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## Research Article

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# Leakage fluctuation characteristics of a spherical seal in a push-the-bit rotary steerable tool under random oscillation

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## Abstract

Leakage fluctuation of the spherical seal is an important factor affecting the hydraulic stability of internal push point-the-bit rotary steerable tools, especially during non-steering hold section drilling. This study investigates the leakage behavior of the spherical sealing structure under guided and random mandrel oscillations. Based on spherical geometry and micro-clearance leakage theory, mathematical models describing mandrel motion, three-dimensional clearance distribution, and transient leakage rate were established. A three-dimensional numerical model incorporating the RNG  $k-\varepsilon$  turbulence model was then developed to analyze the coupled flow through the main passage and spherical clearance. The effects of spherical normal clearance, drilling fluid density, and inlet flow rate on leakage and pressure drop fluctuations were evaluated. The results show that guided oscillation produces a relatively stable leakage state, whereas random oscillation continuously redistributes the local clearance and induces leakage fluctuations. Increasing the spherical clearance raises the mean leakage rate but reduces its deviation from the mean, indicating weaker relative fluctuation. Higher drilling fluid density enhances the sensitivity of leakage to local clearance disturbance, while its effect on pressure drop fluctuation is limited. Reducing the inlet flow rate suppresses leakage fluctuation, although the mean leakage rate increases. These results clarify the coupling between random mandrel motion, local clearance variation, and hydraulic response, providing a theoretical basis for spherical seal optimization and hydraulic stability control in rotary steerable tools.

**Keywords:** Rotary steerable tool; Spherical seal; Random oscillation; Leakage fluctuation; Clearance leakage

## 1 Introduction

The continuous development of deep, ultra-deep, unconventional, and extended-reach reservoirs has imposed increasingly stringent requirements on wellbore trajectory control, drilling efficiency, and downhole tool reliability. Compared with conventional slide drilling, rotary steerable systems (RSSs) enable continuous drill-string rotation while actively controlling the drilling direction, thereby reducing frictional resistance, improving hole quality, and enhancing trajectory control efficiency. According to the steering principle, existing RSSs are generally classified as push-the-bit and point-the-bit systems. Considerable efforts have therefore been devoted to improving steering mechanisms, build up capability, bottom hole assembly (BHA) mechanics, toolface control, and system reliability (Li et al. 2014; Wang et al. 2019; Shi et al. 2020; Liu et al. 2022; Chen et al. 2023; Jia et al. 2024, 2025). Recent reviews further indicate that high build-up capability, high-temperature adaptability, vibration suppression, and intelligent closed-loop control have become major development directions for advanced RSSs (Geng et al. 2025; Tian et al. 2026; Y. Zhang et al. 2026; Zhao et al. 2026).

From the perspective of steering mechanics and trajectory control, extensive theoretical and numerical studies have been conducted. Huang et al. Shi et al. (2020) developed a build-up-rate prediction method for a static push-the-bit RSS by considering the combined leverage, pendulum, and thrust effects. Liu et al. (2022) analyzed the factors governing the trajectory-control capability of a reverse push-the-bit RSS and demonstrated the importance of steering-force transmission and structural parameters. Zeng et al. Zeng et al. (2022) established a mechanical model of a push-the-bit rotary steerable BHA considering drilling and structural parameters. For the internal push point-the-bit rotary steerable tool, Chen et al. Chen et al. (2023) combined mechanical and trajectory methods to establish a build-up-rate

calculation model that accounts for tool structure, borehole geometry, drilling parameters, bit anisotropy, and formation characteristics. More recently, three-dimensional dynamic finite-element approaches have been introduced to capture BHA flexure, nonlinear borehole contact, transient vibration, and steering-induced dynamic responses (Jia et al. 2025; Tian et al. 2026; Zhao et al. 2026). Jia et al. (2024) further investigated the working resistance of steering ribs, whereas Li et al. (2023) analyzed whirling behavior and fatigue life of a push-the-bit rotary steerable BHA. Drilling-parameter optimization has also been investigated to improve the dynamic steering performance of point-the-bit RSSs (Y. Zhang et al. 2026).

Another important research direction concerns the hydraulic actuation and control stability of RSSs. Wang et al. (2019, 2022a, b) investigated toolface control, sensor-fault detection, and disturbance rejection in dynamic point-the-bit RSSs, demonstrating that drilling disturbances and system uncertainty can considerably affect steering accuracy. Liang et al. (2025) developed a closed loop vector force control strategy for a hydraulic push-the-bit RSS, while intelligent and disturbance-observer-based methods have also been introduced to improve attitude and trajectory control (Zhang and Huo 2025; N. Zhang et al. 2026). From the fluid dynamic perspective, Han et al. (2024) established a transient flow model for high rate mud pulse transmission in an RSS, emphasizing the importance of unsteady drilling fluid pressure and flow processes. More directly related to the internal push point-the-bit configuration, Kang et al. (2026) developed and experimentally investigated a three hydraulic cylinder synchronous steering offset actuator driven by a drilling fluid rotary valve distributor.

However, previous sealing studies have primarily focused on planar or cylindrical annular gaps, gas-film seals, conventional mechanical seals, and spherical valve-plate pairs (Sato and Ochiai 2022; Anbarsooz et al. 2022; Ma et al. 2023; Zheng et al. 2023; Xu et al. 2024). The effects of clearance, eccentricity, inclination, and whirl motion on leakage and dynamic seal performance have also been extensively investigated (Li et al. 2022a, b; Zhang et al. 2023; Kim et al. 2025; F. Wang and Y. Wang 2026; X. Zhang et al. 2026). Their working media are typically air, gas, or lubricating oil, and the motion of the sealing surfaces is generally prescribed as steady rotation, fixed eccentricity, controlled tilt, or periodic whirl. The operating characteristics of the spherical seal in an internal push point-the-bit RSS are fundamentally different. During active steering, the mandrel undergoes a relatively regular and directional deflection driven by the steering actuator. During non-steering or hold section drilling, by contrast, assembly clearance, rotational dynamics, bearing constraints, and downhole disturbances may induce small-amplitude spatial random oscillations of the mandrel. Such motion continuously redistributes the spherical sealing clearance and can consequently generate unsteady leakage even when the nominal clearance remains unchanged. Existing RSS studies largely focus on steering mechanics and trajectory control, whereas existing clearance-seal studies generally do not address a spherical sealing interface subjected to three-dimensional random oscillation and drilling-fluid flow. Therefore, the relationship among mandrel random oscillation, local spherical clearance variation, leakage fluctuation, and hydraulic-pressure response remains insufficiently understood.

To address this issue, the present study investigates the leakage fluctuation characteristics of the spherical seal in an internal push point-the-bit rotary steerable tool under different mandrel motion states. First, the structural configuration and operating principle of the spherical sealing system are analyzed, and the differences between guided oscillation during steering and random oscillation during non-steering operation are identified. Based on spherical geometry and micro clearance flow theory, kinematic models of the mandrel, spatial clearance-distribution models, and transient leakage models are then established for the two operating conditions. Subsequently, a three-dimensional numerical model based on the RNG  $k$ - $\varepsilon$  turbulence formulation is developed to investigate the coupled flow through the main passage and spherical leakage clearance. Particular attention is paid to the effects of spherical normal clearance, drilling-fluid density, and inlet flow rate on the mean leakage rate, leakage fluctuation, and pressure-drop response. By distinguishing deterministic steering motion from non-steering random oscillation, this work extends conventional steady or prescribed eccentricity seal analysis to the stochastic motion condition encountered in rotary steerable drilling and provides a theoretical basis for spherical-seal design and hydraulic system stability improvement.

## 2 Materials and methods

The performance of the spherical sealing structure in the internal push point-the-bit rotary steerable tool is highly coupled with the overall structural design, the motion characteristics of the core mechanism, and the operating conditions. To accurately reveal the leakage fluctuation mechanism of the spherical seal under different operating conditions, this section analyzes the design parameters and technical characteristics of the adjustment mechanism and spherical sealing system, distinguishes the steering and non steering operating modes, and compares the differences in mandrel motion between the two conditions. This provides the structural basis for the subsequent analysis of leakage fluctuation characteristics.

### 2.1 Structure and working principle

As shown in Fig.1, the spherical sealing structure incorporates a mandrel assembly. The spherical seat bearings are arranged on both sides of the mandrel ball head and positioned close to the housing. A small clearance exists between the ball head and the bearings, allowing the mandrel to undergo limited oscillation about the spherical center rather than remaining in full contact with the surrounding components. Both the sealing element and the mandrel contain internal flow passages for drilling fluid, and the two components are concentrically nested within the housing. The spherical clearance, defined as the normal mating gap between the center tube of the mandrel and the jet valve seat, constitutes the primary leakage path for the drilling fluid. Similar hydraulic steering principles and actuator configurations have been reported for push-the-bit and internal push point-the-bit rotary steerable systems (Chen et al. 2023; Kang et al. 2026).

