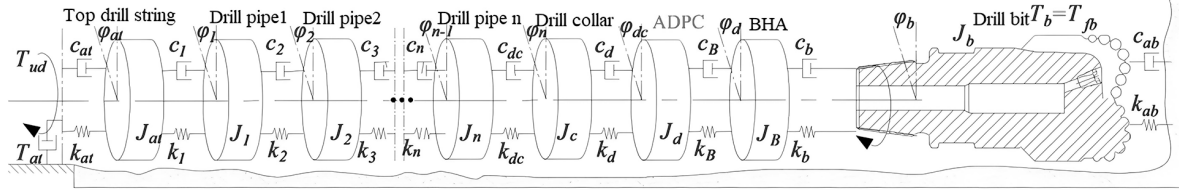


Fig. 5 Horizontal drill string system model

Select any node for stress analysis, as shown in Fig. 6. ϕ_i represents the nodal reversal of the vibration angular shift. The node is deduced by a motion-sub-equation (Tian et al., 2022):

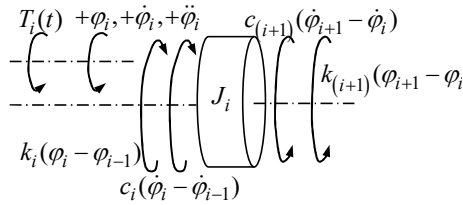


Fig. 6 Single node torsional motion model

$$\begin{cases} J_i \ddot{\phi}_i = -k_i(\phi_i - \phi_{i-1}) + k_{i+1}(\phi_{i+1} - \phi_i) - \\ c_i(\dot{\phi}_i - \dot{\phi}_{i-1}) + c_{i+1}(\dot{\phi}_{i+1} - \dot{\phi}_i) + T_i \\ i = 1, 2, \dots, n, c, d, B, b \end{cases} \quad (7)$$

In combination with the horizontal segment drill system dynamics model, to establish the movement division equation of the drilling system of the Anti-drag pressure controller. In this case, ϕ_i is the rotation angle movement of the nodes i , $\dot{\phi}_i$ is the velocity of the node i , and $\ddot{\phi}_i$ is the acceleration of the angle of the i .

The movement equation of the drill pillar system of the liquid clutch controller is:

$$\begin{cases} J_{at} \ddot{\phi}_{at} = -(k_{at} + k_{p1})\phi_{at} + k_{p1}\phi_{p1} - (c_{at} + c_{p1})\dot{\phi}_{at} + c_{p1}\dot{\phi}_{p1} + T_{ud} \\ J_{p1} \ddot{\phi}_{p1} = -k_{p1}(\phi_{p1} - \phi_{at}) + k_{p2}(\phi_{p2} - \phi_{p1}) - c_{p1}(\dot{\phi}_{p1} - \dot{\phi}_{at}) + \\ c_{p2}(\dot{\phi}_{p2} - \dot{\phi}_{p1}) + T_{f1} \\ J_i \ddot{\phi}_{pi} = -k_{pi}(\phi_{pi} - \phi_{p(i-1)}) + k_{p(i+1)}(\phi_{p(i+1)} - \phi_{pi}) - c_{pi}(\dot{\phi}_{pi} - \dot{\phi}_{p(i-1)}) \\ + c_{p(i+1)}(\dot{\phi}_{p(i+1)} - \dot{\phi}_{pi}) + T_{fi}, i = 1, 2, 3, \dots, n \\ J_c \ddot{\phi}_c = -k_c(\phi_c - \phi_{pn}) + k_{dc}(\phi_d - \phi_c) - c_{ic}(\dot{\phi}_c - \dot{\phi}_{pn}) + \\ c_{id}(\dot{\phi}_d - \dot{\phi}_c) + T_{fc} \\ J_d \ddot{\phi}_d = -(k_{id} + k_{ib})\phi_d + k_{ib}\phi_B - (c_{id} + c_{ib})\dot{\phi}_d + c_{ib}\dot{\phi}_B + T_{fd} \\ J_B \ddot{\phi}_B = -k_{ib}(\phi_B - \phi_d) + k_{ib}(\phi_b - \phi_B) - c_{ib}(\dot{\phi}_B - \dot{\phi}_d) + \\ c_{ib}(\dot{\phi}_b - \dot{\phi}_B) + T_{ib} + T_f \\ J_b \ddot{\phi}_b = -(k_{ib} + k_{iab})\phi_b + k_{ib}\phi_B - (c_{ib} + c_{iab})\dot{\phi}_b + c_{ib}\dot{\phi}_B + T_{PDM} + T_b \end{cases} \quad (8)$$

Simplify the rotary vibrating drill system

Each unit is simplified as a concentrated mass block, and the mass blocks are connected by torsional stiffness K_i and torsional damping C_i springs. The lateral vibration of the drill string, axial vibration at the drill bit, and axial displacement of the top drive rotary table are not considered.

represented in formula (6), which contains a liquid clutch controller, in the form of a matrix:

$$\mathbf{J} \ddot{\boldsymbol{\phi}}(t) + \mathbf{C}_i \dot{\boldsymbol{\phi}}(t) + \mathbf{K}_i \boldsymbol{\phi}(t) = \mathbf{T}(t) \quad (9)$$

In order to facilitate the solving of formula (6) and (7) and their parameters, the equivalent rotational inertia of J_i mass block of a single drill column is as follows:

$$J_i = \rho_i \frac{\pi l_i}{32} (D_i^2 - d_i^2) \quad (10)$$

Where, ρ_i is drill pipe material density; D_i is the outside diameter of the drill pipe; d_i is bore diameter of drill pipe; l_i is length of drill string node.

