Study on pressure-controlled sliding sleeve of jet breaking for natural gas hydrate mining based on throttle pressure drop principle

Yang Tang\textsuperscript{1,2,3,4}, Jiaxin Yao\textsuperscript{1,4}, Yin He\textsuperscript{1,4}, Peng Sun\textsuperscript{1,4}, Xin Jin\textsuperscript{1}

\textsuperscript{1}School of Mechatronic Engineering, Southwest Petroleum University, Chengdu, China
\textsuperscript{2}State Key Laboratory of Oil and Gas Reservoir Geology and Exploitation, Southwest Petroleum University, Chengdu, China
\textsuperscript{3}Key Laboratory of Oil & Gas Equipment, Ministry of Education (Southwest Petroleum University), Chengdu, China
\textsuperscript{4}Guangdong Provincial Laboratory of Southern Marine Science and Engineering (Zhanjiang), Zhanjiang, China

Abstract

In order to reduce operating cost of deep-sea natural gas hydrate (NGH) solid-state fluidization mining (SSFM), a device for controlling opening and closing of jet nozzles is required. It is required to meet the specifications of work process and is not affected by changes in water depth and well depth. Therefore, a design scheme of a pressure-controlled sliding sleeve (PCSS) based on the principle of throttling pressure drop is proposed. The nozzles opening and closing process is controlled by flow rate control and is not affected by environmental pressure. Using computational fluid dynamics software, the corresponding control equations are solved by finite volume method and the pressure drop and axial force generated by different flow rates of drilling fluid flowing through different structural sliding cores are obtained. According to simulation analysis results and actual working conditions, a relatively good set of sliding core structural parameters is preferred. According to structural parameters of this group, the prototype of PCSS was machined and throttle pressure drop principle verification experiment and section pressure difference principle exclusion experiment were carried out to verify the feasibility of the design of PCSS. Through this research, it can help to promote the application of the throttling pressure drop principle in controllable oil drilling and completion tools.

KEYWORDS
jet breaking, natural gas hydrate, pressure-controlled sliding sleeve, solid-state fluidization mining, throttle pressure drop

This is an open access article under the terms of the Creative Commons Attribution License, which permits use, distribution and reproduction in any medium, provided the original work is properly cited.
© 2020 The Authors. Energy Science & Engineering published by the Society of Chemical Industry and John Wiley & Sons Ltd.
1 | INTRODUCTION

At present, there are many ways to exploit NGH. For exploitation of NGH with deep seabed depth and poor cementation, the traditional mining method may lead to the disordered decomposition of submarine NGH in the seabed. It affects the stability of the seabed hydrate reservoir, thereby inducing geological disasters. Based on the above reasons, the process of SSFM is proposed, as shown in Figures 1 and 2. The core of the process is to use the stability of NGH at seafloor temperature and pressure to break up the NGH deposit into fine particles. The fine particles are mixed with seawater and transported to the offshore platform by closed pipes and then postprocessed on the offshore platform. Therefore, a new type of natural gas hydrate jet crushing tool is needed to realize that the jet nozzles can be opened and closed according to the specific process requirements in the drilling operation. It ensures the mechanical crushing and drilling operation and the jet breaking and expanding operation to automatically switch continuously to reduce the number of drills and increase NGH drilling efficiency.

In this paper, a PCSS is innovatively designed by using the throttling pressure drop generated by the variable-section tube to meet the demand of NGH SSFM process.

In recent years, the research on the principle of variable cross-sectional tube and throttling pressure drop mainly focuses on the cavitation phenomenon in the throttling channel and the two-phase flow in the micropipeline with sudden expansion and contraction. Abdelall et al. studied the pressure drop caused by sudden expansion and contraction in the single-phase flow and gas-liquid two-phase flow under the near atmospheric pressure. Roul and Dash used the Euler-Eulerian model to calculate the pressure drop of sudden expansion tube and sudden shrinkage tube and established a relationship between the two-phase flow pressure drop based on the numerical results and the experimental results. The flow of fluid in the throttling channel in the presence of cavitation was studied by Lomakin et al. Chen et al. reviewed the existing efforts concerning two-phase flow across sudden expansions/contractions and to examine the applicability of the existing correlations with respect to the recent data in small channels. Shi et al. carried out experimental and numerical studies on variable-profile tubes of different shapes, and developed and verified a semi-empirical model for predicting the hollowing of variable-section tubes. Most of the above are theoretical and experimental studies on the pressure drop and cavitation caused by single-phase flow or gas-liquid two-phase flow when flowing through a variable cross-sectional tube. There are few studies on the effect of variable cross-sectional tube on the effect of throttling pressure drop due to geometrical dimensional changes. The purpose of the paper is to study the pressure drop caused by the change in geometrical dimension or the flow rate of the inner part of the sliding core which is the main components inside the PCSS. According to the analysis results, the PCSS test prototype was machined to verify the correctness of the theoretical design and the feasibility of using it in practical work.

