Effects of pulsating fluid at nozzle inlet on the output characteristics of common and self-excited oscillation nozzle

Chao Feng¹,² | Yu Wang¹,² | Lingrong Kong¹,²

Abstract
High-speed water jets are key technologies for increased production efficiency in oil drilling, and high-speed self-excited oscillating pulsed water jets (SEOPWs) offer many advantages compared to continuous jets for most industrial applications. Therefore, many studies have focused on this topic. However, the current studies on hydraulic pulsed jets have rarely extended to integrated modeling of the high-speed water jet transmission process. To better apply the SEOPWs, based on fluid mechanics, the theoretical model of the plunger pump–pipeline–nozzle is established, and the sensitivity of different nozzles on the pulsating fluid is studied in the time domain. Furthermore, the vibration characteristics of the nozzle are further investigated in the frequency domain. The results show that the common nozzle has an enlarged role in the pressure amplitude, the Helmholtz nozzle enlarges the pressure peak and amplitude, and the amplification is more obvious with the speedy growth on the plunger pump. However, as the pipeline becomes longer, the difference between the pressure of the inlet and outlet does not change. Therefore, the nozzle is more sensitive to the change in the plunger pump speed. In the frequency domain, the secondary frequency peak for the Helmholtz nozzle is 1.26 times larger than the interference frequency, while it is 1.08 times for the common nozzle. Helmholtz nozzles tend to produce stronger natural vibrations than common nozzles. By comparing the impact force, the experimental results show good agreement with the simulated results. The study provides a theoretical basis for water jet-assisted drilling in deep wells and lays the foundation for the better application of self-oscillating impact drilling tools.

Keywords
deep well drilling, Helmholtz nozzle, pulse water jet, transmission process
1 INTRODUCTION

Along with the finding of deep oil and gas formation in our country, especially the incremental amount of 5000–7000m deep wells and ultradeep wells in the Western district, the complex situation in the drilling process increases. With increasing well depth, the penetration rate decreases greatly. Hence, it is urgent to seek available cost-effective rock-breaking methods for improving the penetration rate in hard formations of deep wells.

It is worth mentioning that due to the stimulation of various industries, high-speed water jet technology has achieved significant progress in the past few decades. With its unique advantages of environmental protection and high efficiency, high-pressure water jets are widely used in energy industries for applications, such as quarry rock-breaking, coal seam drilling, oil drilling, geothermal exploitation, and deep-sea shallow drilling. In summary, the high-pressure water jet is considered one of the most promising assisted rock-breaking technologies and can further improve the rate of penetration. Thus, to better apply the water jet at drilling, it is essential to investigate the water jet transmission system. Moreover, the focus of this study is to provide a theoretical model of the water jet transmission system and to clarify the influence of construction parameters on the characteristics of the water jet outlet.

At present, many scholars have studied the output characteristics of plunger pumps, pipeline losses, and nozzle characteristics. Pei et al. installed piezoelectric accelerometers and inductive frequency-modulated displacement sensors on the valve disc under the action of hydraulic pressure and analyzed the motion parameters (acceleration and displacement) of the valve disc under different working conditions in real-time. Wang et al. established a mathematical model of pump exhaust valve movement and studied the effects of spring stiffness and valve quality on the movement behavior of the exhaust valve of a reciprocating piston pump. Furthermore, Zhang et al. based on the Euler two-fluid model, used ANSYS_CFX16.0 to simulate different inlet air gap fractions and different flow values and studied the pressure pulsation of a multiphase rotary dynamic pump with gas–water two-phase flow. In conclusion, scholars have studied the valve that affects the outlet characteristics, but few scholars have directly studied the effect of pump parameters on the outlet characteristics. Ma et al. deduced the relationship equation between the displacement and velocity of the piston of the multiphase pump and used the computational fluid dynamics method to study the entire working cycle of the reciprocating multiphase pump, focusing on the effect of the jet speed, a very strong pressure fluctuation may have the amplitude to reach 5.6 times that of the flow pressure due to the resonance coupling of the fluid and Helmholtz. Further studies on self-excited oscillating pulsed water jets (SEOPWs) found the mechanism of pulsed jets. Meanwhile, it laid the foundation for some scholars’ research on the influence of nozzle structural parameters on pulsation and frequency characteristics. The vortex structure in the Helmholtz oscillating nozzle and its changes over time have been discovered. In addition, some scholars have established mathematical models of self-excited oscillating water jet devices based on fluid network theory. The results show that when the jet frequency is close to the equipment’s inherent frequency, the jet peak pressure is the highest. Studies have discovered that area discontinuity at the nozzle inlet enhances peak pressure. Recently, some scholars used FLUENT (Ansys Academic Research Mechanical, Fluent 18.0) software to study the speed and pressure characteristics of the nozzle in a low-pressure state and analyzed important factors affecting the pulse jet.