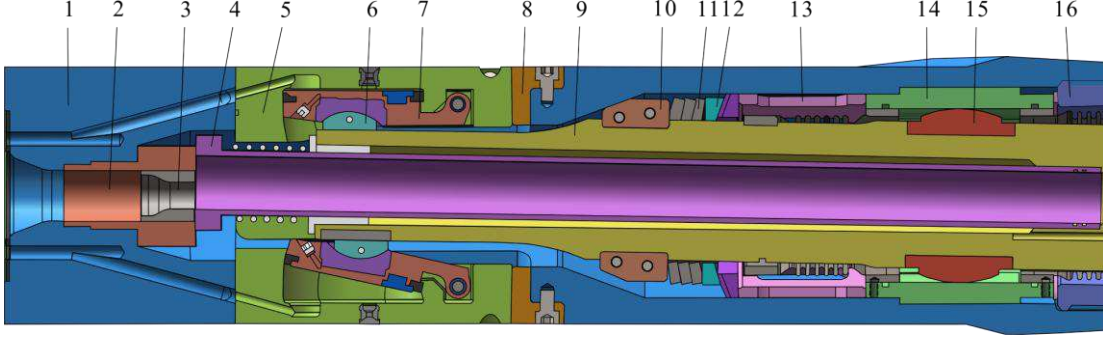


**Fig. 1** Structure of the spherical sealing system. (1) load bearing housing; (2) jet valve seat; (3) jet nozzle; (4) central tube; (5) pad hydraulic cylinder; (6) pad; (7) pad oscillating piston; (8) pad chamber clamping wedge; (9) bit holder; (10) anti drop ring; (11) disc spring; (12) load bearing ring; (13) disc spring washer; (14) spline torque transmission bushing; (15) barrel shaped key; (16) anti drop sub.

The operating process of the spherical sealing structure can be divided into two states: random oscillation and guided oscillation, which exhibit markedly different leakage behaviors and operating characteristics. When a change in the wellbore trajectory is required, the structure enters the guided-oscillation mode. Based on the preset build-up rate and azimuth, the measurement and control system sends commands to the servo hydraulic cylinders located at the corresponding circumferential positions. The servo cylinders drive the adjustment wedges to move axially, and the inclined surface transmission converts the axial force into a radial force. As a result, the mandrel undergoes directional oscillation about the midpoint of the line connecting the centers of the spherical heads, with a maximum deflection angle of  $1^\circ$ . Under this condition, the mandrel exhibits a relatively large oscillation amplitude and a stable motion pattern. The spherical sealing clearance changes smoothly with the directional deflection of the mandrel, resulting in a comparatively uniform and continuously varying flow field within the sealing gap. Consequently, the internal drilling fluid forms a relatively stable steady state leakage flow through the spherical clearance, with a comparatively high mean leakage rate.

When the wellbore trajectory reaches the design target, the tool enters the hold-section drilling stage. At this time, the measurement and control system simultaneously sends identical pressure commands to the three servo hydraulic cylinders, locking the adjustment wedges at the central position and maintaining the mandrel coaxial with the housing. However, owing to inherent factors such as structural assembly clearances, rotational dynamic effects, and bearing constraints, the mandrel still exhibits small amplitude spatial random oscillations during operation. The oscillation amplitude is much smaller than the deflection angle under steering conditions. Under this operating condition, both the average oscillation angle and the mean leakage rate are lower than those in the guided oscillation condition. Meanwhile, the flow field within the spherical sealing clearance remains in a continuously unsteady adjustment state, resulting in dynamic fluctuations in the leakage rate. Fig.2 illustrates the operating states of the spherical sealing structure under random oscillation and guided-oscillation conditions.

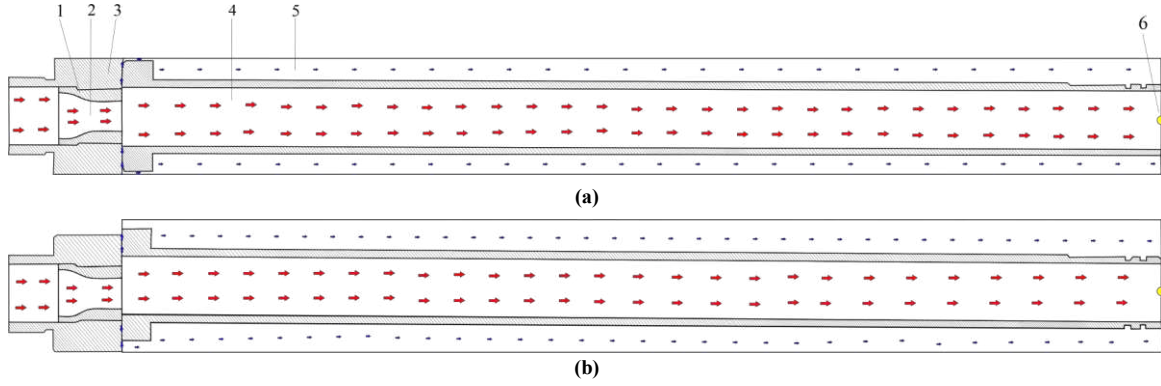


**Fig. 2** Operating states of the spherical sealing system. (a) non steering condition; (b) steering condition; (1) jet piston; (2) piston flow passage; (3) jet valve seat; (4) central tube flow passage; (5) leakage passage; (6) Spherical center pivot point.

## 2.2 Mathematical model

Based on the structural characteristics and operating principle of the adjustment mechanism and spherical sealing system of the rotary steerable tool, as well as the inherent motion characteristics under guided oscillation and random-oscillation conditions, this section establishes the mandrel oscillation model, spherical clearance model, and leakage calculation model for the two operating conditions, respectively. These models provide a theoretical basis for the subsequent numerical analysis and experimental validation of leakage fluctuation characteristics. The geometric model of the spherical sealing structure is shown in Fig.3.

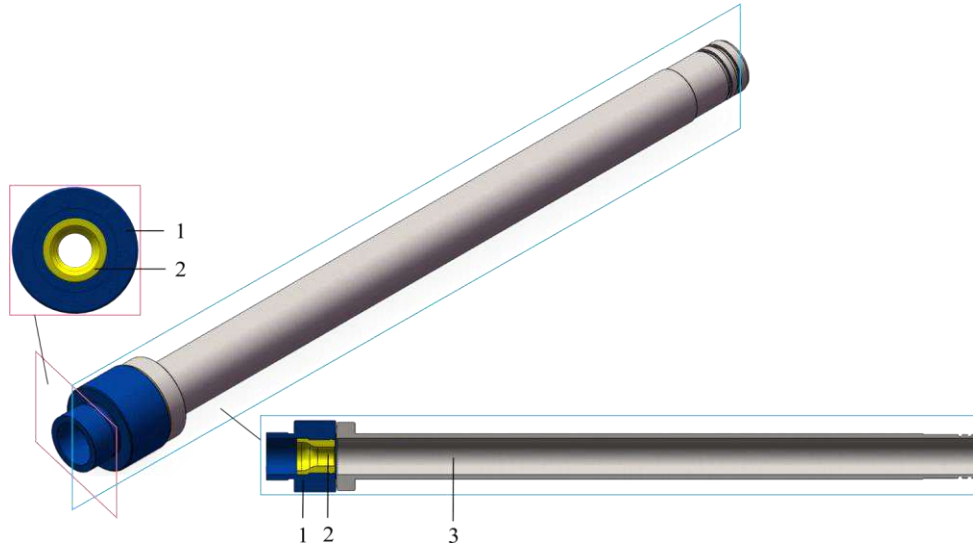


**Fig. 3** Geometric model of the spherical sealing system. (1) jet valve seat; (2) jet piston; (3) center tube.

To highlight the dominant physical mechanisms, reduce computational complexity, and maintain consistency between the model and the actual operating conditions of the internal push point-the-bit rotary steerable tool, the following reasonable assumptions are made:

(1) The effects of microscopic machining errors and surface roughness of the sealing surfaces are neglected. The spherical sealing structure is simplified as an ideal smooth spherical pair, with the radius of curvature of the sealing surfaces fixed at the design value and the nominal sealing clearance maintained at a constant microscale level.

(2) Under both operating conditions, the sealing surfaces remain continuously engaged without contact separation. Mandrel oscillation only changes the spatial distribution of the sealing clearance and does not alter its overall order of magnitude.

(3) The drilling fluid is assumed to be incompressible, and its dynamic viscosity and density are considered insensitive to variations in temperature and pressure. The flow within the sealing clearance is assumed to be laminar, while turbulent effects and inertial forces are neglected.

(4) The spherical bearing seat is assumed to remain stationary, with only the mandrel undergoing oscillatory motion. The boundary pressures are kept constant, and the influence of fluid intrusion on the leakage flow field is neglected.

(5) The effects of temperature variation, elastic deformation of the material, and structural vibration on the sealing clearance and leakage characteristics are neglected. Only the dominant influence of mandrel oscillation on the dynamic response of the system is considered.

Similar idealizations of smooth sealing surfaces, incompressible fluids, and clearance-dominated flow have been widely adopted in theoretical and numerical investigations of micro-clearance seals (Bei et al. 2021; Ma et al. 2023; Xu et al. 2024; Z. Wang et al. 2026).