The paper is organized as follows. In Section 1, we propose a structural scheme of PCSS based on the SSFM process and the throttling pressure drop principle and the theoretical analysis of the throttling pressure drop principle is carried out in Section 2. In Section 3, the flow field simulation of the key components of the PCSS is carried out, and the simulation results are discussed. Based on the simulation results, the relatively optimal sliding core structural parameters are chosen. Section 4 of this paper is mainly to test the PCSS by experiments and analyze and discuss the experimental results. Finally, Section 5 provides some discussions and conclusions.

2 | PRINCIPLE ANALYSIS OF THROTTLING PRESSURE DROP

When the fluid flows through a sudden expansion tube and sudden shrinkage tube, due to the inertial force of the fluid flow, the mainstream is separated from the wall surface, and a vortex zone is formed between the mainstream and the wall surface, as shown in Figures 3 and 4. The vortex
movement aggravates the turbulence of the fluid and increases energy loss. At the same time, the vortex zone and the mainstream zone continue to exchange masses, and the vortex motion particles are brought downstream by the mainstream, which intensifies the mainstream turbulence intensity in a certain downstream range, thereby further increasing the energy loss.\textsuperscript{12} When the incompressible fluid flows through the sudden expansion tube, according to the one-dimensional momentum and mechanical energy conservation equation, the total irreversible pressure drop is as follows:\textsuperscript{8}

$$
\Delta P_e = P_{2,1} - P_{2,3} = \Delta P_{e,R} + \Delta P_{e,I} = \rho L \langle u_t \rangle^2 \left( \sigma ^2 k_{d3} - \sigma k_{d1} \right)
$$

where $\Delta P_e$ is the pressure drop of the sudden expansion tube; $P_{2,1}$ is the pressure of the small diameter tube section in the sudden expansion tube; $P_{2,3}$ is the pressure of large diameter tube.
section in the sudden expansion tube; $\Delta P_{eR}$ is the reversible pressure drop; $\Delta P_{eI}$ is the irreversible pressure drop; $\rho_L$ is the fluid density; $u_1$ is the inlet flow velocity of the small diameter tube section; $\sigma$ is the turbulent Prandtl number; $k_{d1}$ is the momentum correction coefficient at the inlet; $k_{d3}$ is the momentum correction coefficient at the outlet.

$$\Delta P_{eI} = K_e\rho \frac{\langle u_1 \rangle^2}{2}$$  \hspace{1cm} (2)

$$K_e = 1 - 2k_{d1}\sigma + \sigma^2(2k_{d3} - 1)$$  \hspace{1cm} (3)

where $K_e$ is the loss coefficient; $\sigma = A_1/A_3$; $A_1$ is the cross-sectional area of the flow path at sections 1-1; $A_3$ is the cross-sectional area of the flow path at 3-3.

$$k_d = \frac{\langle u^2 \rangle}{\langle u \rangle^2}$$  \hspace{1cm} (4)

$$\Delta P_{e,R} = -\frac{\langle u_1 \rangle^2}{2}(1 - \sigma^2)$$  \hspace{1cm} (5)

Assumption: $k_{d1} = k_{d3} = 1$, then Equation (3) becomes:

$$K_e = (1 - \sigma)^2$$  \hspace{1cm} (6)

In the single-phase flow of the incompressible fluid through the sudden shrinkage tube, as shown in Figure 4. The fluid acceleration on the left side of the section 2-2 can be approximated to an isentropic state. The loss of mechanical energy mainly occurs in the deceleration zone after the shrinkage section 2-2, so the pressure drop caused by the section shrinkage can be expressed as:

$$\Delta P_e = P_{2,3} - P_{2,1} = \Delta P_{e,R} + \Delta P_{e,I}$$

$$= \frac{\langle u_1 \rangle^2}{2} \left[ 1 - \beta_3\sigma^2C_c^2 - 2C_c + 2C_c^2k_{d1} \right]$$  \hspace{1cm} (7)

where $\Delta P_e$ is pressure drop of the sudden shrinkage tube; $P_{2,1}$ is the pressure of the small diameter tube section in the sudden shrinkage tube; $P_{2,3}$ is the pressure of large diameter tube section in the sudden shrinkage tube; $\Delta P_{e,R}$ is the reversible pressure drop; $\Delta P_{e,I}$ is the irreversible pressure drop; $\sigma$ is the turbulent Prandtl number; $k_{d1}$ is momentum correction coefficient at the entrance of the sudden shrinkage tube; $\beta_3$ is the kinetic energy correction coefficient at the outlet; $C_c$ is the contraction coefficient.

$$\beta = \frac{\langle u^3 \rangle}{\langle u \rangle^3}$$  \hspace{1cm} (8)

$$C_c = \frac{A_3}{A_1}$$  \hspace{1cm} (9)

In the above formula, $A_1$ is the cross-sectional area of the flow path at sections 1-1; $A_3$ is the cross-sectional area of the flow path at 3-3; In deriving Equation (8), it is assumed that the velocity is uniformly distributed over the contraction section. When the velocity is assumed to be evenly distributed in sections 1-1 and 3-3, Equation (7) becomes:

$$\Delta P_e = \frac{\langle u_1 \rangle^2}{2} \left[ \left( 1 - \frac{1}{C_c} \right)^2 + 1 - \sigma^2 \right]$$  \hspace{1cm} (10)