Torsional stiffness between mass blocks k_{ii} and the torsional damping c_{ii} is:

$$k_{ii} = \frac{\pi G_i (D_i^4 - d_i^4)}{32 l_i}, \quad c_{ii} = \frac{\xi_{ii} l_i}{3} \quad (11)$$

Where, G_i is shear modulus of drill string material; ξ_{ii} is torsional damping coefficient per unit length of drill string.

Similarly, in order to solve the differential equation, it is necessary to define the system state variable, which can be obtained by combining the drill string dynamics model:

$$\begin{aligned} \mathbf{y} &= (\phi_{at}, \dot{\phi}_{at}, \phi_{p1}, \dot{\phi}_{p1}, \dots, \phi_{pn}, \dot{\phi}_{pn}, \phi_c, \\ &\quad \dot{\phi}_c, \phi_d, \dot{\phi}_d, \phi_B, \dot{\phi}_B, \phi_b, \dot{\phi}_b)^T \\ &= (\mathbf{y}_1, \mathbf{y}_2, \mathbf{y}_3, \mathbf{y}_4, \dots, \mathbf{y}_{m-9}, \mathbf{y}_{m-8}, \mathbf{y}_{m-7}, \\ &\quad \mathbf{y}_{m-6}, \mathbf{y}_{m-5}, \mathbf{y}_{m-4}, \mathbf{y}_{m-3}, \mathbf{y}_{m-2}, \mathbf{y}_{m-1}, \mathbf{y}_m)^T \end{aligned} \quad (12)$$

The drilling system consists of the upper drilling column (top drill, rectangle section, sloping section simplified), drilling, liquid clustering controller, drill combination and drill head, the dynamic model layer of the drill system is $m=2n+10$, where m is the equal number.

$$\begin{cases}
 \dot{y}_1 = y_2 \\
 \dot{y}_2 = -\frac{(k_{iat} + k_{ip1})}{J_{at}} y_1 + \frac{k_{ip1}}{J_{at}} y_3 - \frac{(c_{iat} + c_{ip1})}{J_{at}} y_2 + \frac{c_{ip1}}{J_{at}} y_4 + \frac{T_{ud}}{J_{at}} \\
 \dot{y}_3 = y_4 \\
 \dot{y}_4 = -\frac{k_{ip1}}{J_{ip1}} (y_3 - y_1) + \frac{k_{ip2}}{J_{ip1}} (y_5 - y_3) - \frac{c_{ip1}}{J_{ip1}} (y_4 - y_2) + \frac{c_{ip2}}{J_{ip1}} (y_{4-6} - y_4) + \frac{T_{f1}}{J_{ip1}} \\
 \dot{y}_{i+3} = y_{i+4} \\
 \dot{y}_{i+4} = -\frac{k_{ipi}}{J_{ipi}} (y_{i+3} - y_{i+1}) + \frac{k_{ip(i+1)}}{J_{ipi}} (y_{i+5} - y_{i+3}) - \frac{c_{ipi}}{J_{ipi}} (y_{i+4} - y_{i+2}) + \frac{c_{ip(i+1)}}{J_{ipi}} (y_{i+6} - y_{i+4}) + \frac{T_{\beta}}{J_{ipi}} \quad i=2,3,\dots,n-1 \\
 \dot{y}_{m-9} = y_{m-8} \\
 \dot{y}_{m-8} = -\frac{k_{ipm}}{J_{ipm}} (y_{m-9} - y_{m-11}) + \frac{k_{ic}}{J_{ipm}} (y_{m-7} - y_{m-9}) - \frac{c_{ipm}}{J_{ipm}} (y_{m-8} - y_{m-10}) + \frac{c_{ic}}{J_{ipm}} (y_{m-6} - y_{m-8}) + \frac{T_{fn}}{J_{ipm}} \\
 \dot{y}_{m-7} = y_{m-6} \\
 \dot{y}_{m-6} = -\frac{k_{ic}}{J_c} (y_{m-7} - y_{m-9}) + \frac{k_{id}}{J_c} (y_{m-5} - y_{m-7}) - \frac{c_{ic}}{J_c} (y_{m-6} - y_{m-8}) + \frac{c_{id}}{J_c} (y_{m-4} - y_{m-6}) + \frac{T_{fc}}{J_c} \\
 \dot{y}_{m-5} = y_{m-4} \\
 \dot{y}_{m-4} = -\frac{k_{id}}{J_d} (y_{m-5} - y_{m-7}) + \frac{k_{ib}}{J_d} (y_{m-3} - y_{m-5}) - \frac{c_{id}}{J_d} (y_{m-4} - y_{m-6}) + \frac{c_{ib}}{J_d} (y_{m-2} - y_{m-4}) + \frac{T_{fd}}{J_d} \\
 \dot{y}_{m-3} = y_{m-2} \\
 \dot{y}_{m-2} = -\frac{k_{ib}}{J_B} (y_{m-3} - y_{m-5}) + \frac{k_{tb}}{J_B} (y_{m-1} - y_{m-3}) - \frac{c_{ib}}{J_B} (y_{m-2} - y_{m-4}) + \frac{c_{tb}}{J_B} (y_m - y_{m-2}) + \frac{T_{fb}}{J_B} + \frac{T_f}{J_B} \\
 \dot{y}_{m-1} = y_m \\
 \dot{y}_m = -\frac{(k_{ib} + k_{iab})}{J_b} y_{m-1} + \frac{k_{ib}}{J_b} y_{m-3} - \frac{(c_{ib} + c_{iab})}{J_b} y_m + \frac{c_{ib}}{J_b} y_{m-2} + \frac{T_{PDM}}{J_b} + \frac{T_b}{J_b}
 \end{cases} \quad (13)$$