It is worth noting that the Chinese scholar Li Xiaohong introduced the fluid vibration characteristics of the water jet transmission system through the transfer matrix method in the book “Water Jet Theory and Its Application in Mining Engineering.” In summary, when scholars study the influence of the water jet transmission system on output characteristics, they are more concerned with the mechanism of a certain device. As a
result, integrated modeling of the high-speed water jet transmission process and the outlet responses of different nozzles remains less studied.

Therefore, this study aims to study the effect of a pulsating fluid on the outlet characteristics of different nozzles through the overall model of the water jet transmission process (plunger pump–pipeline–nozzle). More specifically, the influence of the rotational speed of the plunger pump and the pipe length on the nozzle outlet characteristics was studied in the time domain, and the vibration characteristics of different nozzles were studied in the frequency domain. Finally, the theoretical model is verified by experimental work.

2 | MATHEMATICAL MODEL

2.1 | Integral description

During water jet-assisted rock-breaking, the mud is pressurized by a plunger pump and pumped into a tap, after which it passes through the bottom of the well drilling tool to reach the bottom of the well, thus cleaning the bottom of the hole and assisting in rock-breaking. Finally, the mud carries the rock debris back to the settling ponds on the surface. However, the actual water jet transmission process, with the complex structure of the drilling tool at the bottom of the well, is simplified. As shown in Figure 1, the water jet-assisted drilling system can be simplified into a system composed of plunger pumps, pipes, and nozzles.

Figure 1 is a schematic diagram of the operating principles of a Helmholtz oscillator. Generally, a typical Helmholtz nozzle is composed of an upstream nozzle, a downstream nozzle, and an oscillation chamber, as shown in the figure. It should be noted that on the inner side of the downstream nozzle there is a conical wall called an impinging edge. The optimum impinging angle of 120° has been proven experimentally. Moreover, a high-speed SEOPW disturbs the water jet due to the oscillating cavity, thus creating vorticities in the oscillating cavity. Due to the generation and disappearance of the vorticities, a water jet with pulsating characteristics is created.

As shown in Figure 1, the basic idea of the proposed model is to first focus on the transmission process of water jets and the output characteristics of nozzles in the context of pulsed rock-breaking and subsequently simplify the water jet transmission process in drilling systems before finally dividing the transmission system into four parts, which are superimposed to obtain an overall model of the transmission process.

The model primarily includes four steps (see Figure 1): (i) calculate the outlet characteristics of the plunger pump based on its construction characteristics (Section 2.2), (ii) calculate the pressure loss of pipelines using the analogy of the pipeline as a mass–spring system (Section 2.3), (iii) calculate the pressure loss and flow loss of a common nozzle based on fluid mechanics (Section 2.4), and (iv) create a transfer matrix with a Helmholtz nozzle (Section 2.5).

The following are noteworthy, in this study the plunger pump was studied without regard to leaks caused by the structure. To further minimize the computation, it was assume that the plunger pump outlet pipe is directly connected to the drill rod. Flow losses in the pipeline are not taken into account when modeling the pipeline. In addition, the oscillating cavity is assumed to be a point matrix and the effect of the impinging edge is not considered.

![Figure 1](image-url)
2.2 Mathematical model for pump

During the operation of a multiphase pump, the reciprocating motion of the piston is driven by the rotation of the prime mover through the crank link mechanism. The corresponding displacement and velocity of the piston can be expressed as:

\[
\begin{align*}
x_1 &= r \left(1 - \cos \theta + \frac{\lambda}{2} \sin^2 \theta\right), \\
v_1 &= \frac{dx_1}{dt} = \omega_1 t \left(\sin \theta + \frac{\lambda}{2} \sin 2\theta\right),
\end{align*}
\]

where \(x_1\) is the piston displacement, \(r\) is the crank radius, \(\theta = \omega_1 t\) is the crank angle, \(\lambda\) is the link ratio, \(v_1\) is the piston velocity, \(\omega_1 = \frac{n\pi}{30}\) is the crank angular velocity, and \(n\) is the plunger pump speed.

In this paper, a three-cylinder single action reciprocating mud pump is studied. It is assumed that the inhalation flow is equal to the discharge flow. Because the crank angle is 120°, the inhalation and discharge law of the pump is studied when the crank rotates to 120°. When the crank angle is 0–60°, the two cylinders are in the suction state, and a cylinder is in the discharge state; when the crank angle is 60–120°, it is exactly the opposite. Therefore, when considering the volumetric efficiency of the pump, the actual instantaneous flow can be expressed as:

\[
\begin{align*}
Q_1(\theta) &= \left\{ \begin{array}{ll}
\eta A_1 r \omega_1 \left(\sin \theta + \frac{\lambda}{2} \sin 2\theta + \sin \left(\theta + \frac{2\pi}{3}\right)\right), & \theta \in \left[0, \frac{\pi}{3}\right) \\
\eta A_1 r \omega_1 \left(\sin \theta + \frac{\lambda}{2} \sin 2\theta\right), & \theta \in \left[\frac{\pi}{3}, \frac{2\pi}{3}\right].
\end{array} \right.
\end{align*}
\]

where \(Q_1\) is the actual instantaneous flow, \(\eta\) is the volumetric efficiency, and \(A_1\) is the cylinder area.