For laminar flow in a circular tube, $\beta = 2$ and $k_d = 1.33$; if the inside of the circular tube is turbulent, $k_d \approx \sigma \approx 1$. The contraction coefficient is a function of the Reynolds number. Therefore, the axial force generated by the internal pressure drop of the sliding core is:

$$F = \Delta P_e(A_1 - A_2)$$  \hspace{1cm} (11)

### 3 | NUMERICAL SIMULATION ANALYSIS OF THE PCSS

#### 3.1 | Structure and working principle of the PCSS

The PCSS is mainly composed of an outer cylinder, a sliding core, a spring, an overcurrent connector and 24 jet nozzles, as shown in Figure 4. Its working principle is mainly based on the NGH drilling process shown in Figures 1 and 2.
PCSS uses the drilling fluid to flow through the sliding core to generate local resistance loss and drive along the path to drive it to work. Its working state is related to the flow rate of drilling fluid that is introduced into the PCSS and will not change due to changes in environmental pressure. Its working process can be divided into the following two stages.

Stage 1: Horizontal well borehole drilling

This stage is the normal drilling stage, so the flow of drilling fluid is small and the nozzle on the PCSS will be closed. The drilling fluid will flow to the drill bit through the PCSS to provide the drilling fluid required for drilling.

Stage 2: Jet breaking

At this stage, the horizontal well drilling has been completed, so it is necessary to pull back the PCSS to further expand the breaking space. Therefore, the drilling fluid flow rate needs to be increased to drive the sliding core compression spring, thereby opening the nozzle and blocking the drilling fluid passage into the drill bit. Large flow of drilling fluid will be ejected at high speed from the nozzle of the PCSS to expand the breaking diameter (Figure 5).

3.2 Establishment of numerical simulation model for the sliding core

The structure of the sliding core is shown in Figure 6A. The simplified flow field calculation domain is mainly divided into three types as shown in Figure 6B. The basic structural dimensions of the sliding core are shown in Table 1. During the flow field simulation of the sliding core, the specific variables of the simulation are shown in Table 2.

3.3 Theoretical analysis of numerical simulation for internal flow field in the slide core

3.3.1 Fluid control equation for the sliding core

During the flow of the drilling fluid inside the sliding core of the PCSS, it is necessary to observe the basic laws of conservation of energy, such as conservation of energy, conservation of momentum, and conservation of mass. Based on these basic laws of conservation of physics, there are three basic equations in computational fluid dynamics, namely continuity equations (mass conservation equations), momentum equations (motion equations or Navier-Stokes equations, abbreviated as N-S equations), and energy equation. These equations reflect the basic motion of the fluid in the flow field. Now, the flow action inside the sliding core is assumed as follows:

1. Fluid is an incompressible Newtonian fluid.
2. The physical properties of the fluid inside the sliding core remain unchanged.
3. The flow process is an isothermal process, so the energy equation is not needed.

Based on the above assumptions, the basic governing equation inside the sliding core is as follows:

1. Fluid is an incompressible Newtonian fluid.
2. The physical properties of the fluid inside the sliding core remain unchanged.
3. The flow process is an isothermal process, so the energy equation is not needed.

FIGURE 5 Structure diagram of PCSS
Continuity equation:
\[
\frac{\partial \rho}{\partial t} + \rho \frac{\partial (\rho u)}{\partial x} + \rho \frac{\partial (\rho v)}{\partial y} + \rho \frac{\partial (\rho w)}{\partial z} = 0
\]  

Momentum equation:
\[
\begin{cases}
\rho \frac{du}{dt} + \text{div}(\rho uv) = \text{div}(\mu \text{grad}u) - \frac{\partial p}{\partial x} + S_u \\
\rho \frac{dv}{dt} + \text{div}(\rho vv) = \text{div}(\mu \text{grad}v) - \frac{\partial p}{\partial y} + S_v \\
\rho \frac{dw}{dt} + \text{div}(\rho ww) = \text{div}(\mu \text{grad}w) - \frac{\partial p}{\partial z} + S_w
\end{cases}
\]  

Turbulence model:
There are many mathematical models for turbulence calculation in the simulation calculation process. It mainly includes Spalart-Allmaras model, \(k-\varepsilon\) model, \(k-\omega\) model, Reynolds stress model (RSM), and large-eddy simulation (LES).\(^{20,21}\) However, the Spalart-Allmaras model is mainly suitable for aerodynamic flow problems and is not suitable for solving shear flow and wall flow problems.\(^{22}\) The \(k-\omega\) model is mainly applied to the simulation of flow phenomena such as curvature flow, separation flow, and jet flow, and the calculation results are relatively difficult to converge.\(^{20}\) Large-eddy simulation is mainly used for thermal fatigue, vibration, and buoyancy flow of ships.\(^{23}\)