For the above established drill pillar system dynamics model, by means of numerical calculation, the dynamic model is solved, and the result of the

drilling pillar dynamics characteristic calculation is obtained. As shown in Table. 2.

Table. 2 Kinetic calculation parameters table

Parameter	Value	Parameter	Value
Input weight on bit, Wob (kN)	100	Input torque, Tm (N·m)	15,000
Drill tool outer diameter, DT (mm)	172	Bit outer diameter, Db (mm)	216
Drill pipe outer diameter, Di (mm)	114.3	Drill pipe inner diameter, di (mm)	97.18
Tool outer diameter, Dt (mm)	172	Tool inner diameter, dt (mm)	71.4
Horizontal section length (m)	2000	Length per drill pipe segment (m)	50
Tool length, Lc (m)	10.1	Mass per drill pipe segment (kg)	1432.3
Tool mass, mc (kg)	1400	Drill tool mass, mT (kg)	540.5
Bit mass, mb (kg)	120	Moment of inertia per drill pipe, Ji (kg·m ²)	61.63
Tool moment of inertia, Jc (kg·m ²)	612	Drill tool moment of inertia, JT (kg·m ²)	45.4
Bit moment of inertia, Jb (kg·m ²)	410	Torsional stiffness between drill pipe segments, kri(N/m)	685
Torsional stiffness between pipe & tool, kci (N/m)	980	Torsional stiffness between tool & drill tool, kcT (N/m)	865
Torsional stiffness between tool & bit, kcb (N/m)	830	Torsional stiffness between bit & rock, kc(b+1) (N/m)	224
Torsional damping between drill pipe segments, cri (Ns/m)	135.6	Torsional damping between pipe & tool, crc (Ns/m)	185
Torsional damping between tool & drill tool, crT (Ns/m)	146	Torsional damping between tool & bit, crb (Ns/m)	168
Torsional damping between drill tool & bit, crb (Ns/m)	184	Torsional damping between bit & rock, cr(b+1) (Ns/m)	76
Elastic modulus of drill string, E (Pa)	2.1×10 ¹¹	Shear modulus of drill string, G (Pa)	8.6×10 ¹⁰
Density of drill string material, ρ (kg/m ³)	7850	PDC-rock interface coefficient, c2	0.6
Dynamic friction coefficient, μ	0.3	Rayleigh damping coefficient, α	0.1
Rayleigh damping coefficient, β	0.001	Max static friction factor (bit-rock), Tfs	0.3
Viscous friction coefficient, μb	0.4	Static friction coefficient, μs	0.8
Coulomb friction coefficient, μc	0.5	Stribeck constant, ωb	0.5
Motor torque from downhole mud motor, TPDM (N·m)	8000	Mud motor bend housing angle, d (°)	1.5

CASE STUDIES AND RESULTS ANALYSES

Case studies

The key parameters of the Anti-drag pressure controller include the rotary speed, WOB, and pressure drop across the rotor ends. Changes in the rotary speed and pressure drop cause variations in the driving torque of the Anti-drag pressure controller, while changes in WOB lead to changes in the drill string system's anti-torque. Therefore, considering actual operating conditions, the rotary speed is divided into intervals of 30r/min, 45r/min, and 60r/min, the WOB is set within the range of 4t, 6t,

and 8t, and the pressure drop ranges from 6MPa, 8MPa, to 10MPa. By keeping other parameters constant and analyzing the same parameter, the dynamic results of each module's unit can be obtained for comparative analysis to study the dynamic effects of key parameter variations on the Anti-drag pressure controller.