For the plunger pump, the relationship between flow and pressure is as follows:

\[
P_1 = \left\{ \begin{array}{ll}
\frac{c}{\eta A_1 r \omega_1 \left(\sin \theta + \frac{\lambda}{2} \sin 2\theta + \sin \left(\theta + \frac{2\pi}{3}\right)\right)}, & \theta \in \left[0, \frac{\pi}{3}\right) \\
\frac{c}{\eta A_1 r \omega_1 \left(\sin \theta + \frac{\lambda}{2} \sin 2\theta\right)}, & \theta \in \left[\frac{\pi}{3}, \frac{2\pi}{3}\right].
\end{array} \right.
\]

where \(P_1\) is the discharge pressure and \(c\) is the plunger pump power.

Therefore, the plunger pump pressure can be given as:

\[
Q_1 \times P_1 = c,
\]

\[\theta \in \left[0, \frac{\pi}{3}\right) \cup \left[\frac{\pi}{3}, \frac{2\pi}{3}\right], \quad \theta \in \left[\frac{\pi}{3}, \frac{2\pi}{3}\right].
\]

2.3 Mathematical model for a pipeline

Based on the Navier–Stokes equation and Newton’s second law, a model of the hydraulic resistance, inductance, and capacitance of the drilling fluid flow in the pipeline is given. The model can be expressed as:

\[
\begin{align*}
R_d &= \frac{128\mu l}{\pi d^4}, \\
L_n &= \frac{\rho l}{A_2}, \\
C_{n1} &= \frac{V}{k},
\end{align*}
\]

where \(R_d\) is the hydraulic resistance, \(\mu\) is the dynamic viscosity, \(l\) is the pipeline length, \(d\) is the pipeline diameter, \(L_n\) is the hydraulic inductance, \(\rho\) is the density, \(A_2\) is the pipeline cross-sectional area, \(C_{n1}\) is the hydraulic capacitance considering the pipeline length, \(V\) is the pipeline volume, and \(k\) is the bulk modulus.

This model is established on the basis of ignoring pipeline compressibility. However, in the actual problem, the effect of pipeline compressibility on dynamic characteristics must be considered. Therefore, the continuous equations of compressible and nonconstant flow in the thin-walled tube are expressed as:

\[
A_2 \frac{\partial \rho}{\partial t} + \frac{\partial}{\partial x}(\rho u A_2) = 0.
\]

Based on the resilient equation of the pipeline, hydraulic capacitance considering the pipeline compression is obtained as:
where $C_{n2}$ is the hydraulic capacitance considering the pipeline compression, $E$ is the elastic modulus, and $\delta$ is the pipeline thickness.

Therefore, the hydraulic capacitance in the pipeline can be expressed as:

$$ C_n = C_{n1} + C_{n2} = \frac{V}{k} + \frac{dV}{E\delta}. \quad (8) $$

The pipeline system is compared with the mechanical spring vibrator system, with comparison parameters given in Table 1. According to the parameters shown in Table 1, the formulas can be expressed as:

$$ \begin{aligned}
M\ddot{\delta} &= F(t) - K\delta - B\dot{\delta} \\
L_n\frac{dQ_1}{dt} + C_n\dot{Q}_1 + R_1 \times Q_1 &= \Delta P = |P_1 - P_2|,
\end{aligned} \quad (9) $$

where $P_2$ is the outlet pressure of the pipeline.

### 2.4 Mathematical model for common nozzles

Based on fluid mechanics, the mathematical model of common nozzles is established. Continuity equations and energy equations can be expressed as:

$$ \begin{aligned}
v_{\|}A_{\|} &= c_4 v_1 A_1, \\
Z_1 + \frac{P_1}{\gamma} + \frac{Q_1^2}{2gA_1} &= Z_{\|} + \frac{P_{\|}}{\gamma} + \frac{Q_{\|}^2}{2gA_{\|}} + h_{m1}, \\
h_{m1} &= 0.5 \left(1 - \frac{A_{\|}}{A_1}\right)\frac{v_{\|}^2}{2g},
\end{aligned} \quad (10) $$

where $v_1$ is the flow rate in the pipeline, $A_1$ is the pipeline cross-sectional area, $c_4$ is the flow coefficient, is the flow rate in the nozzle, $A_{\|}$ is the nozzle cross-sectional area, $Z_1$ is the pipeline cross-sectional head, $P_1$ is the outlet pressure of the pipeline, $Q_1$ is the outlet flow of the pipeline, $Z_{\|}$ is the nozzle head, $P_{\|}$ is the outlet pressure of the nozzle, $Q_{\|}$ is the outlet flow of the nozzle, and $h_{m1}$ is the local pressure loss.