The operating process of the spherical sealing structure can be divided into two states: random oscillation and guided oscillation, which exhibit markedly different leakage behaviors and operating characteristics. When a change in the wellbore trajectory is required, the structure enters the guided oscillation mode. Based on the preset build-up rate and azimuth, the measurement and control system sends commands to the servo hydraulic cylinders located at the corresponding circumferential positions. The servo cylinders drive the adjustment wedges to move axially, and the inclined-surface transmission converts the axial force into a radial force. As a result, the mandrel undergoes directional oscillation about the midpoint of the line connecting the centers of the spherical heads, with a maximum deflection angle of  $1^\circ$ . Under this condition, the mandrel exhibits a relatively large oscillation amplitude and a stable motion pattern. The spherical sealing clearance changes smoothly with the directional deflection of the mandrel, resulting in a comparatively uniform and continuously varying flow field within the sealing gap. Consequently, the internal drilling fluid forms a relatively stable steady state leakage flow through the spherical clearance, with a comparatively high mean leakage rate.

Guided oscillation is an actively controlled motion driven by the servo hydraulic cylinders. It is characterized by low frequency, directional motion, and regularity, with the primary objective of achieving precise wellbore trajectory control through the directional deflection of the mandrel.

Taking the midpoint of the line connecting the centers of the spherical heads as the origin  $O$ , the initial rotation axis of the mandrel is defined as the  $z$ -axis, with the horizontal direction as the  $x$ -axis and the vertical direction as the  $y$ -axis. Guided oscillation occurs only in the  $xOy$  plane, and the instantaneous deflection angle of the mandrel varies sinusoidally with time, consistent with the closed loop control characteristics of the servo hydraulic cylinders:

$$\theta_1(t) = \theta_{\max} \cdot \sin(2\pi f_1 t) \quad (1)$$

where  $\theta_1(t)$  is the instantaneous deflection angle under guided oscillation, rad;  $\theta_{\max}$  is the maximum guided oscillation angle, rad;  $f_1$  is the guided oscillation frequency, Hz; and  $t$  is time, s.

This model describes the directional oscillation of the mandrel under guided oscillation conditions, in which the instantaneous deflection angle varies smoothly and periodically with time.

Based on the mandrel oscillation model, a spherical coordinate system is established with the common spherical center as the origin. When the mandrel undergoes an instantaneous deflection angle  $\theta_1(t)$  in the  $xOy$  plane, the instantaneous clearance distribution at any point on the sealing surface can be expressed as:

$$h_1(\theta, \phi, t) = h_0 + \Delta R \cdot \theta_1(t) \cdot \sin \theta \cos \phi \quad (2)$$

where  $\theta$  is the polar angle at an arbitrary point, rad;  $\phi$  is the azimuth angle at an arbitrary point, rad; and  $h_0$  is the initial uniform clearance thickness, m.

The model indicates that the sealing clearance remains constant and is independent of mandrel oscillation. Fig.4 shows the clearance model between the spherical surface of the mandrel and that of the sealing element.

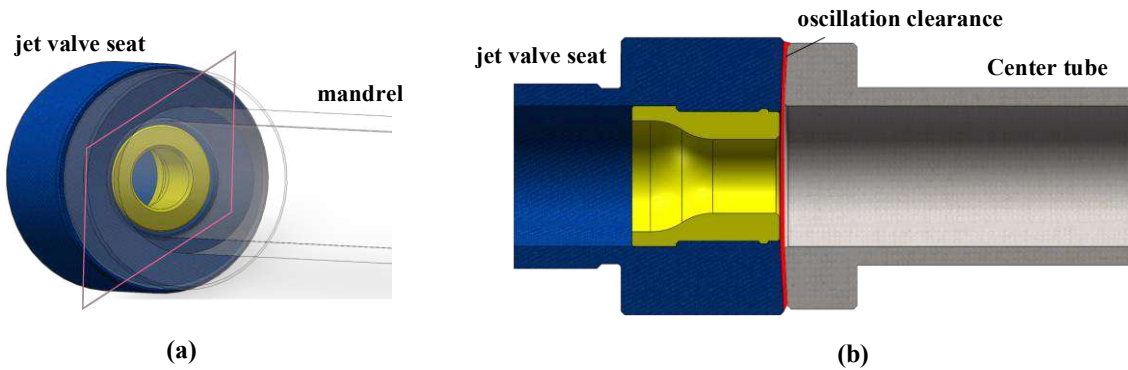


**Fig. 4** spherical clearance model between mandrel and seal. (a) leakage clearance configuration; (b) leakage gap flow domain.

The leakage flow through the spherical seal can be regarded as pressure driven micro clearance flow. For a sufficiently small clearance, the local flow region can be approximated as a parallel plate slit and described using the Poiseuille flow assumption, which has been widely employed in leakage models for narrow sealing interfaces (Bei et al. 2021; van der Walt et al. 2024; Z. Wang et al. 2026). An infinitesimal annular region is considered on the spherical sealing surface. At the local microscale, this region can be approximated as a parallel plate slit, and the corresponding differential leakage rate is expressed as:

$$dQ = \frac{\Delta p \cdot h^3}{12\mu R} \cdot R^2 \sin \theta d\theta d\phi \quad (3)$$

where  $\Delta p$  is the pressure difference across the sealing surfaces, Pa;  $h$  is the local clearance thickness at the infinitesimal element, m;  $\mu$  is the dynamic viscosity of the drilling fluid, Pa·s;  $R$  is the radius of curvature of the spherical surface, m;  $\theta$  is the polar angle of the infinitesimal element, rad; and  $\Phi$  is the azimuth angle of the infinitesimal element, rad.

By integrating Eq.(3) over the entire spherical sealing surface, with the polar angle ranging from  $\alpha_1 \leq \theta \leq \alpha_2$  and the azimuth angle ranging from  $0 \leq \Phi \leq 2\pi$ , the instantaneous leakage rate of the spherical sealing structure under guided oscillation conditions can be expressed as:

$$Q_1(t) = \frac{\Delta p R}{12\mu} \int_{\alpha_1}^{\alpha_2} \int_0^{2\pi} h_1(\theta, \phi, t)^3 \sin \theta d\phi d\theta \quad (4)$$

By combining and simplifying the differential leakage rate expression, the analytical expression for the instantaneous leakage rate can be obtained as follows:

$$Q_1(t) = Q_0 \left( 1 + 1.5\theta_1(t)^2 \right) \quad (5)$$

where  $Q_0$  is the baseline leakage rate when the mandrel is coaxial with the tool axis, and is expressed as:

$$Q_0 = \frac{\pi \Delta p h_0^3}{6\mu R} \cdot \frac{\cos^3 \alpha_1 - \cos^3 \alpha_2}{\cos \alpha_1 - \cos \alpha_2} \quad (6)$$

Random oscillation is induced by the combined effects of mandrel rotation, structural fit clearances, and dynamic interactions, and is a primary factor causing fluctuations in seal leakage and system pressure. Based on random process theory, an unsteady coupled mathematical model is established for the random oscillation condition. Previous transient seal flow studies have demonstrated that eccentricity, whirl motion, and external disturbances can continuously alter the local clearance distribution and consequently modify the hydraulic response of the sealing flow field (Zhang et al. 2020; F. Li et al. 2021; Kim et al. 2025).

Unlike the planar directional motion associated with guided oscillation, random oscillation may occur in any plane passing through the spherical center and therefore needs to be described in vector form. Let the instantaneous oscillation angle vector be  $\theta_r(t)$ , with its components in the  $x$ - and  $y$ -directions of the Cartesian coordinate system denoted by  $\theta_{rx}(t)$  and  $\theta_{ry}(t)$ , respectively. The instantaneous deflection-angle magnitude and oscillation azimuth angle can then be expressed as:

$$\theta_r(t) = \sqrt{\theta_{rx}(t)^2 + \theta_{ry}(t)^2} \quad (7)$$

$$\phi_r(t) = \arctan 2(\phi_{ry}(t), \phi_{rx}(t)) \quad (8)$$

where  $\theta_r(t)$  is the instantaneous random-oscillation angle vector of the mandrel, rad;  $\theta_{rx}(t)$  is the component of the oscillation angle in the  $x$ -direction, rad;  $\theta_{ry}(t)$  is the component of the oscillation angle in the  $y$ -direction, rad; and  $\Phi_r(t)$  is the instantaneous oscillation azimuth angle, rad.

According to the geometric distance relationship between two points on a spherical surface, the instantaneous clearance thickness at any point on the sealing surface can be expressed as:

$$h_r(\theta, \phi, t) = \Delta R \left[ 1 - \cos \theta \cos \theta_r(t) - \sin \theta \sin \theta_r(t) \cos(\phi - \phi_r(t)) \right] \quad (9)$$

For the small amplitude random oscillation condition investigated in this study, a small angle approximation is adopted. By neglecting second and higher order infinitesimal terms, the expression can be simplified into the following three dimensional clearance distribution model:

$$h_r(\theta, \phi, t) \approx h_0 + \Delta R \cdot \sin \theta [\theta_{rx}(t) \cos \phi + \theta_{ry}(t) \sin \phi] \quad (10)$$

The main flow passage of the internal push point-the-bit rotary steerable tool is a circular straight pipe, and its flow state directly determines the system pressure drop characteristics, thereby affecting the pressure difference across the spherical seal and the resulting leakage rate. Based on the operating parameters considered in this study, the calculated mean flow velocity in the main passage indicates a fully turbulent flow regime. Therefore, the standard RNG  $k$ - $\varepsilon$  turbulence model is employed for the numerical analysis (Yakhot and Orszag 1986; Ma et al. 2026).