Therefore, the turbulence model applicable to the calculation of the internal flow of the sliding core mainly includes the RSM and the \(k-\varepsilon\) model. However, the RSM takes into account the rapid changes in streamlined bending, eddies, rotation, and tension. It has the potential for higher accuracy predictions for complex flows, but it consumes a lot of computing resources and requires high-quality computational grids.\(^{24}\) The \(k-\varepsilon\) model is a semi-empirical formula summed up from experimental phenomena. It has a wide range of applications, economy, and reasonable precision. It is widely used in engineering flow field calculation.\(^{25,26}\) Because the flow of drilling fluid inside the sliding core is relatively simple and this paper mainly focuses on engineering practical problems, the calculation accuracy of \(k-\varepsilon\) model can fully meet the demand. Therefore, based on the above factors, the \(k-\varepsilon\) turbulence model is selected. The transport equations of \(k\) and \(\varepsilon\) are:
\[
\frac{\partial (\rho k)}{\partial t} + \frac{\partial (\rho ku_i)}{\partial x_i} = \frac{\partial}{\partial x_i} \left( \mu + \frac{\mu_t}{\sigma_k} \right) \frac{\partial k}{\partial x_i} + G_k + G_b - \rho \varepsilon - Y_M + S_k
\]
\[
\frac{\partial (\rho \varepsilon)}{\partial t} + \frac{\partial (\rho \varepsilon u_i)}{\partial x_i} = \frac{\partial}{\partial x_i} \left( \mu + \frac{\mu_t}{\sigma_{\varepsilon}} \right) \frac{\partial \varepsilon}{\partial x_i} + C_{1\varepsilon} \frac{\varepsilon}{k} (G_k + G_{3\varepsilon} G_b) - C_{2\varepsilon} \frac{\varepsilon^2}{k} + S_{\varepsilon}
\]
where \(G_k\) is the turbulent energy generation term caused by the mean velocity gradient; \(G_b\) is the turbulent energy generation term caused by buoyancy; \(Y_M\) is the pulsation expansion term in the compressible turbulence; \(C_{1\varepsilon}, C_{2\varepsilon}\), \(C_{3\varepsilon}\) are empirical constants; \(\sigma_k\) is the number of Prandtl with the kinetic energy \(k\); \(\sigma_{\varepsilon}\) is the Prandtl number corresponding to the dissipation rate \(\varepsilon\); \(S_k\) and \(S_{\varepsilon}\) are user-defined source terms.

The calculation formulas for each of the Equations (15) and (16) are as follows.

First, for the generation term of turbulent energy \(G_k\) caused by the mean velocity gradient is:
\[
G_k = \mu_t \left( \frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} \right) \frac{\partial u_i}{\partial x_j}\]
where \(\mu_t\) is turbulent viscosity coefficient and can be expressed as a function of \(k\) and \(\varepsilon\).
\[
\mu_t = \rho C_{\mu} \frac{k^2}{\varepsilon}
\]
where \(C_{\mu}\) is the empirical constant.

Because \(B\) is the generation term of the turbulent energy caused by buoyancy; \(Y_M\) is the pulsation expansion term in the compressible turbulent flow. In this paper, it is assumed that the drilling fluid for simulation is an incompressible fluid, thus \(G_b=0, Y_M=0\).

In the standard \(k-\varepsilon\) model, according to the recommended values of Launder et al, and subsequent experimental verification, the values of the model constants \(C_{1\varepsilon}, C_{2\varepsilon}, C_{3\varepsilon}\), \(\sigma_k\) and \(\sigma_{\varepsilon}\) are:
\[
C_{1\varepsilon} = 1.44, C_{2\varepsilon} = 1.92, C_{\mu} = 0.09, \sigma_k = 1.0, \sigma_{\varepsilon} = 1.3
\]

\(C_{3\varepsilon}\) is the coefficient related to the buoyancy in the calculation of the compressible fluid flow. When the main flow direction is parallel to the gravity direction, there is \(C_{3\varepsilon} = 1\), and when the main flow direction is perpendicular to the gravity direction, there is \(C_{3\varepsilon} = 0\).\(^{27}\)

Based on the above analysis, when user-defined source items are not considered, the \(k-\varepsilon\) model for incompressible fluids is:
\[
\frac{\partial (\rho k)}{\partial t} + \frac{\partial (\rho k u_i)}{\partial x_i} = \frac{\partial}{\partial x_i} \left( \mu + \frac{\mu_t}{\sigma_k} \right) \frac{\partial k}{\partial x_i} + G_k - \rho \varepsilon
\]
\[
\frac{\partial (\rho \varepsilon)}{\partial t} + \frac{\partial (\rho \varepsilon u_i)}{\partial x_i} = \frac{\partial}{\partial x_i} \left( \mu + \frac{\mu_t}{\sigma_{\varepsilon}} \right) \frac{\partial \varepsilon}{\partial x_i} + C_{1\varepsilon} \frac{\varepsilon}{k} (G_k + C_{3\varepsilon} G_b) - C_{2\varepsilon} \frac{\varepsilon^2}{k} + S_{\varepsilon}
\]
Boundary conditions and wall treatment of flow field analysis model