### 2.5 Mathematical model for Helmholtz nozzles

Figure 2 shows the transfer matrix map of the self-excited oscillating nozzle, which is predominantly comprised of the upper nozzle, the oscillating cavity, and the lower nozzle. The transfer matrix mathematical model of the Helmholtz nozzle is given by Professor Tang Chuanlin, Hunan University of Technology, which is as follows:

$$ \begin{bmatrix} H \\ Q \end{bmatrix}_{\text{out}} = F_1 P_0 F_2 \begin{bmatrix} H \\ Q \end{bmatrix}_{\text{in}}, \quad (11) $$

$F_1$, $F_2$, and $P_0$ can be expressed as:

$$ F_1 = \begin{bmatrix} \cos k_1 \omega & -jC_1 \sin k_1 \omega \\ -j \sin k_1 \omega & \cos k_1 \omega \end{bmatrix}, \quad (12) $$

$$ F_2 = \begin{bmatrix} \cos k_2 \omega & -jC_2 \sin k_2 \omega \\ -j \sin k_2 \omega & \cos k_2 \omega \end{bmatrix}, \quad (13) $$

$$ P_0 = \begin{bmatrix} 1 & 0 \\ -jV_0 \omega a^2 & 1 \end{bmatrix}, \quad (14) $$

$$ F_2 P_0 F_1 = \begin{bmatrix} a_{11} & a_{12} \\ a_{21} & a_{22} \end{bmatrix}, \quad (15) $$

where $k_i = l_i/a_i (i = 1, 2)$, $C_i = a_i/gA_{\|i} (i = 1, 2)$, $l_i$ is the length of the upper or lower nozzle, $\omega$ is the angular frequency, $a_i$ is the speed of sound, $A_{\|i}$ is the

---

**Table 1** The fluid versus mechanical spring parameters

| Springs vibration model | Damping ($B$) | Stiffness ($K$) | Mass ($M$) | Displacement ($x_2$) | Velocity ($v_2$) | Force ($F$) |
|------------------------|--------------|----------------|-----------|---------------------|----------------|------------|
| Fluid model            | Hydraulic resistance ($R_d$) | Hydraulic capacitance ($C_n$) | Hydraulic inductance ($L_n$) | Volume ($V$) | Flow ($Q_1$) | Differential pressure ($\Delta P$) |

---

**FIGURE 2** Transfer matrix diagram of self-excited oscillating nozzles
cross-sectional area of the upper or lower nozzle, \( g \) is the gravitational acceleration, \( V_0 \) is the volume of the oscillating cavity, \( a \) is the wave speed, and \( j = \sqrt{-1} \).

\[ H_{\text{out}}, Q_{\text{out}} \text{ can be expressed as:} \]
\[
\begin{align*}
H_{\text{out}} &= a_{11} \times H_{\text{in}} + a_{12} \times Q_{\text{in}}, \\
Q_{\text{out}} &= a_{21} \times H_{\text{in}} + a_{22} \times Q_{\text{in}}.
\end{align*}
\]  

(16)

When the water head is ignored, the following can be obtained:

\[
\begin{align*}
P_{\text{out}} &= a_{11} \times P_{\text{in}} + \gamma \times a_{12} \times Q_{\text{in}}, \\
Q_{\text{out}} &= a_{21} \times P_{\text{in}} \gamma + a_{22} \times Q_{\text{in}},
\end{align*}
\]  

(17)

where \( P_{\text{out}} \) and \( P_{\text{in}} \) are the outlet and inlet pressure of the Helmholtz nozzles, respectively, \( \gamma \) is the volumetric weight; and \( Q_{\text{out}} \) and \( Q_{\text{in}} \) are the outlet and inlet flow of the Helmholtz nozzles, respectively.

3 | NUMERICAL SIMULATION AND ANALYSIS

3.1 | Computational process and parameters

As shown in Figure 3, the basic idea of the proposed model is to first calculate the output characteristics of the plunger pump according to the structural characteristics of the plunger pump and subsequently calculate the pressure drop of the pipeline and nozzle separately before finally superimposing each part to obtain the output characteristics of the nozzle.

In the mathematical model of the pump, the pump parameters are set as work and power \( c = 11 \text{ kW} \), crank radius \( r = 0.01 \text{ m} \), link ratio \( \lambda = 1/8 \), volumetric efficiency \( \eta = 95\% \), and cylinder area \( A_1 = 1.76 \times 10^{-4} \text{ m}^2 \).

In the mathematical model of the pipeline, the parameters of the rubber pipeline with a steel ring are set as the dynamic viscosity \( \mu = 1.01 \times 10^{-3} \text{ Pa s} \), the pipeline diameter \( d_2 = 0.016 \text{ m} \), the pipeline cross-sectional area \( A_2 = 2 \times 10^{-4} \text{ m}^2 \), the bulk modulus \( k = 2.06 \times 10^9 \text{ Pa} \), the elastic modulus \( E = 2 \times 10^{11} \text{ Pa} \), and the pipeline thickness \( \delta = 1.7 \times 10^{-3} \text{ m} \).