Assuming that the drilling fluid is an incompressible Newtonian fluid and neglecting the effects of body forces and energy dissipation, the continuity equation and the Reynolds-averaged momentum equation can be expressed as:

$$\nabla \cdot \mathbf{u} = 0 \quad (11)$$

$$\rho \frac{\partial \mathbf{u}}{\partial t} + \rho \left[ (\mathbf{u} - \mathbf{u}_g) \cdot \nabla \right] \mathbf{u} = -\nabla p + \nabla \cdot \left( \mu \left( \nabla \mathbf{u} + (\nabla \mathbf{u})^T \right) - \rho \overline{\mathbf{u} \mathbf{u}'} \right) \quad (12)$$

where  $\rho$  is the density of the drilling fluid, kg/m<sup>3</sup>, and  $\mathbf{u}_g$  is the moving-mesh velocity vector, m/s.

Transport equation for turbulent kinetic energy:

$$\frac{\partial(\rho k)}{\partial t} + \nabla \cdot (\rho \bar{\mathbf{u}} k) = \nabla \cdot \left[ \left( \mu + \frac{\mu_t}{\sigma_k} \right) \nabla k \right] + G_k - \rho \varepsilon \quad (13)$$

Transport equation for the dissipation rate:

$$\frac{\partial(\rho \varepsilon)}{\partial t} + \nabla \cdot (\rho \bar{\mathbf{u}} \varepsilon) = \nabla \cdot \left[ \left( \mu + \frac{\mu_t}{\sigma_\varepsilon} \right) \nabla \varepsilon \right] + C_1 \varepsilon \frac{G_k}{k} - C_2 \rho \frac{\varepsilon^2}{k} \quad (14)$$

The turbulent viscosity is defined as:

$$\mu_t = \rho C_\mu \frac{k^2}{\varepsilon} \quad (15)$$

where  $\mathbf{u}$  is the fluid velocity, m/s, and  $\mu$  is the dynamic viscosity of the fluid, Pa·s. In the standard RNG  $k$ - $\varepsilon$  turbulence model, the model constants are  $C_1 = 1.44$ ,  $C_2 = 1.92$ ,  $C_\mu = 0.09$ ,  $\sigma_k = 1.00$ , and  $\sigma_\varepsilon = 1.30$ . The governing equations follow the conventional two-equation turbulence-model framework (Launder and Spalding 1974; Yakhot and Orszag 1986).

A significant velocity gradient and flow boundary layer exist near the wall of the spherical leakage passage. To improve the numerical accuracy of the near wall flow solution, the standard wall function method is adopted to correct the wall shear stress, velocity distribution, and turbulence parameters, thereby ensuring the numerical stability and computational accuracy of the wall bounded flow (Launder and Spalding 1974; Ma et al. 2026). Fig.5 shows a schematic illustration of the near wall flow profile based on the wall function method.

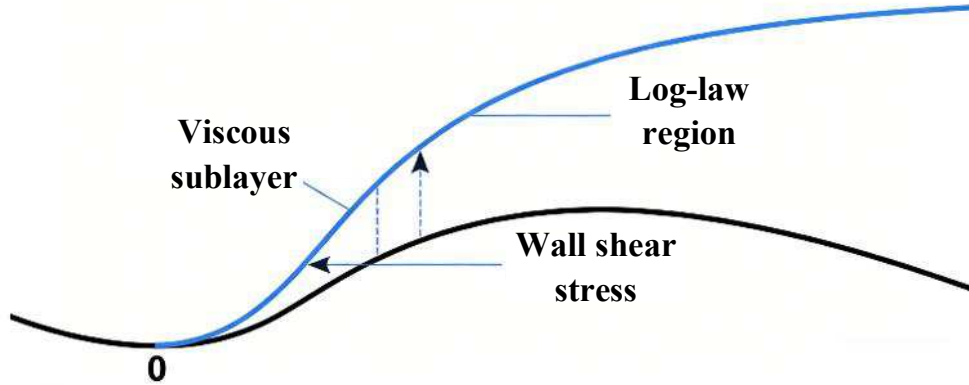


**Fig. 5** Near wall flow at a solid wall using the wall function method.

## 2.3 Numerical method

For the leakage flow through the spherical micro clearance, the extremely small flow channel dimensions and the steep pressure and velocity gradients make the numerical results particularly sensitive to mesh resolution. This issue is particularly important in close-clearance seal simulations because the characteristic gap dimension is considerably smaller than the overall computational domain (van der Walt et al. 2024; Zhang et al. 2021). To avoid significant numerical deviations caused by differences in mesh density, a mesh independence study was conducted prior to the

formal simulations. Mesh independence assessment is commonly used to quantify the sensitivity of CFD predictions to spatial discretization and to ensure that further mesh refinement does not materially alter the predicted quantities (Celik et al. 2008). The leakage rate was selected as the evaluation criterion, and several mesh schemes with different levels of refinement were established to investigate the influence of mesh size on the predicted leakage rate. On this basis, a mesh scheme that simultaneously satisfies the required computational accuracy and provides a reasonable computational efficiency was selected for the subsequent simulations.

To determine an optimal mesh that balances computational accuracy and efficiency, four mesh schemes with different levels of refinement were designed for the mesh independence study, with the leakage rate of the spherical sealing system selected as the key evaluation parameter. The number of mesh elements and the computational time for each case were also recorded to assess the influence of mesh resolution on the numerical results. Table 1 summarizes the predicted leakage rates and computational times for the four mesh schemes.

**Table 1** Leakage rates and computational times for different mesh schemes.

| Parameter                           | Scheme 1 | Scheme 2 | Scheme 3 | Scheme 4 |
|-------------------------------------|----------|----------|----------|----------|
| Number of domain mesh elements      | 341189   | 667968   | 2060198  | 6060106  |
| Computational time / min            | 21.3     | 24.6     | 46.6     | 272.9    |
| Leakage rate / ( $L \cdot s^{-1}$ ) | 0.311    | 0.268    | 0.262    | 0.266    |
| Relative error / %                  | 16.9     | 0.7      | 1.5      | -        |

Based on the simulation results corresponding to the different mesh schemes, the result obtained using the finest mesh, Scheme 4, was taken as the reference. The leakage rates predicted by the other mesh schemes were then compared with this reference value. The relative errors of Schemes 2 and 3 were both below 3%, indicating that further mesh refinement had only a negligible effect on the predicted leakage rate and that mesh independence had essentially been achieved. However, the computational time increased substantially with further mesh refinement. Considering both numerical accuracy and computational efficiency, Scheme 2 was therefore selected for the subsequent numerical simulations.

A numerical model of the spherical leakage test device was established in COMSOL Multiphysics. The computational domain was discretized using the built in meshing module, with local mesh refinement applied to regions with complex flow channel geometries. The resulting mesh is shown in Fig.6.

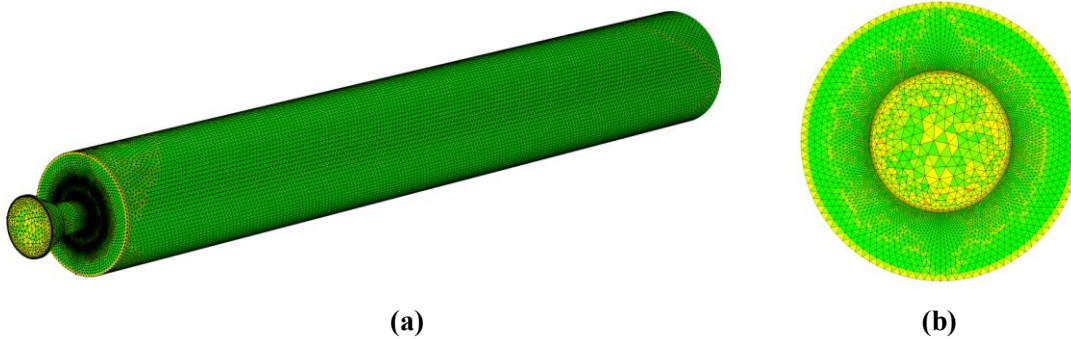


**Fig. 6** Schematic diagram of the mesh generation results. (a) overall mesh; (b) leakage gap flow domain.

## 2.4 Simulation conditions

The RNG  $k-\varepsilon$  turbulence model was employed for the numerical simulations. Drilling fluid was introduced from the left side of the computational domain, where a flow rate inlet boundary condition was specified. Two outlets were defined on the right side of the computational domain, corresponding to the leakage flow through the spherical clearance and the main flow through the center tube, respectively. Pressure outlet boundary conditions were imposed at both outlets. The leakage outlet is connected to the external wellbore and was therefore assigned a pressure of 0 Pa, whereas the main flow outlet pressure was set to 4 MPa according to the prescribed pressure at the drill bit located downstream of the rotary steerable tool. The flow of drilling fluid through the internal passages of the device was described by the Navier–Stokes equations. Similar CFD-based approaches have been successfully employed to resolve pressure-driven and dynamically disturbed flows in annular and mechanical sealing systems (Zhang et al. 2021;

Zahorulko et al. 2023a, b; Ma et al. 2026). The corresponding computational model and boundary conditions are shown in Fig.7.