The inlet of the sliding core adopts the speed inlet boundary condition. It is mainly suitable for incompressible flow, as it allows the stagnation condition to float, and if it is used for compressible flow, it will result in nonphysical results. The outlet adopts the pressure outlet boundary condition. The wall adopts nonslip wall boundary conditions. Since the wall roughness has a great influence on the simulation results when simulating the turbulent flow with wall limitation, it is necessary to modify the roughness of the wall law to consider the influence of the wall roughness. For rough walls, the wall law of average velocity is:

\[ \frac{u^*u^*}{\tau_w/\rho} = \frac{1}{\tau} \ln \left( \frac{E\rho u^* y_p}{\mu} \right) - \Delta B \]  

(20)

where \( u^* = C_{\mu}^{1/4} k^{1/2} \), \( \tau_w \) is the wall shear stress; \( \mu \) is the hydrodynamic viscosity; \( E \) is the empirical constant, usually \( E \) takes 9.81; \( u^* \) is the time-averaged velocity parallel to the wall at a certain point; \( y_p \) is the distance from a point to the wall; \( k \) is the turbulent energy; \( \Delta B \) is the roughness function, which measures the transfer of the cutoff point due to the influence of the roughness. The magnitude of the value depends mainly on the type and size of the roughness. Related studies have shown that \( \Delta B \) is not a single-valued function of the dimensionless rough height \( K^*_y \), but rather has a different form depending on the value of \( K^*_y \).

1. Liquid dynamic smooth: \( K^*_y < 3.5 \)
2. Transition zone: \( 3.5 < K^*_y < 70 - 90 \)
3. Full-rough: \( K^*_y > 70 - 90 \)

According to the above data, the influence of roughness is negligible in the smooth region, but the influence of the roughness in the transition region and the completely rough region is getting larger and larger. In FLUENT, the roughness function is mainly derived from the data of Nikuradse, proposed by Cebeci and Bradshaw.

For liquid dynamic smooth zone \( (K^*_y < 2.25) \):

\[ \Delta B = 0 \]  

(21)

For the transition zone \( (3.5 < K^*_y < 70 - 90) \):

\[ \Delta B = \frac{1}{\kappa} \ln \left( \frac{K^*_y - 2.25}{87.25} + C_{K^*_y} K^*_y \right) + \sin \left[ 0.4258 (\ln K^*_y - 0.811) \right] \]  

(22)

where \( C_{K^*_y} \) is a rough constant, depending on the type of roughness, and \( \kappa \) is the Karman constant.

In a completely rough area \( (K^*_y > 70 - 90) \):

\[ \Delta B = \frac{1}{\kappa} \ln \left( 1 + C_{K^*_y} K^*_y \right) \]  

(23)

In the simulation process, to simulate the influence of wall roughness, two parameters must be specified: rough height \( K^*_y \) and roughness constant \( C_{K^*_y} \). When \( K^*_y = 0 \), the default wall surface is a smooth wall surface. Because the roughness height is mainly related to the processing technology of the inner surface, the influence of wall facing pressure drop should be considered in the simulation process. Therefore, the rough height of the inner wall surface of the
sliding core is uniformly set to 0.025 mm. In order to satisfy the use of the $k-\varepsilon$ turbulence model, Nikuradse resistance data can be reproduced in a tube filled with fluid of the same roughness, and the roughness constant $C_{K_s}$ takes the default value of 0.5.29

3.4 | Numerical simulation result

The throttling pressure drop principle of the sliding core is analyzed in detail, and it mainly includes the change in the size of the sliding core inlet and outlet, the change in the taper of the sliding core inlet, and the influence of the change in the internal flow rate of the sliding core in the same size on the internal pressure drop and axial force of the sliding core. The results are shown in Figures 7-14.

In the same flow rate and outlet diameter, the internal pressure and axial force of the sliding core are changed with the inlet diameter as shown in Figures 7 and 8. As the diameter of the sliding core inlet increases, the pressure drop inside the sliding core is also gradually increased; when the diameter of the sliding core inlet is 30 mm, the inner part of the sliding core is a straight tube section. The pressure drop inside the sliding core is 0.068 MPa, and the axial force is 107.6 N. When the diameter of the sliding core inlet section is 35 mm, and the sliding core is caused by sudden contraction, the section pressure is rapidly increased to 0.174 MPa, and the corresponding axial force is also increased to 275 N. When the diameter of the sliding core is 50 mm, the pressure drop inside the sliding core is 0.306 MPa, and the axial force is 483 N. With the diameter of the sliding core inlet increasing, the pressure drop and axial force in the sliding core also continue to increase, but the growth rate gradually decreases.

Under the same outlet and inlet diameter and flow conditions, the pressure and axial force changes inside the sliding

**FIGURE 7** Internal pressure of the sliding core changes with the $D_i$ value

**FIGURE 8** Internal pressure drop and axial force of the sliding core changes with the $D_i$ value

**FIGURE 9** Internal pressure of the sliding core changes with the $\theta$ value

**FIGURE 10** Internal pressure drop and axial force of the sliding core changes with the $\theta$ value
When the sliding core inlet angle is 10°, the pressure drop inside the sliding core is 0.163 MPa and the axial force is 301 N. When the sliding core inlet angle is 180°, the pressure drop inside the sliding core is 0.279 MPa and the axial force is 513 N. The relative growth rate of the pressure drop in the sliding core is the largest when the sliding core inlet angle is increased from 10° to 30°. With the angle of the sliding core inlet increasing, the pressure drop inside the sliding core and the axial force also increase and the growth rate of the pressure drop in the sliding core gradually decreases.