Then, the mathematical model of the nozzle is composed of common Helmholtz nozzles. The parameters of the common nozzle are set as the flow coefficient \( c_d = 0.82 \), and the nozzle cross-sectional area \( A_{11} = 7.85 \times 10^{-7} \text{ m}^2 \). Meanwhile, the parameters of the Helmholtz nozzle are set as the length of the upper nozzle \( l_1 = 0.03 \text{ m} \), the length of the lower nozzle \( l_2 = 0.015 \text{ m} \), the angular frequency \( \omega = 5.75 \text{ Hz} \), the speed of sound \( a_1 = a_2 = 1200 \text{ m/s} \), the cross-sectional area of the upper and lower nozzles \( A_{11} = A_{12} = 7.85 \times 10^{-7} \text{ m}^2 \), the gravitational acceleration \( g = 9.8 \text{ m/s}^2 \), the volume of oscillating cavity \( V_0 = 2.16\pi \times 10^{-7} \text{ m}^3 \), the wave speed \( a = 500 \text{ m/s} \), and the volumetric weight \( \gamma = 1000 \text{ kg/m}^3 \).
3.2 | Results and discussion

3.2.1 | Sensitivity analysis

The inlet and outlet characteristics of the different types of nozzles are shown in Figure 4. It is clearly shown in the figure that the nozzle outlet flow is less than the inlet flow under certain conditions and that this effect largely depends on the nozzle flow coefficient. Moreover, the effect on outlet flow is the same for different types of nozzles, probably because the transfer matrix method was used to model the Helmholtz nozzles by considering the oscillating chamber as a point matrix, thus neglecting the effect of the oscillating chamber on flow. In contrast, a common nozzle has the capability of enhancing the pressure amplitude (Figure 4A), while Helmholtz nozzles have a large influence on the pressure value, such that the outlet peak pressure is generally greater than the inlet pressure (Figure 4B). In other words, it is found that the amplitude of the outlet pressure is increased by common nozzles, while the peak pressure is increased by the Helmholtz nozzle. These conclusions are consistent with the results of the laboratory test performed by Li et al.26

Figure 5 illustrates the effects of rotational speed on the dynamic characteristics of the nozzle. The flow changes in the inlet and outlet are almost parallel straight lines. That is, the speed of the plunger pump has a small effect on the trend of the peak and amplitude of the flow. In addition, the peak and amplitude of the inlet

FIGURE 4 Dynamic characteristics of the inlet and outlet of the nozzle. (A) Common nozzle. (B) Helmholtz nozzle

FIGURE 5 Dynamic characteristics under different rotational speeds. (A) Common nozzle. (B) Helmholtz nozzle
and outlet pressures increase as the speed increases, with the peak pressure increasing approximately as a quadratic curve and the amplitude of the pressure increasing approximately as a straight line. In contrast, the outlet pressure amplitude of the common nozzle increases slowly with the speed (Figure 5A), but the peak outlet pressure of the Helmholtz nozzle increases rapidly with the speed (Figure 5B).

To study the influence of the pipeline length on the output characteristics of the nozzle, the output dynamic characteristics of the nozzle at 25, 50, 75, 100, 125, and 150 m are simulated. It is clear that the peak pressure and amplitude decrease as the length of the pipe increases (Figure 6), probably because the length of the pipe increases the pressure loss. Moreover, the common nozzle outlet pressure amplitude is greater than that of the inlet (Figure 6A), while the Helmholtz nozzle outlet pressure peak and amplitude are greater than that of the inlet (Figure 6B). In other words, the performance of Helmholtz nozzles is better than the performance of common nozzles.

### 3.2.2 Vibration characteristics analysis

Based on the fluid resonance generated by a self-resonating water jet, its characteristic is largely determined by resonance frequency; thus, it is significant to study the frequency characteristics of the nozzle. When the plunger pump speed is 100 rpm, the simulation is carried out, and the spectrum obtained by the fast Fourier transform of the jet pressure signal is shown in Figure 7. As seen in Figure 7, the frequency band is mainly concentrated at 5, 10, 20, and 30 Hz, which is the integer times the interference frequency (5 Hz). The secondary frequency peak (30.62 MPa) of the common nozzle is 1.08 times greater than the interference frequency peak (28.30 MPa), and the secondary frequency peak (23.77 MPa) of the Helmholtz nozzle is 1.26 times greater than the interference frequency peak (18.84 MPa). That is, Helmholtz nozzles typically generate stronger natural vibrations than common nozzles.

### 4 EXPERIMENTAL VALIDATION

As shown in Figure 8, a laboratory experiment is conducted to study the proposed model. The experimental setup is improved from the erosion tester independently developed by our research team. The setup can not only provide relevant output dynamic characteristic measurements for water jets but can also be used for surface-strengthened metal abrasion and abrasive jet erosion experiments. The test system consists of an erosion tester and a data acquisition and processing system. The erosion tester is mainly composed of a water tank, plunger pump, pipeline, fitting, nozzle, and target plate; the data acquisition and processing system are composed of a pressure transducer, data logger, and laptop.