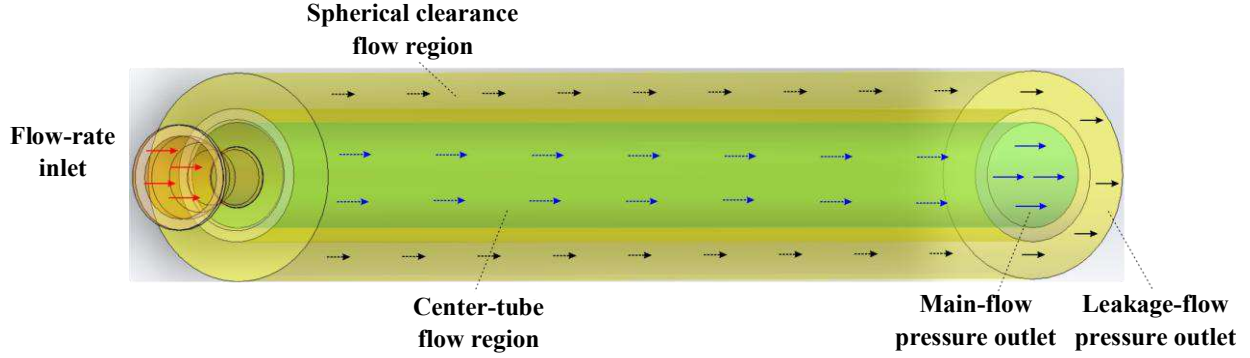


**Fig. 7** Computational model of the spherical leakage device.

### 3 Results and discussion

To investigate the effects of different operating parameters on the leakage rate and pressure drop of the device under non steering conditions, three simulation schemes were designed to represent the random oscillation environment of the system. In each scheme, the oscillation points were randomly generated in three dimensional space, and the spatial oscillation range of the mandrel was controlled within 0–1° according to the actual operating condition. Fig.8 shows the randomly generated oscillation points for the three schemes.

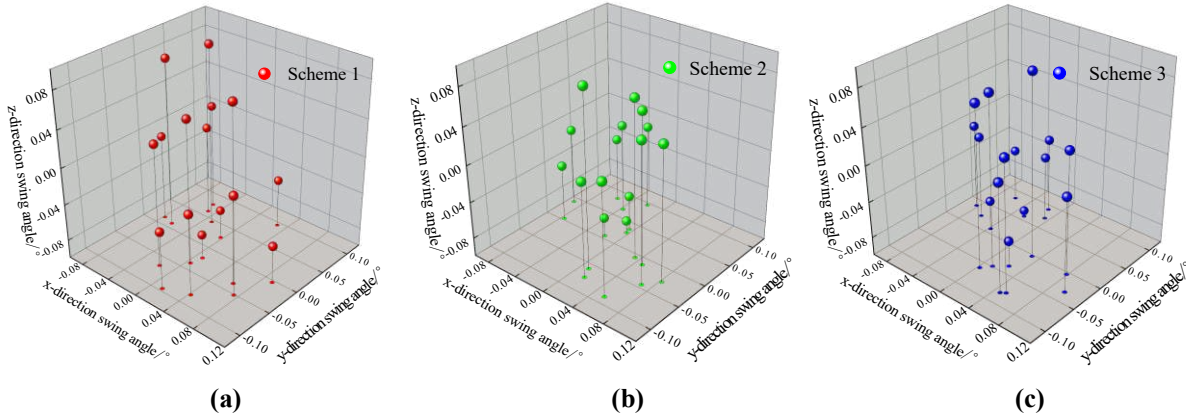


**Fig. 8** Random oscillation points generated for different schemes. (a) scheme 1; (b) scheme 2; (c) scheme 3.

To clarify the effects and underlying mechanisms of the key operating parameters on sealing performance, and to characterize the leakage behavior and pressure drop response of the device under non steering conditions, the spherical normal clearance, drilling fluid density, and drilling fluid flow rate were selected as the primary influencing factors. The effects of these parameters on the leakage magnitude, leakage fluctuations, and pressure drop were quantitatively analyzed.

#### 3.1 Effect of mandrel–seal clearance

Clearance is one of the key geometric parameters governing leakage through sealing interfaces, and previous numerical and experimental studies have consistently reported a strong dependence of leakage on gap size (Anbarsooz et al. 2022; Zhang et al. 2023; Son et al. 2025; Z. Wang et al. 2026). Numerical simulations were therefore conducted to investigate the influence of different clearance values on the leakage rate under random oscillation conditions. The corresponding simulation parameters are listed in Table 2.

**Table 2** Leakage rates and computational times for different mesh schemes.

| Scheme | Inlet flow rate / ( $\text{L} \cdot \text{s}^{-1}$ ) | Clearance / mm | Drilling fluid density / ( $\text{g} \cdot \text{cm}^{-3}$ ) | Jet end face diameter / mm |
|--------|------------------------------------------------------|----------------|--------------------------------------------------------------|----------------------------|
| 1      | 30                                                   | 0.1            | 1.75                                                         | 20                         |

|   |    |     |      |    |
|---|----|-----|------|----|
| 2 | 30 | 0.2 | 1.75 | 20 |
| 3 | 30 | 0.3 | 1.75 | 20 |

The influence of random oscillation on the leakage rate under different clearances is shown in Fig.9. Figs.9(a)–9(c) present the leakage rates calculated using three random oscillation schemes at different clearance values. In these figures, the light yellow  $xOy$  plane represents the mean leakage rate for each simulation group. Fig.9(d) shows the deviation of the leakage rate at each oscillation point from the corresponding mean leakage rate.

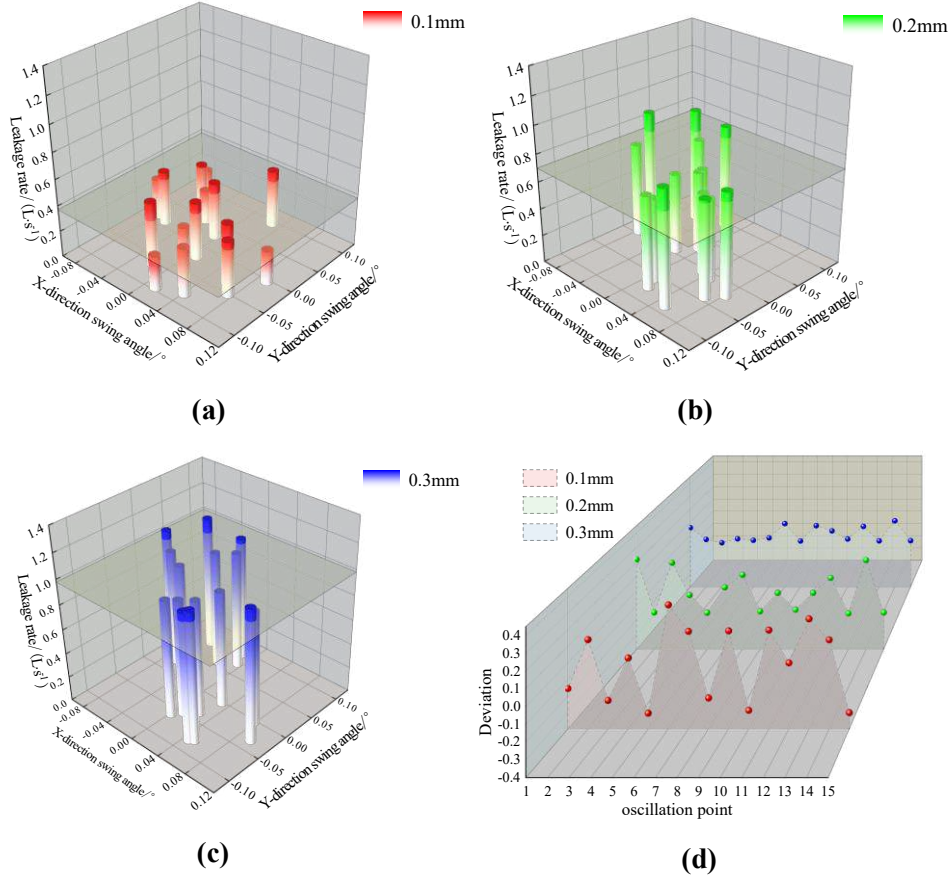


**Fig. 9** Leakage rate and its fluctuation under random oscillation at different clearances. (a) 0.1mm clearance; (b) 0.2mm clearance; (c) 0.3mm clearance; (d) leakage deviation at different clearances.

With respect to the effect of clearance on leakage, increasing the spherical normal clearance increases the mean leakage rate of the device. This increase in mean leakage with clearance is qualitatively consistent with previous investigations of clearance type sealing systems (Anbarsooz et al. 2022; Zhang et al. 2023; Son et al. 2025). In contrast, the reduction in relative leakage fluctuation observed here is associated with the random-oscillation-induced redistribution of the spherical clearance. This indicates that, under small clearance conditions, the leakage rate is more sensitive to local clearance disturbances and therefore exhibits stronger fluctuations. As the clearance between the sealing element and the mandrel increases, the relative variation in clearance caused by the same oscillation amplitude becomes smaller, resulting in a more stable opening of the leakage flow passage. Consequently, the local throttling effect induced by random oscillation is weakened, and the deviation of the leakage rate from its mean value decreases.

