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Paper

Pulse Response and Structure Optimization of a Novel Disc-Type Pulse Jet Generating Tool

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ABSTRACT

During drilling extended reach wells and horizontal wells, friction force can be reduced effectively by pulse tools like hydro-oscillator. Rotary percussion and torsional percussion can effectively improve the rock breaking efficiency. In order to provide a controllable pulse source, a disc-type pulsed water jet generator, driven by hydrodynamic impeller is proposed, whose flow area is changed by the rotation of discs. Therefore, the continuous jet is modulated into a pulse jet. A series of numerical simulations are performed to analyze the pulse characteristics of the generator. Specifically, the response characteristics of rotation speed at various flow rates, inner diameter of the impeller, blade helix angle, number of blades are calculated. In these cases, as the inner diameter of the impeller increases, the impeller rotating speed, which decreases first and then increases, varies from 500 rpm to 1200rpm. As the helix angle and flow rate increase, the impeller rotating speed increases significantly. When increasing flow rate of water from 20 L/s to 35 L/s, the pressure fluctuation amplitude increases to 0.7 MPa. The research results show that it can be realized to control the pulse characteristics by adjusting flow rate and structure parameters. This work originally provides the basis for parameters optimization and field application of this generator.

1. INTRODUCTION

Statistics show that the drilling time occupies nearly 1/3 of the well cycle (Anderson, E. E, et al. 1990), and increasing the rate of penetration（ROP）can significantly shorten the drilling cycle and reduce drilling costs eventually. Pulsed jet drilling technology can effectively utilize the bottom hole hydraulic energy, optimize the distribution of jet energy, improve the efficiency of rock-breaking and cuttings-cleaning, which will increase the rate of penetration. Another method, namely pulse vibration drilling technology, can introduce a vibration impact on the rock while rotating by using a pulse jet and a rotary shock, thus leading to a high ROP. For both the pulse jet drilling technology and the pulse vibration drilling technology, the pulse generator is required to provide a stable pulse power source to improve the rate of penetration ultimately.

To improve the ROP by pulsed power, Jack Kolle et al. (Kolle, Jack, and Mark Marvin.1999) developed a "Hydropusle" negative pressure pulse tool, which created a transient partial underbalance and a negative pressure pulse in the bottom hole by controlling the flow connectivity between the drill bit and the tool. Based on the transient flow principle (Johnson Jr, V. E., et al.1981), the LPMS-I pulse modulator used the mechanical oscillation blocking method to make the main valve of the tool periodically shut down the flow path of the drilling fluid, forming a negative pressure underbalance state near the bit, improving the efficiency of rock breaking. Another pulse cavitation jet generator designed by China University of Petroleum (Beijing) could generate a pulse by means of mechanical blocking, and enhance the pulse through the self-excited oscillation cavity. The pulse cavitation can be formed with the assistance of pulse cavitation nozzles(Li, Gensheng, et al. 2012; Fu, Jiasheng, et al. 2012; Shi, Huaizhong, et al. 2010; Li, Gensheng, and Shi, Huaizhong. 2008; Li, Gensheng, et al. 2009; Li, Gensheng, et al. 2005). In such a kind of pulse cavitation generator, the drilling fluid changed the flow direction when passing through a fluid guiding body. Then it impacted and pushed the impeller to rotate, changed the flow area, and generated speed fluctuation and pressure fluctuation of the drilling fluid, forming a pulse. Furthermore, the pulse could be amplified by the self-excited oscillatory cavity, and acted on the bottom of the well. Cui Longlian(Cui, Longlian, et al. 2012) et al. developed a frequency-adjustable pulse speed-up tool. When the drilling fluid impacted turbine blades, it lead the center drive shaft of the turbine to rotating and changing the flow area, which resulted in a periodic sealing by the periodical pulsed drilling fluid through the tool.

In this paper, a new kind of disc-type pulsed water jet generating device was proposed based on previous studies. A series of numerical simulation studies were carried out, including effects of impeller inner diameter, number of blades, blade helix angle and displacement on the dynamic characteristics of the impeller, as well as the effects of the speed of the dynamic disc valve, the diameter and number of the open hole, and the displacement on the pulse characteristics.

2. DISC-TYPE PULSED WATER JET GENERATOR

In this paper, a new kind of disc-type pulsed water jet generating device is described as shown in Figure.1, which consists of a power system and a pulse generating system. The power system includes an axial-rotating impeller, while the pulse generating system is composed of a dynamic disc valve and a static disc valve. The dynamic disc valve is connected to the impeller through the impeller shaft while the static disc valve is fixed together with the wall of the shell. Both the dynamic disc valve and the static disc valve are distributed with the same number of circular holes. When the drilling fluid passes through the impeller, it drives the impeller shaft to rotate, transforming the kinetic energy of the drilling fluid into the mechanical energy of the impeller shaft, and the rotating impeller shaft drives the dynamic disc valve to rotate. With the rotation of the dynamic disc valve, the circular holes on these two disc valves alternately overlap or disconnect, modulating the continuous drilling fluid flow into a pulse jet (Figure. 2). The advantage of this tool is that the amplitude and frequency of the hydraulic pulsed jet are easily controlled by adjusting structural parameters and drilling fluid parameters.