Results analyses

(1) Effect of rotary speed

The angular velocity results of different rotary speeds for each module are shown in Fig. 7. From the calculation results, the torsional vibration angular velocity of the drill string system initially increases and then decreases, showing an increasing trend

before gradually stabilizing at a certain value. The stable angular velocities of the bit for the three rotational speeds are 30.94rad/s, 34.05rad/s, and 37.21rad/s, while the stable angular velocities of drill pipe 3 are 26.42rad/s, 27.19rad/s, and 27.96rad/s, respectively. Combining Fig. 7(a), 7(b), and 7(c), it can be observed that as the rotary speed increases, the stable angular velocities of the bit, drill string assembly, and clutch controller also increase, with similar growth rates. This is because the increased driving torque generated by the Anti-drag pressure controller compensates for a portion of the drill string system's anti-torque, leading to increased angular velocities of the units below the controller. This demonstrates that the use of a hydraulic clutch can enhance the mechanical rotational speed of the drill bit and improve drilling efficiency. Fig. 7(d) shows the angular velocity results of drill pipe 3, which also increases with the rotational speed but to a lesser extent than the modules below the controller. This is due to the disengagement of the torque transmission between the controller and the upper drill string system from the drill collar, resulting in less impact on the upper drill string, but the angular velocity of drill pipe 3 still follows the increasing trend due to overcoming the drill string system's anti-torque.

In combination with Figs. 7(b), (c), and (d), it

can be observed that the angular velocities of the drill tool, controller, and Drill Pipe 3 exhibit a transient trend of increasing, then decreasing, followed by a secondary rise before finally stabilizing. A similar transient pattern is clearly visible in Fig. 9. This phenomenon is primarily attributed to the damping characteristics of the drill string system and the transient dynamics associated with torque transmission. When the top drive torque is initially applied, the upper part of the drill string responds quickly, leading to a rapid increase in angular velocity. However, due to the system's distributed inertia and torsional stiffness, the lower components—such as the bit and bottom-hole assembly (BHA)—exhibit delayed responses. This mismatch in angular velocity induces a transient counter-torque effect. The combined influence of this mismatch, torsional damping, and frictional resistance—modeled using the Karnopp friction model as described in Section 3—results in a temporary decrease in angular velocity. As the applied torque continues, and once the transient oscillations are dissipated by damping effects, the system begins to accelerate again toward a new dynamic balance. This re-acceleration phase is driven by the sustained input torque overcoming both damping and frictional resistance, eventually leading to a steady-state rotational condition.

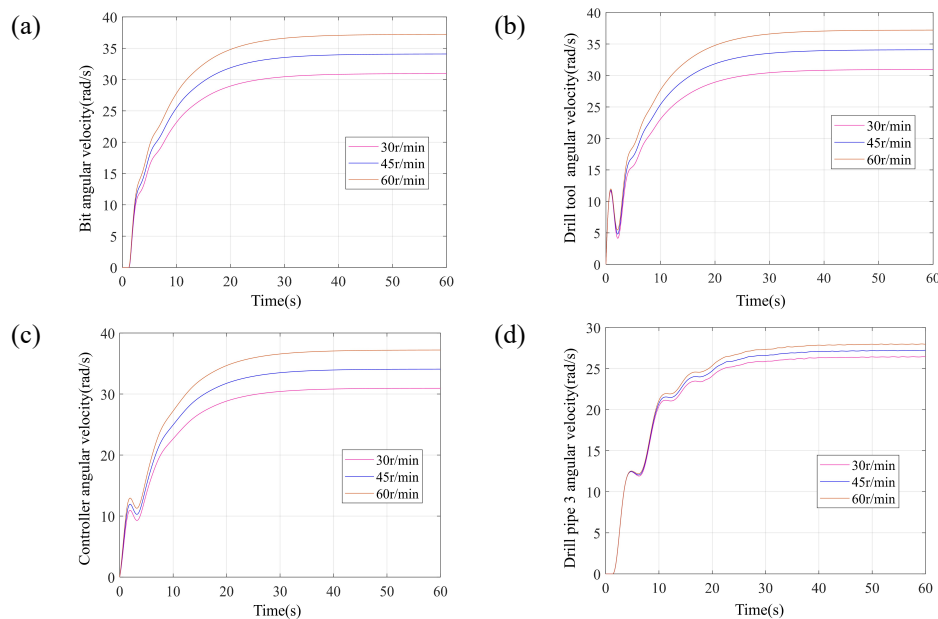


Fig. 7 Angular velocity of each part under different rotary speeds:(a) Bit,(b) BHA,(c) Controller,(d) Drill pipe 3

The torsional vibration phase diagrams of the modules at different rotary speeds are shown in Fig. 8. The bit, drill string assembly, and controller exhibit similar phase trajectories that gradually converge to steady-state rotational motion, without sustained oscillations or periodic closed loops. Taking the bit as an example, the angular displacement lags behind the rotary table initially, then the angular velocity rapidly increases to its peak, gradually decreases, and finally

stabilizes at a straight line. As the rotary speed increases, the phase trajectories of the bit, drill string assembly, and controller extend towards the upper right direction. However, at the initial stage, the angular velocity of the bit relative to the rotary table is the highest, followed by the drill string assembly and the controller. This is because the drill string assembly is closer to the controller, resulting in a more noticeable effect of balancing the anti-torque.