When the inlet angle is less than 180°, the sliding core can be regarded as a sudden shrinkage tube. In the sudden shrinkage tube, accelerated decompression along the path, the fluid particle is subjected to a positive pressure consistent with the flow direction and the velocity is continuously increased. Therefore, the vortex region does not appear in the reducer. If the angle of contraction is large, due to inertia, a small vortex area will also be generated at the back of the tube.

When the sliding core outlet diameter is 50 mm, the inlet angle is 180°, and the flow rate is 800 L/min, the relationship between the outlet diameter of the sliding core and the internal pressure and axial force is shown in Figures 11 and 12. When the sliding core outlet diameter is 30 mm and the total pressure drop inside the sliding core is 0.31 MPa, the sliding core reverse force is 0 N and the total axial force is 483.9 N. When the outlet diameter is 40 mm, the total pressure drop inside the sliding core rises to 0.36 MPa, and the sliding core has a reverse force of 69 N and a total axial force of 484.4 N. When the sliding core outlet diameter reaches 50 mm, the total internal pressure drop of the sliding core is reduced to 0.32 MPa, and the sliding core has a reverse force of 3.2 N and the total axial is 492 N. With the diameter of the sliding core outlet increasing, the positive axial force and the total axial force of the sliding core gradually increase, while the reverse axial force increases first and then decreases.
Through the above analysis results, a set of sliding core parameters is selected as shown in Table 3. Considering that under the actual working conditions, the required flow rate during normal drilling is 400 L/min, and the opening flow rate of the PCSS nozzles is 800 L/min. Under the condition of meeting the actual working conditions, when drilling fluid through the sliding core, the axial force generated by the drilling fluid is required as large as possible, because when the nozzles are turned on, drilling fluid flows out of the nozzles and the axial drilling fluid flow path at the bottom end of the sliding core will be blocked. Because the bottom of the sliding core is sealed by the end face sealing method, when nozzles are opened, the axial force generated by the drilling fluid flowing through the sliding core is larger, and the sealing effect of the lower end surface is better.

When the size of the sliding sleeve is constant, the relationship between the flow rate of the sliding sleeve and the internal pressure drop and axial force of the sliding sleeve is shown in Figures 13 and 14. When the internal flow rate of the sliding core is 100 L/min, the internal pressure drop of the sliding core is 0.004 MPa and the axial force is 7 N. When the internal flow of the sliding core reaches 1000 L/min, the pressure drop inside the sliding core reaches 0.333 MPa and the axial force is 611 N. As the flow rate of the sliding core increases, the pressure drop and axial force inside the sliding core also increase.

4 | EXPERIMENTAL TEST OF THE PCSS

In order to verify the feasibility of the PCSS, an experimental prototype of the PCSS was machined according to the simulation results, as shown in Figure 15. The throttle pressure drop principle verification experiment and the section pressure difference principle exclusion experiment of the PCSS were carried out by using the experimental devices in Figure 15. In the experiment, clean water was used instead of drilling fluid for experiment, the specific experimental equipment models and parameters are shown in Table 4.

4.1 | Verification experiment of throttle pressure drop principle

The experimental process is shown in Figure 16. The experiment uses a multistage pump system to provide drilling...
fluid for the PCSS. The multistage pump absorbs water from water tank through throttle valve II, electromagnetic flow regulating valve, throttle valve III, and intake pipeline into the PCSS and then returns to the water tank through backwater pipeline. During the experiment, a portion of the multistage pump discharge liquid was reintroduced into the water tank by adjusting the electronic flow regulating valve on the bypass to control the flow rate of the drilling fluid flowing into the PCSS. The sensor used to measure the data during the experiment is shown in Figure 17. In order to improve measurement accuracy and reduce data delay, the flow rate is measured during the experiment using an ultrasonic flow sensor. In order to measure the actual pressure drop change inside the PCSS, HBck-804 pressure

| Serial number | Working condition           | Specific parameters | Parameter and device name                  | Specific parameters |
|---------------|----------------------------|---------------------|-------------------------------------------|---------------------|
| 1             | Normal drilling (L/min)    | 0–400               | Experimental normal working flow (L/min)  | ≤400                |
| 2             | Jet breaking (L/min)       | 800                 | Nozzles are opened (L/min)               | ≥800                |
| 3             | Nozzles are closed (L/min)| <400                | Spring free height (mm)                  | 220                 |
| 4             | Spring wire diameter (mm) | 5                   | Pressure test pressure (MPa)             | 40                  |
| 5             | Spring height (mm)         | 220                 | Power drill test bench                   |                     |
| 6             | Spring preload height (mm)| 174                 | Electric pressure test pump              | 3DY-400/60          |

**FIGURE 16** Schematic diagram of the experimental devices
During the experiment, in order to ensure the accuracy of the measured data and slow the delay of data transmission delay, when adjusting the flow rate of the PCSS, it should be adjusted by means of segmentation adjustment, which is mainly divided into four stages as follows.