Fig.10 shows the internal flow channel structure of the jet piston. The pressure drop in the vicinity of the jet piston is evaluated by calculating the average pressures at the upstream and downstream end faces of the flow passage.

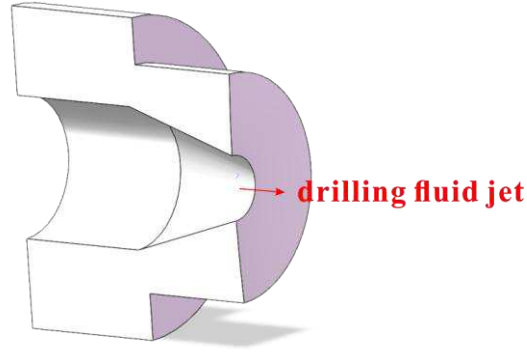


**Fig. 10** Internal flow channel structure of the jet piston.

The pressure drop and its fluctuation under random oscillation at different clearances are shown in Fig.11. Fig.11(a)–11(c) present the pressure drop values calculated using three random oscillation schemes at different clearance values. In these figures, the light yellow  $xOy$  plane represents the mean pressure drop for each simulation group. Fig.11(d) shows the deviation of the pressure drop at each oscillation point from the corresponding mean pressure drop. Previous studies of annular and labyrinth seals have shown that pressure losses are strongly associated with local contraction, expansion, and throttling effects in sealing passages (Asok et al. 2008; Zahorulko et al. 2023a).

Regarding the effect of clearance on pressure drop, the clearance between the mandrel and the sealing element is located upstream of the jet piston. Variations in the clearance size therefore have no significant effect on the pressure drop across the jet piston. The mean pressure drop at the jet piston is approximately 8.05 MPa, and the deviation of the pressure drop at each oscillation point from the corresponding mean value remains below 1%.

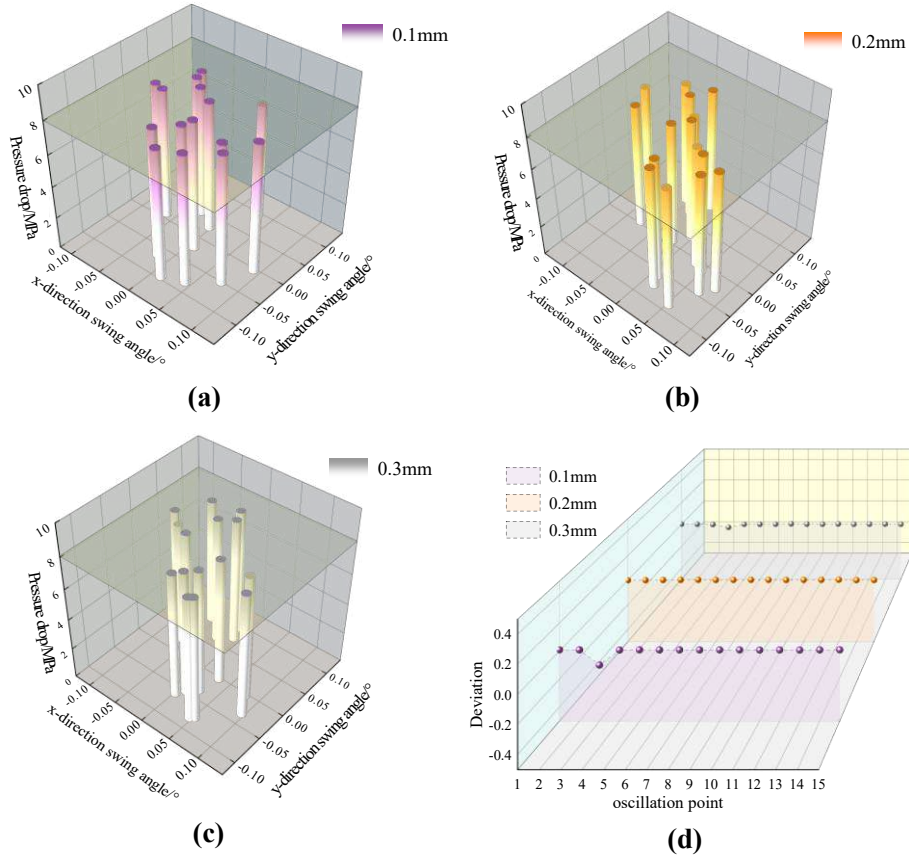


**Fig. 11** Pressure drop and its fluctuation under random oscillation at different clearances. (a) 0.1mm clearance; (b) 0.2mm clearance; (c) 0.3mm clearance; (d) pressure drop deviation at different clearances.

### 3.2 Effect of drilling-fluid density

Based on the three random oscillation schemes, the drilling fluid density was varied to investigate its influence on the leakage rate under random oscillation conditions. The corresponding simulation parameters are listed in Table 3.

**Table 3** Simulation parameters for different drilling-fluid densities.

| Scheme | Inlet flow rate / ( $\text{L}\cdot\text{s}^{-1}$ ) | Clearance / mm | Drilling fluid density / ( $\text{g}\cdot\text{cm}^{-3}$ ) | Jet end face diameter / mm |
|--------|----------------------------------------------------|----------------|------------------------------------------------------------|----------------------------|
| 1      | 30                                                 | 0.1            | 1.75                                                       | 20                         |
| 2      | 30                                                 | 0.1            | 1.4                                                        | 20                         |
| 3      | 30                                                 | 0.1            | 1.05                                                       | 20                         |

The influence of random oscillation on the leakage rate and its fluctuation at different drilling fluid densities is shown in Fig.12. The annotations in each subfigure are the same as those in Figs.9(a)–9(d).

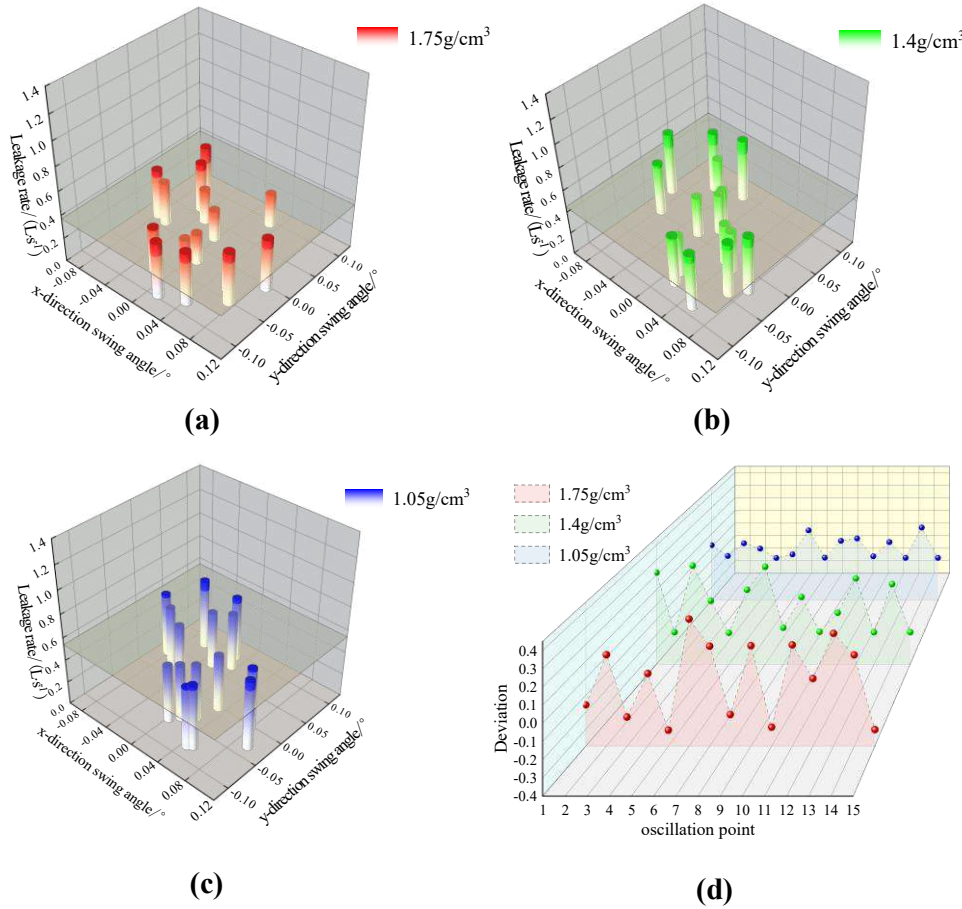


**Fig. 12** Leakage rate and its fluctuation under random oscillation at different drilling fluid densities. (a) 1.75  $\text{g}/\text{cm}^3$ ; (b) 1.4  $\text{g}/\text{cm}^3$ ; (c) 1.05  $\text{g}/\text{cm}^3$ ; (d) leakage deviation at different densities.