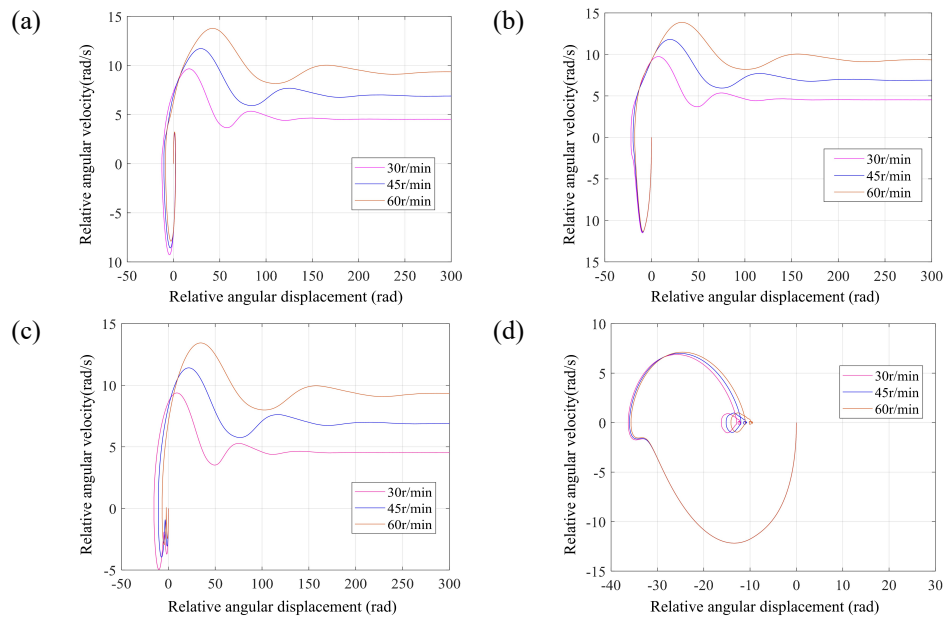


Fig. 8 Different rotary speed, each part of the phase diagram:(a) Bit, (b) BHA, (c) Controller, (d) Drill pipe 3

(2) The effect of weight on bit

The angular velocity results of each module for different WOBs are shown in Fig. 9. It can be observed that the angular velocities of the modules increase first, then decrease, and then increase again before gradually stabilizing. However, under the same conditions, the angular velocities of the controller, drill string assembly, and bit are much higher than those of drill pipe 3. The stable angular velocities of the bit for the three WOBs are 47.54rad/s, 39.25rad/s, and 30.95rad/s, while the stable angular velocities of drill pipe 3 are 30.52rad/s, 28.51rad/s, and 26.41rad/s, respectively. As the WOB

increases, the angular velocities of the modules decrease.

This is because the increased WOB leads to an increase in the friction torque of each unit in the drill string system, with larger torques closer to the bit. The stable angular velocity of drill pipe 3 has a longer duration, but the influence of WOB on drill pipe 3 is relatively small. In actual drilling operations, it is possible to reduce the upper WOB appropriately to increase the mechanical rotational speed of the bit. However, the WOB should not be too low, as it may lead to bit bouncing and other phenomena.

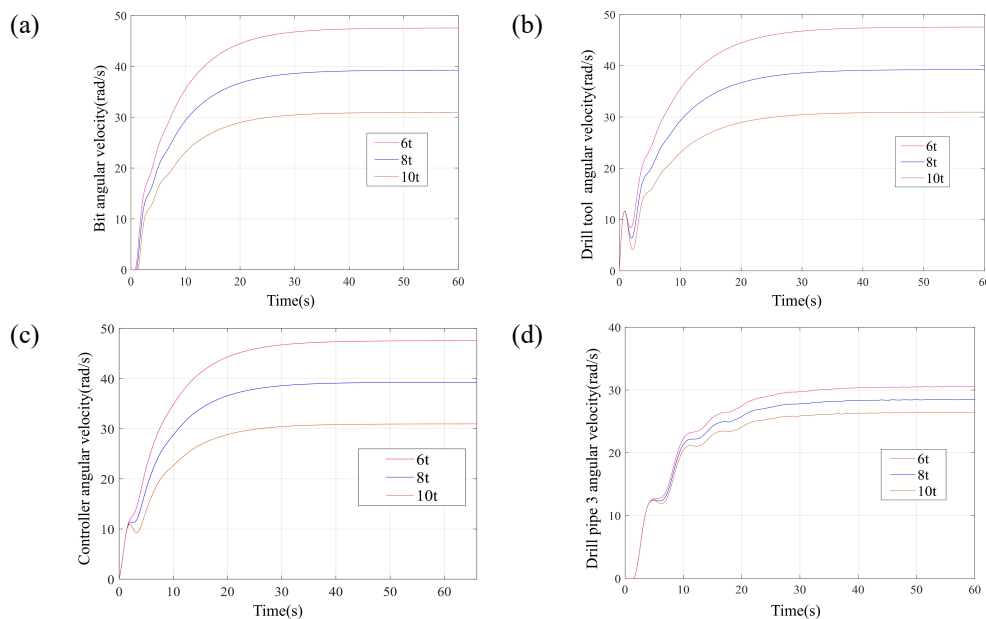


Fig. 9 Angular velocity of each part under different bit weight:(a) Bit,(b) BHA,(c) Controller,(d) Drill pipe 3