Stage 1: When the flow rate is less than 400 L/min, the opening of the electronic flow regulating valve can be increased by 5% each time and kept for 5 seconds. The flow to be displayed is stable, and the corresponding experimental data and experimental phenomena can be observed and recorded before the adjustment can be continued.

Stage 2: When the flow rate is greater than 400 L/min and less than 700 L/min, the opening of the electronic flow regulating valve can be increased by 3% each time and kept for 8 seconds; when the flow to be displayed is stable, we can continue to adjust the flow rate after the corresponding experimental data and experimental phenomena can be observed and recorded.

Stage 3: When the inlet flow rate is greater than 700 L/min, the opening of the electronic flow regulating valve can be increased by 1% each time and kept for 10 seconds. After the flow rate is stable, the adjustment can be continued until the nozzles are fully opened, and the flow rate of the return line is 0 L/min.

Stage 4: According to the above steps, reduce the flow rate of the drilling fluid flowing into the PCSS, and observe the relevant data and experimental phenomena. The comparison between the data measured by multiple experiments and the simulation data is shown in Figure 18. When the flow rate of the PCSS is 400 L/min, the pressure drop is 0.062 MPa. The difference between the experimental value and the simulated value is 0.024 MPa. At this time, the nozzles are in a closed state and the flow inside the nozzles of the PCSS mainly flows out from the bottom end of the PCSS and enters the water tank through the backwater pipeline of the test system. However, there is a small amount of leakage at the nozzles due to the sealing problem. When the flow rate of the PCSS reaches 550 L/min, the measured pressure drop is 0.142 MPa, and the corresponding axial force is 224.72 N. At this time, the flow from the nozzles gradually increases. The flow from the lower end of the PCSS begins to decrease. As the internal flow rate of the PCSS is further increased, the flow rate discharged from the nozzles is gradually injected, and the reduction flow rate of the lower end of the PCSS has also been further increased. When the flow rate of the PCSS reaches 833 L/min, the water column ejected from the nozzles is in a stable
TANG et al. state and the flow rate from the bottom end of the PCSS is gradually reduced to 0 L/min. The measured pressure drop reached 0.609 MPa, and the difference from the simulated value was 0.14 MPa. The axial force corresponding to the experiment also reached 963.78 N and the difference from the simulated value reached 216.28 N.

It can be seen from Figure 18 that with the increase in the flow rate of the drilling fluid inside the PCSS, the simulated and experimental values of the pressure drop and the axial force are gradually increased and the trend of the curve is about the same. However, as the flow rate increases, the difference between the simulated value and the experimental value gradually increases. The main reason for this trend is that the simulation value is mainly for the simulation of the pressure drop inside the sliding core to establish a simulation model. The experimental measurement value mainly measures the difference between the pressure transmitters at both ends of the PCSS. When the nozzles are fully opened and the bottom channel of the sliding core is blocked, the measured value of the pressure transmitter at the bottom channel of the PCSS will approach 0 MPa. The pressure measured by the experiment is mainly the pressure inside the PCSS, so when the internal flow of the PCSS reaches 833 L/min, the bottom channel of the sliding core is sealed. When the bottom channel of the PCSS is blocked, the pressure drop will rise suddenly. It can be seen from the experimental results that the actual flow rate of the nozzles fully opened is 833 L/min, and the error between it and the theoretical design value is 4.13%, which is within the acceptable error range (Figure 19).

4.2 Section pressure difference principle exclusion experiment

This experiment mainly verifies whether the sliding core inside the PCSS will move due to the change in its
ambient pressure. The 3DY400/60 electric pressure test pump mainly used in the experiment provides the required experimental pressure for the experiment. The maximum pressure that can be reached is 60 MPa, and the experimental medium is clean water. The experimental steps are as follows.

Step 1: Using the plugging head to block the bottom outlet of the PCSS and connect the electric test pump to the upper end of the pressure-controlled sliding sleeve.
Step 2: Adjusting the flow control valve on the test pump to control the pressure, so that the pressure loading gradient is maintained at 5 MPa/time, and the voltage is stabilized for 2 minutes after each loading.
Step 3: Recording experimental data and corresponding experimental phenomena.
Step 4: Repeating the above experimental steps twice to ensure the accuracy of the experimental results.

Because there was a leak at each joint during the experiment, the pressure could not continue to increase when the internal pressure of the PCSS reached 38 MPa. The pressure has exceeded the actual working pressure. When the pressure value reached about 38 MPa, there was a small amount of leakage due to the sealing problem at the joint of the joint and no water was sprayed out at the nozzles. Therefore, the injection process of the PCSS nozzles is not affected by the pressure environment in which it is placed. (Table 5).

5 CONCLUSIONS

Aiming at the characteristics of SSFM technology, this paper innovatively designed a PCSS for NGH SSFM using the principle of throttling pressure drop. The numerical simulation method was used to analyze the flow field of the sliding core of one of the key components inside the PCSS. The PCSS experimental prototype was machined according to the preferred parameters. The experimental bench was built according to the actual working conditions of the PCSS, and the throttle pressure drop principle verification experiment and the section pressure difference principle exclusion experiment were carried out on the prototype, and the following conclusions were obtained.