Regarding the effects of drilling fluid density on the leakage rate and its fluctuation characteristics, the instantaneous leakage rate exhibits larger deviations from the mean value at different oscillation points when a high density drilling fluid is used, indicating that higher fluid density increases the sensitivity of the local clearance flow to random oscillation. As the drilling fluid density decreases, the fluctuation in leakage rate is reduced. Previous studies of annular sealing systems have shown that fluid density can significantly affect the hydraulic loading and dynamic response within narrow sealing passages (Li et al. 2021; Zahorulko et al. 2023b). In the present spherical sealing system, leakage fluctuations are jointly governed by the local clearance disturbance induced by random oscillation and the inertial characteristics of the drilling fluid. A high density drilling fluid produces a stronger response to local clearance

disturbances, whereas a low density drilling fluid exhibits weaker inertial effects, causing the instantaneous leakage rate to remain closer to its mean value.

The pressure drop and its fluctuation under random oscillation at different clearances are shown in Fig.13. The annotations in each subfigure are the same as those in Figs.11(a)–11(d).

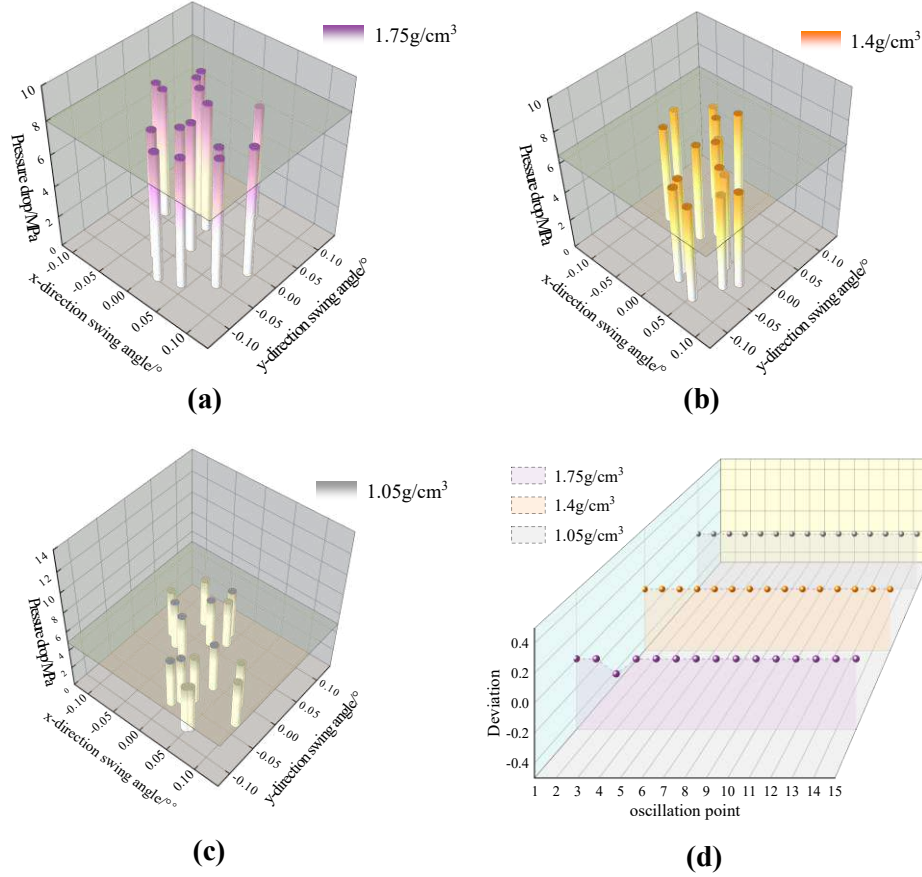


**Fig. 13** Pressure drop and its fluctuation under random oscillation at different drilling-fluid densities. (a)  $1.75\text{g/cm}^3$ ; (b)  $1.4\text{g/cm}^3$ ; (c)  $1.05\text{g/cm}^3$ ; (d) pressure drop deviation at different densities.

Regarding the effect of drilling-fluid density on the pressure drop and its fluctuation characteristics, variations in fluid density mainly affect the magnitude of the pressure drop across the jet piston, while the deviations of the instantaneous pressure drop at different oscillation points from the corresponding mean value remain nearly unchanged. This indicates that changes in drilling-fluid density have little influence on pressure-drop fluctuations. Previous seal-flow studies have demonstrated that variations in fluid density can modify the absolute hydraulic loading and fluid-induced forces within sealing passages (Li et al. 2021; Zahorulko et al. 2023b). However, in the present spherical sealing system, the relative pressure-drop fluctuation remains nearly insensitive to density, indicating that the local clearance disturbance induced by random oscillation is the dominant factor governing the fluctuation component..

### 3.3 Effect of flow rate

The inlet flow rate at the jet piston was varied, and three schemes with different flow rates were used to investigate their effects on the leakage rate and its fluctuation characteristics. The corresponding simulation parameters are listed in Table 4.

**Table 4** Simulation parameters for different drilling fluid flow rates

| Scheme | Inlet flow rate / ( $\text{L}\cdot\text{s}^{-1}$ ) | Clearance / mm | Drilling fluid density / ( $\text{g}\cdot\text{cm}^{-3}$ ) | Jet end face diameter / mm |
|--------|----------------------------------------------------|----------------|------------------------------------------------------------|----------------------------|
| 1      | 30                                                 | 0.1            | 1.75                                                       | 20                         |
| 2      | 20                                                 | 0.1            | 1.75                                                       | 20                         |
| 3      | 10                                                 | 0.1            | 1.75                                                       | 20                         |

The influence of random oscillation on the leakage rate and its fluctuation at different flow rates is shown in Fig. 14. The annotations in each subfigure are the same as those in Figs.9(a)–9(d).

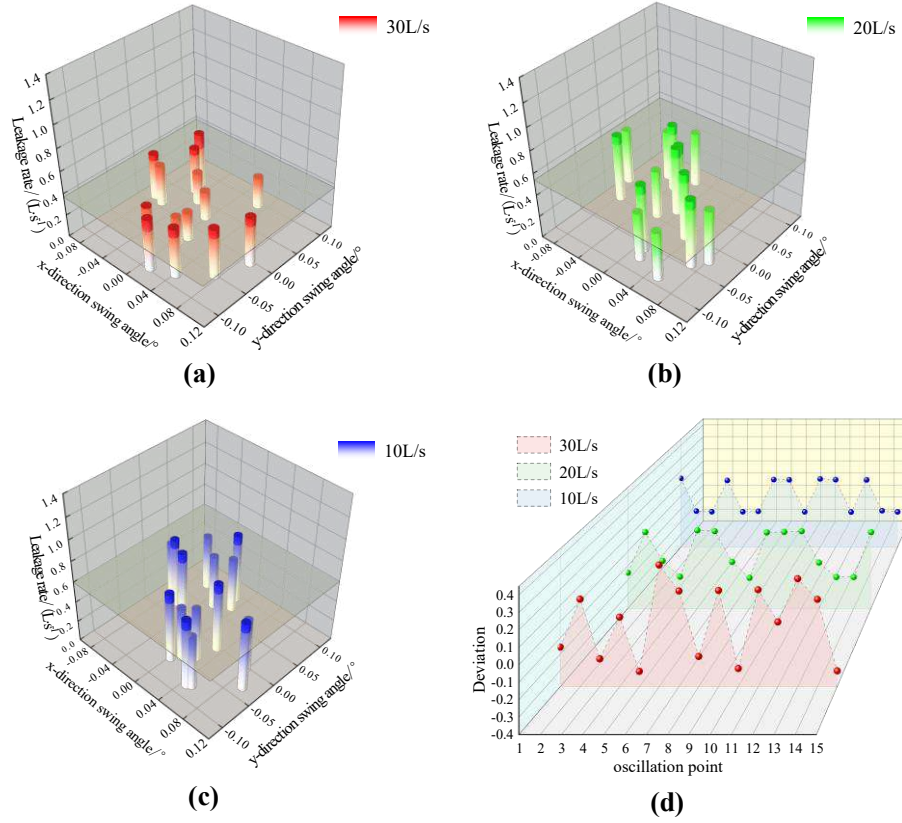


**Fig. 14** Leakage rate and its fluctuation under random oscillation at different flow rates. (a) 30L/s; (b) 20L/s; (c) 10L/s; (d) leakage deviation at different flow rates.

Regarding the effects of flow rate on the leakage rate and its fluctuation characteristics, the deviation of the leakage rate at each oscillation point from the corresponding mean value decreases as the flow rate is reduced, indicating a lower level of leakage fluctuation. Previous experimental and numerical investigations have demonstrated that flow rate is closely associated with leakage magnitude and local hydraulic losses in clearance-type sealing passages (Asok et al. 2008; Son et al. 2025). In the present spherical sealing system, decreasing the flow rate weakens the contraction effect in the jet-piston flow passage and reduces the influence of random oscillation on local leakage disturbances. Therefore, the drilling-fluid flow rate affects not only the leakage magnitude but also the fluctuation response of the spherical clearance flow.

The pressure drop and its fluctuation under random oscillation at different flow rates are shown in Fig.15. The annotations in each subfigure are the same as those in Figs.11(a)–11(d).