In the experiment, the water was provided by a plunger pump whose working flow can be continuously regulated from 0 to 15 L/min by changing the frequency of the motor powering for the pump. The maximum pressure of the plunger pump is 35 MPa. Moreover, because the nozzle is small, it is difficult to install the pressure transducer on it. Therefore, as shown in Figure 8, a dynamic pressure transducer (Model: CYB-20S/SA) that
had been calibrated by the manufacturer in advance was installed on a target plate that had at its center a pressure hole with a diameter of 1 mm communicating to the transducer. The parameters of the pressure transducer are shown in Table 2. Meanwhile, the data are transmitted to the data logger (Model: DH5909) produced by the Donghua Testing Company to realize the real-time monitoring of the data. Finally, the data are imported into a computer for data processing.

The model is verified by the test system, while the impact force requires further calculation. It is noteworthy that Enrico Fuchs derived the formula for the impact force (eq. 27). Considering that the pressure transducer is used, Formula (18) is deduced, and Formula (19) of the theoretical impact force calculation is further obtained.

\[ F = \frac{\pi}{4} d_p^2 \rho u_{jet}^2, \]  
\[ F_i = \frac{d_p^2}{d_n^2} \rho u_{jet}^2, \]  

**Table 2** Primary parameters of the pressure transducer

| Model       | Parameter | Value  |
|-------------|-----------|--------|
| CYB-20S/SA  | Range     | 0–30 MPa |
|             | Accuracy  | 0.2% FS |
|             | Power supply | 24VDC   |
|             | Output signal | 0–10 V   |

**Figure 7** Spectrum maps. (A) Common nozzle. (B) Helmholtz nozzle

**Figure 8** Schematic diagram of the experimental setup
where $F$ is the impact force, $d_N$ is the nozzle diameter, $u_{\text{jet}}$ is the velocity, $F_I$ is the theoretical impact force, and $d_h$ is the diameter of the pressure hole.

The above experimental conditions are put into the proposed model, and the simulation result is shown in Figure 9. From Figure 9, the changing trend of the outlet pressure in the simulation result is consistent with the experimental result. However, the maximum outlet pressure in the experiment is larger than that of the simulation. This is because the actual conditions are more complex than the assumptions in the paper. From the experiment, the feasibility of the theoretical models can be verified. The simulation and experimental results collectively show that the proposed model in this paper can meet the engineering requirements.

**5 | CONCLUSION**

Based on fluid mechanics, an integrated model of the water jet system is established. Then, the sensitivity of different nozzles on the pulsating fluid is analyzed in the time domain, and the effects of speed and pipeline length on the outlet characteristics of the nozzle are studied. Furthermore, the vibration characteristics of different nozzles in the frequency domain are investigated. The experimental results show that the simulated impact force is in good agreement with the actual values, and the accuracies of the simulation results are verified.

According to the time domain analysis, the common nozzle enlarges the pressure amplitude, and the Helmholtz nozzle enlarges the peak pressure and amplitude. In addition, the pump speed has a greater influence on the amplification effect. This may be because changes in speed are more likely to cause unstable flow in the plunger pump, resulting in pressure pulsation. Thereafter, the influence of the pipeline length on the outlet characteristics of the nozzle showed that the pressure characteristics of the nozzle at the inlet and outlet gradually decreased, which may be due to the increase in fluid energy loss. However, the difference between the pressure of the inlet and outlet has not changed. Put simply, the nozzle is more sensitive to the change in plunger pump speed. And a water jet with a larger pulsation amplitude by increasing the rotational speed of the plunger pump is easily obtained.

Further studies on the frequency domain demonstrated that the secondary frequency peak for the Helmholtz nozzle is 1.26 times larger than the interference frequency, while it is 1.08 times for the common nozzle. That is, Helmholtz nozzles provide stronger amplification of inlet flow. Therefore, under the same construction conditions, the Helmholtz nozzle is a better choice to provide a more efficient water jet.