The torsional vibration phase diagrams of the modules at different rotary speeds for different WOB are shown in Fig. 10. The phase trajectories of the bit, drill string assembly, and controller continue to extend toward the right as WOB increases, indicating that these components reach higher steady-state angular velocities under increasing load. The bit does not exhibit stick-slip behavior. Moreover, although the increase in WOB causes the bit's phase trajectory

to shift downward, its motion converges more rapidly to a steady rotational condition, with minimal oscillatory behavior. For Drill Pipe 3, the phase trajectory gradually shifts to the left with increasing WOB, suggesting a delayed response in angular acceleration due to higher resistance. Over time, its relative angular velocity with respect to the rotary table approaches zero, indicating convergence toward uniform rotation.

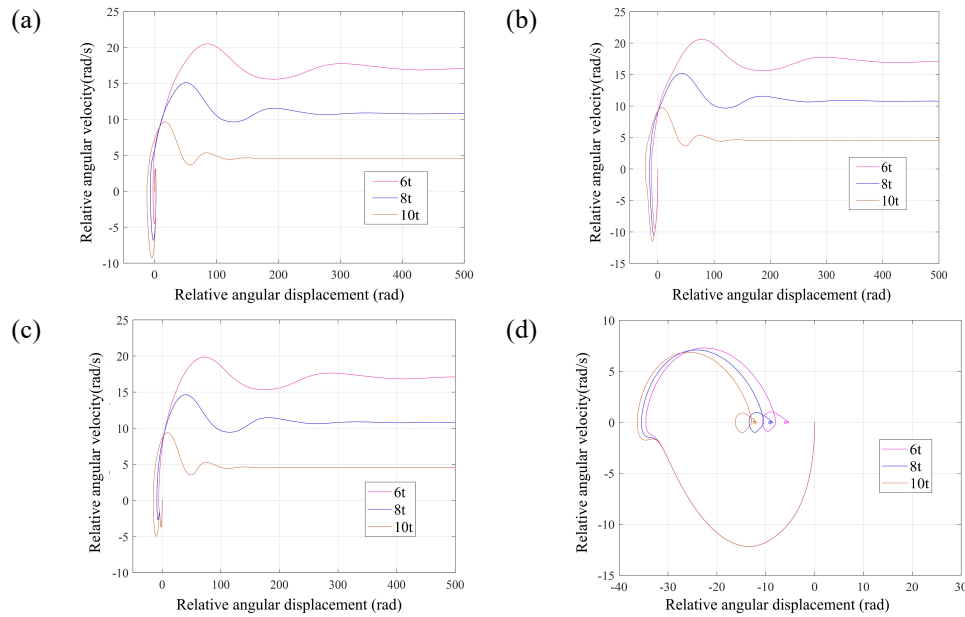
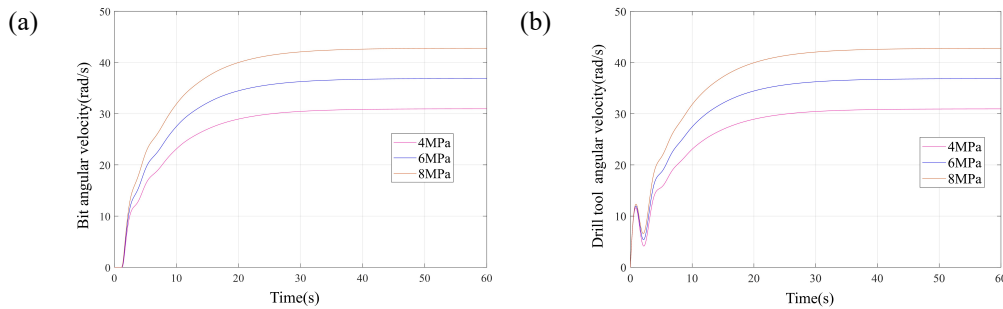


Fig. 10 Different WOB, each part of the phase diagram:(a) Bit,(b) BHA,(c) Controller,(d) Drill pipe 3

(3) The effect of pressure drop

Fig. 11 shows the angular velocity results of each module for different pressure drops. It can be observed that the angular velocities of the modules increase first, then decrease, and then increase again before gradually stabilizing. However, under the same conditions, the angular velocities of the controller, drill string assembly, and bit are much higher than those of drill pipe 3. The stable angular velocities of the bit for the three pressure drops are 30.95rad/s, 36.85rad/s, and 42.76rad/s, while the stable angular velocities of drill pipe 3 are 26.41rad/s,

27.86rad/s, and 29.32rad/s, respectively. As the pressure drop increases, the angular velocities of the modules increase, and the time required for drill pipe 3 to reach stability becomes longer. Similarly, the increase in pressure drop has a relatively small impact on drill pipe 3 compared to the changes in the Anti-drag pressure controller's driving torque. This is because an increased pressure drop leads to an increase in the driving torque of the Anti-drag pressure controller, which effectively balances the drill string system's anti-torque.