Regarding the effects of flow rate on the pressure drop and its fluctuation characteristics, the flow rate mainly affects the magnitude of the pressure drop across the jet piston, while the deviations of the instantaneous pressure drop at different oscillation points from the corresponding mean value remain nearly unchanged. This indicates that the flow rate has little influence on the relative pressure-drop fluctuation. The increase in pressure-drop magnitude with increasing flow rate is qualitatively consistent with previous experimental and numerical observations for pressure-driven sealing passages (Asok et al. 2008; Zahorulko et al. 2023a). Combined with the variation in leakage rate, a higher flow rate enhances the contraction effect in the jet-piston flow passage, making the associated pressure-drop variation more pronounced. In contrast, under low-flow-rate conditions, this contraction effect is significantly weakened. The nearly unchanged relative fluctuation further suggests that the fluctuation component is governed primarily by the oscillation-induced local clearance disturbance rather than by the bulk flow rate itself.

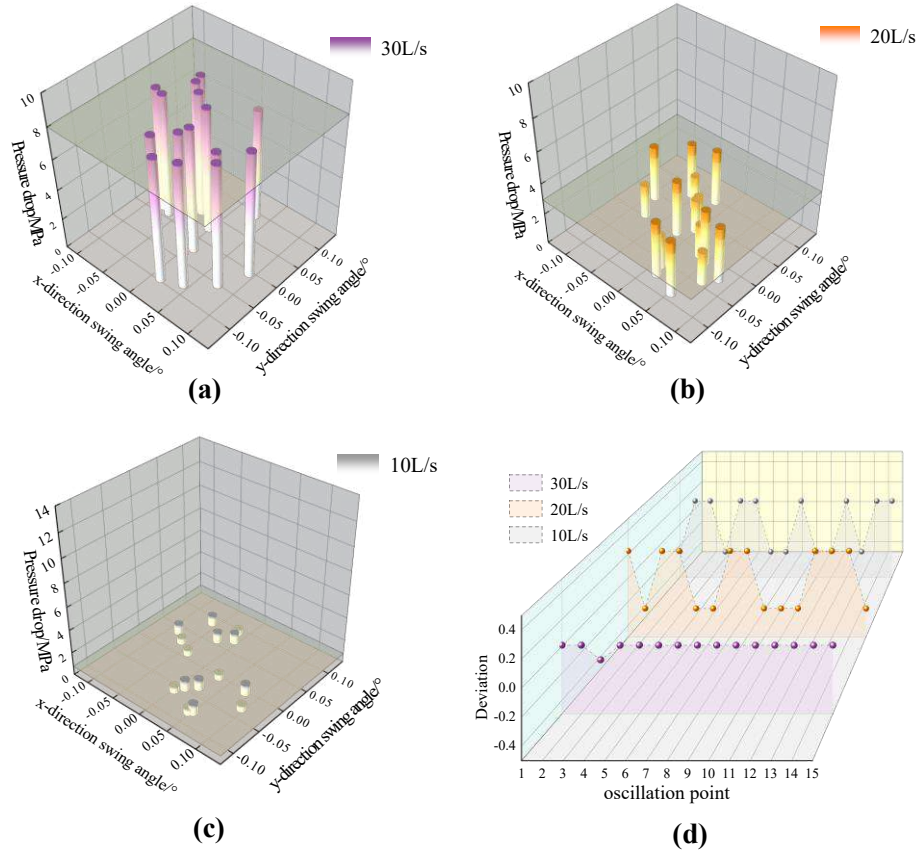


**Fig. 15** Pressure drop and its fluctuation under random oscillation at different flow rates. (a) 30L/s; (b) 20L/s; (c) 10L/s; (d) Pressure drop deviation at different flow rates.

## 4 Conclusions

- (1) For the spherical sealing structure of the adjustment mechanism in the internal push point-the-bit rotary steerable tool, the differences in mandrel motion between guided oscillation and non-steering conditions were clarified. Under non steering conditions, the mandrel undergoes small-amplitude spatial random oscillations due to the combined effects of assembly clearances, rotational dynamics, and bearing constraints. These oscillations keep the flow field within the spherical sealing clearance in a continuously unsteady state and constitute the primary source of dynamic leakage fluctuations.
- (2) Based on spherical geometry and micro clearance leakage theory, calculation models were established for mandrel oscillation, spherical clearance distribution, and transient leakage rate. The results show that guided oscillation produces a relatively stable leakage state, whereas random oscillation alters the local clearance opening and causes the leakage rate to fluctuate with the oscillation position. By incorporating the RNG  $k-\epsilon$  turbulence model, the relationship between the pressure drop in the main flow passage and leakage through the spherical clearance can be further analyzed.
- (3) The parametric analysis shows that increasing the spherical normal clearance increases the mean leakage rate but reduces the deviation of the leakage rate from its mean value. Drilling-fluid density mainly affects the absolute magnitude of the pressure drop, while its influence on pressure-drop fluctuations is relatively weak. A reduction in flow rate significantly increases the leakage rate but decreases the leakage fluctuation. These results can provide a basis for optimizing the spherical sealing structure and improving the stability of the hydraulic control system during non-steering hold-section drilling.

## Declarations

**Author contributions:** Zixiao Chen performed the methodology development, numerical simulations, data analysis, and manuscript preparation. Gonghui Liu contributed to the conceptualization, supervision, project administration, funding acquisition, and manuscript revision. Lincong Wang contributed to the methodology, investigation, validation,

and manuscript revision. Shichang Li contributed to data curation and visualization. Tian Chen contributed to formal analysis and manuscript revision. All authors read and approved the final manuscript.

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**Data availability:** The data supporting the findings of this study are available from the corresponding author upon reasonable request.

**Conflict of interests:** The authors declare that there are no conflicts of interest.

**Ethics approval:** Not applicable. This study did not involve human participants, human data, or animals.

**Consent to participate:** Not applicable.

**Consent for publication:** All authors have reviewed the manuscript and approved its publication.

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## Supplementary information

### List of symbols

The symbols and acronyms used throughout this study are summarized below.

#### Latin letters

|           |                                                                                          |
|-----------|------------------------------------------------------------------------------------------|
| $C_1$     | Model constant in the RNG $k - \varepsilon$ turbulence model                             |
| $C_2$     | Model constant in the RNG $k - \varepsilon$ turbulence model                             |
| $C_\mu$   | Turbulent-viscosity model constant                                                       |
| $dQ$      | Differential leakage rate, $m^3 \cdot s^{-1}$                                            |
| $f_1$     | Guided-oscillation frequency, $Hz$                                                       |
| $G_k$     | Production term of turbulent kinetic energy, $kg \cdot m^{-1} \cdot s^{-3}$              |
| $h$       | Local clearance thickness, $m$                                                           |
| $h_0$     | Initial uniform clearance thickness, $m$                                                 |
| $k$       | Turbulent kinetic energy, $m^2 \cdot s^{-2}$                                             |
| $p$       | Fluid pressure, $Pa$                                                                     |
| $Q_0$     | Baseline leakage rate when the mandrel is coaxial with the tool axis, $m^3 \cdot s^{-1}$ |
| $Q_1(t)$  | Instantaneous leakage rate under guided oscillation, $m^3 \cdot s^{-1}$                  |
| $R$       | Radius of curvature of the spherical sealing surface, $m$                                |
| $t$       | Time, $s$                                                                                |
| $u$       | Fluid velocity vector, $m \cdot s^{-1}$                                                  |
| $u_g$     | Moving-mesh velocity vector, $m \cdot s^{-1}$                                            |
| $x, y, z$ | Cartesian coordinate directions                                                          |

#### Greek letters

|               |                                                                 |
|---------------|-----------------------------------------------------------------|
| $\alpha_1$    | Lower polar-angle limit of the spherical sealing surface, $rad$ |
| $\alpha_2$    | Upper polar-angle limit of the spherical sealing surface, $rad$ |
| $\Delta p$    | Pressure difference across the sealing surfaces, $Pa$           |
| $\varepsilon$ | Turbulent kinetic-energy dissipation rate, $m^2 \cdot s^{-3}$   |
| $\mu$         | Dynamic viscosity of the drilling fluid, $Pa \cdot s$           |

|                      |                                                                             |
|----------------------|-----------------------------------------------------------------------------|
| $\mu_t$              | Turbulent dynamic viscosity, $Pa \cdot s$                                   |
| $\rho$               | Drilling-fluid density, $kg \cdot m^{-3}$                                   |
| $\sigma_k$           | Turbulent Prandtl number for $k$                                            |
| $\sigma_\varepsilon$ | Turbulent Prandtl number for $\varepsilon$                                  |
| $\theta$             | Polar angle of an arbitrary point on the spherical sealing surface, $rad$   |
| $\theta_l(t)$        | Instantaneous deflection angle under guided oscillation, $rad$              |
| $\theta_{max}$       | Maximum guided-oscillation angle, $rad$                                     |
| $\theta_r(t)$        | Instantaneous random-oscillation deflection angle, $rad$                    |
| $\theta_{rx}(t)$     | Random-oscillation angle component in the $x$ -direction, $rad$             |
| $\theta_{ry}(t)$     | Random-oscillation angle component in the $y$ -direction, $rad$             |
| $\Phi$               | Azimuth angle of an arbitrary point on the spherical sealing surface, $rad$ |
| $\Phi_r(t)$          | Instantaneous azimuth angle of random oscillation, $rad$                    |

### Acronyms

|     |                         |
|-----|-------------------------|
| BHA | Bottom hole assembly    |
| RNG | Renormalization group   |
| RSS | Rotary steerable system |