**NOMENCLATURE**

- $A_1$: cylinder area, m$^2$
- $A_2, A_3$: pipeline cross-sectional area, m$^2$
- $A_{\text{HI}}$: nozzle cross-sectional area, m$^2$
- $A_{\text{HI}}$: cross-sectional area of the upper or lower nozzle, m$^2$
- $a$: wave speed, m/s
- $a_{\text{i}}$: speed of sound, m/s
- $C_{\text{n1}}$: hydraulic capacitance considering the pipeline length
- $C_{\text{n2}}$: hydraulic capacitance considering the pipeline compression
- $c$: plunger pump power, W
- $c_d$: flow coefficient
- $d_2$: pipeline diameter, m
- $d_h$: diameter of the pressure hole, m
- $d_N$: nozzle diameter, m
- $E$: elastic modulus, Pa
- $F$: impact force, N
- $F_I$: theoretical impact force, Pa
- $g$: gravitational acceleration, m/s$^2$
- $h_{\text{m1}}$: local pressure loss, m
- $k$: bulk modulus, Pa
- $L_n$: hydraulic inductance
- $l$: pipeline length, m
- $l_{\text{i}}$: length of the upper or lower nozzle, m
- $n$: plunger pump speed, rpm
- $P_1$: discharge pressure, Pa
- $P_2, P_{\text{II}}$: outlet pressure of the pipeline, Pa
- $P_{\text{I}}, P_{\text{II}}$: outlet pressure of common nozzle, Pa
- $d_N$: nozzle diameter, m
\( P_{\text{out}} \) outlet pressure of Helmholtz nozzle, Pa
\( P_{\text{in}} \) inlet pressure of Helmholtz nozzle, Pa
\( Q_1 \) actual instantaneous flow, m³/s
\( Q_{\text{II}} \) outlet flow of common nozzle, m³/s
\( Q_{\text{III}} \) inlet flow of Helmholtz nozzle, m³/s
\( Q_{\text{out}} \) outlet flow of Helmholtz nozzle, m³/s
\( R_d \) hydraulic resistance
\( r \) crank radius, m
\( v_0 \) volume of the oscillating cavity, m³
\( V \) pipeline volume, m³
\( v_t \) flow rate in the pipeline, m³/s
\( v_{\text{in}} \) flow rate in the nozzle, m³/s
\( x_0 \) piston displacement, m
\( Z_{\text{I}} \) pipeline cross-sectional water head, m
\( Z_{\text{II}} \) nozzle cross-sectional water head, m
\( \gamma \) volumetric weight, kg/m³
\( \delta \) pipeline thickness, m
\( \eta \) volumetric efficiency
\( \theta \) crank angle, rad
\( \lambda \) link ratio
\( \mu \) dynamic viscosity, Padas
\( \rho \) density, kg/m³
\( \omega \) angular frequency, Hz
\( \omega_1 \) crank angular velocity, rad/s

AUTHOR CONTRIBUTIONS
Chao Feng: Methodology; writing of the original draft; data processing. Yu Wang: Project administration. Lingrong Kong: Supervision.

ACKNOWLEDGMENTS
The work is supported by the National Key Research and Development Program of China (2018YFC1802404). The authors sincerely acknowledge the precious researchers for their excellent work, which greatly assisted their academic study.

CONFLICT OF INTEREST
The authors declare no conflict of interest.

ORCID
Chao Feng  
http://orcid.org/0000-0002-4372-8507

REFERENCES
1. Li G, Shi H, Liao H, et al. Hydraulic pulsed cavitating jet-assisted drilling. Pet Sci Technol. 2009;27(2):197-207.
2. Fu JS, Li GS, Shi HZ, et al. Analysis on adaptability of hydraulic pulse cavitating jet drilling technology. Oil Drill Prod Technol. 2012;34(5):10-14.
3. Cheng RC, Ge YH, Wang HG, et al. Self-oscillation pulsed percussive rotary tool enhances drilling through hard igneous formations. Paper presented at: The IADC/SPE Asia Pacific Drilling Technology Conference and Exhibition; 2012.
4. Guha A, Barron RM, Balachandar R. An experimental and numerical study of water jet cleaning process. J Mater Process Technol. 2011;211(4):610-618.
5. Li D, Kang Y, Wang X, Ding X, Fang Z. Effects of nozzle inner surface roughness on the cavitational erosion characteristics of high speed submerged jets. Exp Therm Fluid Sci. 2016;74:444-452.
6. Chahine GL, Kapahi A, Choi JK, Hsiao CT. Modelling of surface cleaning by cavitation bubble dynamics and collapse. Ultrason Sonochem. 2016;29:528-549.
7. Reinsch T, Paap B, Hahn S, Wittig V, van den Berg S. Insights into the radial water jet drilling technology-application in a quarry. J Rock Mech Geotech Eng. 2018;10(2):236-248.
8. He L, Liu Y, Wu Y, Sun H, Shen K, Yang X. The effects of process parameters on the rock-breaking efficiency of multi-nozzle jet. J Pet Sci Eng. 2021;206:108857.
9. Yang R, Hong C, Huang Z, Song X, Zhang S, Wen H. Coal breakage using abrasive liquid nitrogen jet and its implications for coalbed methane recovery. Appl Energy. 2019;253:113485.
10. Xu YP, Wang LG. Technical parameters of hydraulic punching in a typical coal seam and an investigation of outburst prevention effect: a case study in the Machi mine, China. Geotech Geol Eng. 2020;38(2):1971-1986.
11. Dickinson W, Dickinson R. Horizontal radial drilling system. Paper presented at: The SPE California Regional Meeting; 1985.
12. Salimzadeh S, Grandahl M, Medetbekova M, Nick HM. A novel radial jet drilling stimulation technique for enhancing heat recovery from fractured geothermal reservoirs. Renew Energy. 2019;139:395-409.
13. Tang Y, Sun P, Wang GR, et al. Rock-breaking mechanism and efficiency of straight-swirling mixed nozzle for the nondiagenetic natural gas hydrate in deep-sea shallow. Energy Sci Eng. 2020;8(10):3740-3752.
14. Wang P, Zhao B, Ni HJ, et al. Research on the modulation mechanism and rock breaking efficiency of a cuttings waterjet. Energy Sci Eng. 2019;7(5):1687-1704.
15. Pei J, He C, Lv M, Huang X, Shen K, Bi K. The valve motion characteristics of a reciprocating pump. Mech Syst Signal Process. 2016;66-67:657-664.
16. Wang G, Zhong L, He X, et al. Dynamic behavior of reciprocating plunger pump discharge valve based on fluid structure interaction and experimental analysis. PLoS One. 2015;10(10):e0140396.
17. Zhang WW, Yu ZY, Li YJ, et al. Numerical analysis of pressure fluctuation in a multiphase rotodynamic pump with air-water two-phase flow. Oil Gas Sci Technol Rev IFP Energies Nouvelles. 2019;74:18.
18. Ma Y, Ni Y, Zhang H, Zhou S, Deng H. Influence of valve’s lag characteristic on pressure pulsation and performance of reciprocating multiphase pump. J Pet Sci Eng. 2018;164:584-594.
19. Lin J, Wang Y, Zhou S, Wu W, Ma H, Han Q. Simulation and experimental analysis of pressure pulsation characteristics of pump source fluid. Appl Sci. 2021;11:9559.
20. Khudayarov BA, Komilova KHM, Turaev FZH. Numerical simulation of vibration of composite pipelines conveying pulsating fluid. *Int J Appl Mech*. 2019;11(9):1950090.