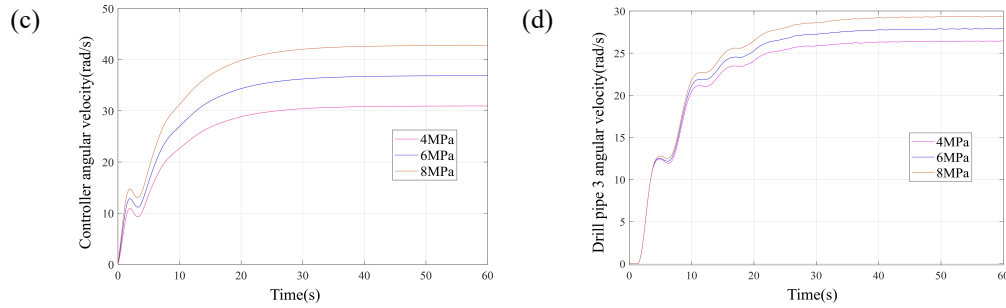


Fig. 11 Angular velocity of each part under different pressure drops:(a)Bit,(b)BHA,(c)Controller,(d)Drill pipe 3

The torsional vibration phase diagrams of the modules at different rotary speeds for different pressure drops are shown in Fig. 12. It can be observed that the phase trajectories of the bit, drill string assembly, and controller still extend towards the right, indicating that the bit, drill string assembly, and controller can maintain stable angular velocities with increasing pressure drop. The bit does not exhibit sticking-slipping phenomena. Moreover, as

the pressure drop increases, the bit's phase trajectory shifts upwards, and the trajectory remains stable. With increasing pressure drop, the phase trajectory of drill pipe 3 gradually shifts to the right, indicating a shorter time for drill pipe 3 to start moving and reach stability. The magnitude of this change in trajectory is similar to that observed during variations in rotational speed.

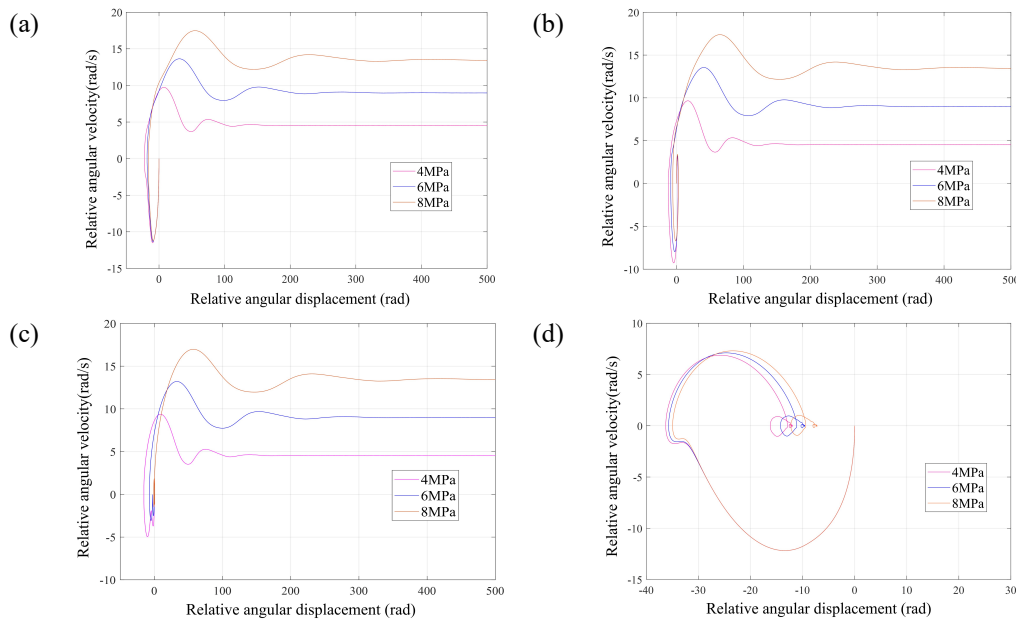


Fig.12 Different pressure drop, each part of the phase diagram: (a) Bit, (b) BHA, (c) Controller, (d) Drill pipe 3