1. The results of the throttle pressure drop principle verification experiment show that the full opening flow rate of the PCSS is 833 L/min. The error value is 4.13% compared with the design opening flow. After the nozzles are opened, it can be effectively closed when the drilling fluid flow reaches 300 L/min. This experiment effectively verifies the feasibility of the throttling pressure drop principle for controlling the opening and closing of the PCSS nozzles.
2. The experimental results of the section pressure difference principle show that due to the sealing problem, when the internal pressure of the PCSS reaches 38 MPa, there is a small amount of leakage at the nozzles but it is not opened, and the pressure has exceeded the actual working
pressure. The experiment proves that the driving method of the PCSS sliding core is different from the driving method of the existing differential pressure sliding sleeve. The sliding core does not move due to changes in the ambient pressure.

3. It is feasible to drive the sliding core to control the opening and closing of the jet nozzles by the pressure drop generated by the throttling pressure drop principle, which can meet the demand of NGH solid fluidized mining operation. The use of PCSS will reduce the number of drills, the risk of drilling operations, and operating costs during the drilling process, and improve the safety of high drilling. The use of the throttling pressure drop control principle on the PCSS will drive its use in other downhole tools.

4. In the future research, the sealing problem of the PCSS existing in the experimental process and the stability of the sliding core caused by the drilling fluid flow will be further studied.

ACKNOWLEDGMENTS
This work is supported by the National Key Research and Development Program (2018YFC0310201), National Science and Technology Major Project (2016ZX05028-001-006), China Postdoctoral Innovative Talents Support Program (BX20190292), Scientific Research Starting Project of SWPU (No. 2018QHZ017), Open Fund of State Key Laboratory of Oil and Gas Reservoir Geology and Exploitation (Southwest Petroleum University) (PLN201827), and Miaozi Engineering Cultivation Project of Sichuan Science and Technology Department (No. 2019090).

CONFLICT OF INTEREST
The authors declare that there is no conflict of interests regarding the publication of this paper.

ORCID
Yang Tang https://orcid.org/0000-0001-7919-1409

REFERENCES
1. Chong ZR, Yang SHB, Babu P, Linga P, Li X-S. Review of natural gas hydrates as an energy resource: prospects and challenges. Appl Energy. 2016;162:1633-1652.
2. Gao Y, Yang M, Zheng J-N, Chen B. Production characteristics of two class water-excess methane hydrate deposits during depressurization. Fuel. 2018;232:99-107.
3. Shouwei Z, Wei C, Qingping LI. Solid state fluidized green mining technology for deep water shallow gas hydrate. China Offshore Oil Gas. 2014;5:1-7.
4. Sun J, Ning F, Liu T, et al. Gas production from a silty hydrate reservoir in the South China Sea using hydraulic fracturing: a numerical simulation. Energy Sci Eng. 2019. 7(4):1106-1122.
5. Xiao K, Zou C, Yang Y, Zhang H, Li H, Qin Z. A preliminary study of the gas hydrate stability zone in a gas hydrate potential region of China. Energy Sci Eng. 2019. https://doi.org/10.1002/ese.3.569
6. Wei N, Sun W, Meng Y, et al. Multiphase non equilibrium pipe flow behaviors in the solid fluidization exploitation of marine natural gas hydrate reservoir. Energy Sci Eng. 2018;6(6):760-782.
7. Abdelali FF, Hahn G, Ghiaasiaan SM, et al. Pressure drop caused by abrupt flow area changes in small channels. Exp Thermal Fluid Sci. 2005;29(4):425-434.
8. Roul MK, Dash SK. Two-phase pressure drop caused by sudden flow area contraction/expansion in small circular tubes. Int J Numer Meth Fluids. 2011;66(11):1420-1446.
9. Lomakin VO, Kuleshova MS, Kraeva EA. Fluid flow in the throttle channel in the presence of cavitation. Procedia Eng. 2015;106:27-35.
10. Chen IY, Wongsawat S, Yang B-C, Wang C-C. Two-phase flow across small sudden expansions and contractions. Heat Transfer Eng. 2010;31(4):12.
elbow—comparative analysis of the simulation with measurements results obtained by the ultrasonic flowmeter. *J Therm Sci.* 2018;27(05):11-18.

27. Fujun W. *Computational Fluid Dynamics—The Application of CFD Software Principles.* Beijing, China: Tsinghua University Press; 2004.

28. Růžička P. Modeling of boundary layer and the influence on heat transfer with help of CFD. *AIP Conf Proc.* 2018;2047(1).

29. Singh JP, Kumar S, Mohapatra SK. Modelling of two phase solid-liquid flow in horizontal pipe using computational fluid dynamics technique. *Int J Hydrogen Energy.* 2017;42(31):20133-20137.

**How to cite this article:** Tang Y, Yao J, He Y, Sun P, Jin X. Study on pressure-controlled sliding sleeve of jet breaking for natural gas hydrate mining based on throttle pressure drop principle. *Energy Sci Eng.* 2020;8:1422–1437. [https://doi.org/10.1002/ese3.616](https://doi.org/10.1002/ese3.616)