21. Khudayarov BA, Komilova KHM. Dynamic analysis of the suspended composite pipelines conveying pulsating fluid. *J Nat Gas Sci Eng*. 2020;75:103148.

22. Gorman DG, Reese J, Zhang YL. Vibration of a flexible pipe conveying viscous pulsating fluid flow. *J Sound Vib*. 2000;230(2):379-392.

23. Li XL, Yan YY. Influence of high pressure pulsating fluid on seal of pipeline fittings based on multi-scale model. *Int J Press Vessels Pip*. 2021;190:104300.

24. Pałczyński T. Influence of air temperature on dynamic properties of pipes supplied with pulsating flow. In: Awrejcewicz J, ed. *Dynamical Systems in Applications, DSTA 2017*. Springer Proceedings in Mathematics & Statistics. Vol 249. Springer; 2018:291-302.

25. Tuna B, Rockwell D. Self-sustained oscillations of shallow flow past sequential cavities. *J Fluid Mech*. 2014;758:655-685.

26. Li D, Kang Y, Ding X, Wang X, Fang Z. Effects of area discontinuity at nozzle inlet on the characteristics of high speed self-excited oscillation pulsed waterjets. *Exp Therm Fluid Sci*. 2016;79:254-265.

27. Crow SC, Champagne F. Orderly structure in jet turbulence. *J Fluid Mech*. 1971;48(3):547-591.

28. Morel T. Experimental study of a jet-driven Helmholtz oscillator. *J Fluids Eng*. 1979;101:383-390.

29. Chahine GL, Conn AF, Johnson JVE. Cleaning and cutting with self-resonating pulsed water jets. Paper presented at: The 2nd US Water Jet Conference, 1983.

30. Liao Z, Tang C. Theoretical analysis and experimental study of the self-excited oscillation pulsed jet device. Paper presented at: The Proceedings of the Fourth US Water Jet Conference, 1987:26-28.

31. Kolšek T, Jelić N, Duhovnik J. Numerical study of flow asymmetry and self-sustained jet oscillations in geometrically symmetric cavities. *Appl Math Model*. 2007;1(10):2355-2373.

32. Pei JH. Study on the relation between oscillation frequency of self-excited oscillation pulsed water jet and self-excited pulsed water jet devices. Paper presented at: The Advanced Materials Research, 2011:547-552.

33. Zhang X, Li X, Nie S, Wang L, Dong J. Study on velocity and pressure characteristics of self-excited oscillating nozzle. *J Braz Soc Mech Sci Eng*. 2021;43:5.

34. Liu Q, Wu J, Liu W, Wang R. A frequency-domain propagation model of bypass downlink system with transfer matrix method. *J Pet Sci Eng*. 2017;159:724-730.

35. Atangana Njock PG, Shen JS, Modoni G, Arulrajah A. Recent advances in horizontal jet grouting (HJG): an overview. *Arabian J Sci Eng*. 2018;43(4):1543-1560.

36. Luo XC. Analysis of attenuation characteristics of hybrid fluid. Jiangxi University of Science and Technology; 2019.

37. Lin JZ, Ruan XD, Guo C, et al. *Fluid Mechanics*. Tsinghua University Press; 2005.

38. Wu X. The dynamic study on the series pulsed jet nozzle. Hunan University of Technology, 2007.

39. Fuchs E, Koehler H, Majschak JP. Measurement of the impact force and pressure of water jets under the influence of jet break-up. *Chem Ing Tech*. 2019;91(4):455-466.

How to cite this article: Feng C, Wang Y, Kong L. Effects of pulsating fluid at nozzle inlet on the output characteristics of common and self-excited oscillation nozzle. *Energy Sci Eng*. 2022;10:3189-3200. doi:10.1002/ese3.1213