(4) Placement Position Analysis

To investigate the optimal placement of the hydraulic clutch controller, the controller was positioned at three different locations: before the stabilizer, after the stabilizer, and behind drill pipe 5. The dynamic responses of the drill bit at these positions were analyzed. As shown in Fig. 13, the angular velocity of the drill bit gradually increased from zero and stabilized at 30.96 rad/s when the controller was placed before the stabilizer, following the sequence of drill bit, drill string, controller, stabilizer, and drill pipe 5. In contrast, when the controller was placed after the stabilizer or behind drill pipe 5, the drill bit exhibited a delayed response, with the delay increasing as the controller was

positioned farther from the drill bit. In these two cases, the angular velocity of the drill bit oscillated in a pattern of increase-decrease-increase-decrease, eventually stabilizing at approximately 15.71 rad/s and 15.23 rad/s after 40 seconds, respectively. The significantly lower stable angular velocity in these configurations compared to the first placement is attributed to the fact that shifting the controller upward requires balancing a greater reaction torque. Since the generated driving torque remains constant under the same operating parameters, the drill bit's stable angular velocity is much lower in the latter two cases. Additionally, as the controller is moved farther upward, its influence on the drill bit diminishes, leading to similar stable angular velocities in the last

two configurations.

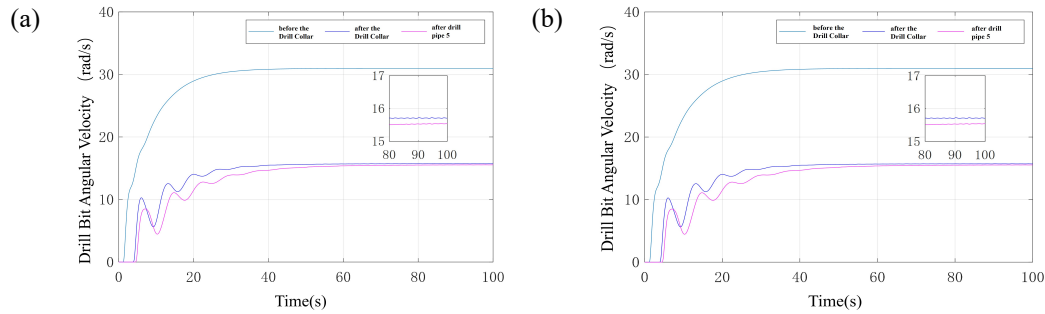


Fig. 13 Dynamic Results of the Drill Bit at Different Placement Positions: (a) Drill Bit Angular Velocity Results, (b) Drill Bit Phase Diagram

From the drill bit phase diagram results, it can be observed that when the controller is placed before the stabilizer, the drill bit initially lags behind the rotary table but quickly accelerates to its peak amplitude, then decays and stabilizes. The phase trajectory extends to the right, indicating that the drill bit rotates faster than the rotary table. In contrast, when the controller is placed after the stabilizer or behind drill pipe 5, the phase trajectory initially extends downward from the origin (0, 0) and continues fluctuating downward, exhibiting instability until the drill bit angular velocity stabilizes. During this period, the phase trajectory extends continuously to the left, signifying that the drill bit persistently lags behind the rotary table.

Positioning the controller closer to the upper rotary table enhances the mechanical rotational speed of the upper drill string system, such as the individual drill pipes. This facilitates the lowering of the drill string and the application of weight on bit (WOB), but it is less favorable for increasing the drill bit speed. Conversely, placing the controller farther down requires it to generate a greater driving torque to counteract the reaction torque under the same operating conditions. This increases tool wear and shortens its service life. Moreover, with more lower-string units involved, the controller's directional drilling capability diminishes, making it harder to stabilize the tool face angle. Therefore, placing the hydraulic clutch controller before the stabilizer yields better overall performance.

CONCLUSIONS

This paper proposes an anti-drag pressure controller (ADPC) that integrates directional drilling with compound drilling. By controlling the upper driving speed, the ADPC enables both compound and directional drilling functions. Experimental testing confirms that as the flow rate increases, the threshold speed required to stabilize the ADPC housing also increases.

Considering the working mechanism of the ADPC and the actual conditions of directional wells,

a dynamic model of the drill string is established to calculate the torsional vibration characteristics at each node. The results show that with increasing rotational speed and pressure drop, the angular velocities of the units below the controller initially increase and then stabilize at a constant value. When the rotational speed reaches 60 rpm, the drill bit achieves its maximum stable angular velocity of 37.21 rad/s. When the pressure drop reaches 8 MPa, the ADPC achieves its maximum stable angular velocity of 42.77 rad/s. However, as the weight on bit (WOB) increases, the angular velocities of the drill bit and other components decrease. At a WOB of 10 t, drill pipe 3 exhibits the lowest stable angular velocity of 26.41 rad/s.

Finally, the optimal placement position of the ADPC is evaluated by performing dynamic calculations with the ADPC positioned in front of the drill collar, behind the drill collar, and behind drill pipe 5. The results indicate that placing the ADPC in front of the drill collar significantly increases the drill bit angular velocity, shortens the response time, reduces vibration and counter torque, and enhances tool longevity. This configuration enables more efficient and stable directional drilling performance.

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