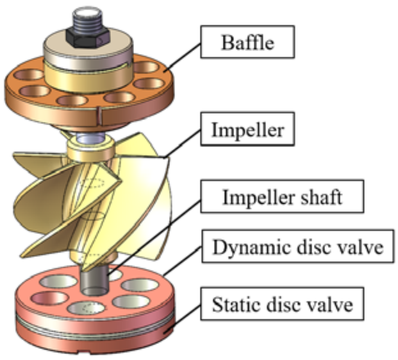


Figure.1 Novel Disc-type Pulsed Water Jet Generator

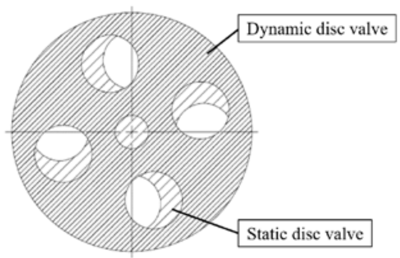


Figure.2 Mechanism Diagram of Pulsed Jet Flow

3. THEORETICAL ANALYSIS OF POWER SYSTEM

3.1 Basic Parameters

The impeller with right-pitch helical blades is used in this paper, whose outer diameter D_1 is 96mm and the blade thickness H is 4mm. The diameter D_0 of the sleeve through which the drilling fluid flows is 100mm, the radius R of the contact surface of impeller shaft and baffle is 5mm, and the coefficient of friction k is 0.1. In the working process, the blade generates torque, rotates under the impact of the fluid, transmits torque to the dynamic disc valve through the shaft, and drives the flow area to change. In this paper, pure water was used as the fluid medium, and the influence of structural parameters and drilling fluid parameters on the power output characteristics of the impeller motor was analyzed through theoretical calculation.

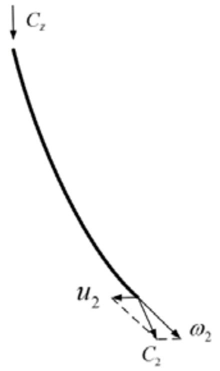


Figure 3. Diagram of Impeller Inlet and Outlet Velocity

3.2 Theoretical Calculation

According to vectors of impeller inlet and outlet velocity in Figure.3, The torque produced by liquid impinging on blades is given by：

$$M = \rho Q \frac{\bar{D}}{2}\left(C_z \sin\beta - u_2\right) \tag{1}$$

Where: ρ is the fluid density, kg • m^{-3}; Q is the flow rate through the impeller, L • s^{-1}; $\bar{D}$ is middle diameter of the impeller, m; C_z is actual axial velocity, m • s^{-1}; β is helix angle, rad; u_2 is circumferential velocity.

Outlet circumferential velocity is given by:

$$u_2 = \frac{\pi n \bar{D}}{60} \tag{2}$$

Considering that there is a 2mm gap between the blade and the sleeve wall, the actual flow through the blade should be relatively small, thus:

$$Q = aQ' \tag{3}$$

According to the relationship of the overcurrent area, a=0. 9293.

Axial velocity is given by：

$$C_z' = \frac{Q}{\pi \bar{D} b} \tag{4}$$

Since the blade has a certain thickness and number, the influence of the excretion coefficient on the axial velocity should also be considered, so the actual axial velocity is:

$$C_z = \frac{C_z'}{\varphi} \tag{5}$$

Excretion coefficient is given by:

$$\varphi = 1 - \frac{zc}{\pi \bar{D}} \tag{6}$$

Where: z is the thickness of blade, m; c is the number of blades, b is the radial length of the blade.

The weight of entire impeller and the drilling fluid impact force act on the action surface of the impeller shaft and the baffle, thus torque generated by friction is:

$$M_b = \int_0^R \frac{\mu\left[mg + \rho Q(C_z - C_2)\right]}{R^2} 2r^2 \mathrm{d}r = \frac{2\mu R\left[mg + \rho Q(C_z - C_2)\right]}{3} \tag{7}$$

Where: m is the weight of dynamic disc valve and blades, kg; C_2 is axial velocity of impeller outlet, m • s^{-1}, $C_2 = C_z \cdot cos\beta$; μ is the friction coefficient between the impeller shaft and the baffle; R is the radius of contact surface between impeller shaft and baffle, m.

When the impeller shaft rotates at a constant speed, it can be considered to be in an equilibrium state，unite the formula (1) and formula (7):

$$n = \frac{60C_z \sin\beta}{\pi \bar{D}} - \frac{120M_b}{\pi \rho Q \bar{D}^2} \tag{8}$$

According to the above equation, the output parameters corresponding to different drilling fluid parameters and different structure parameters can be calculated, shown as Figure.4.

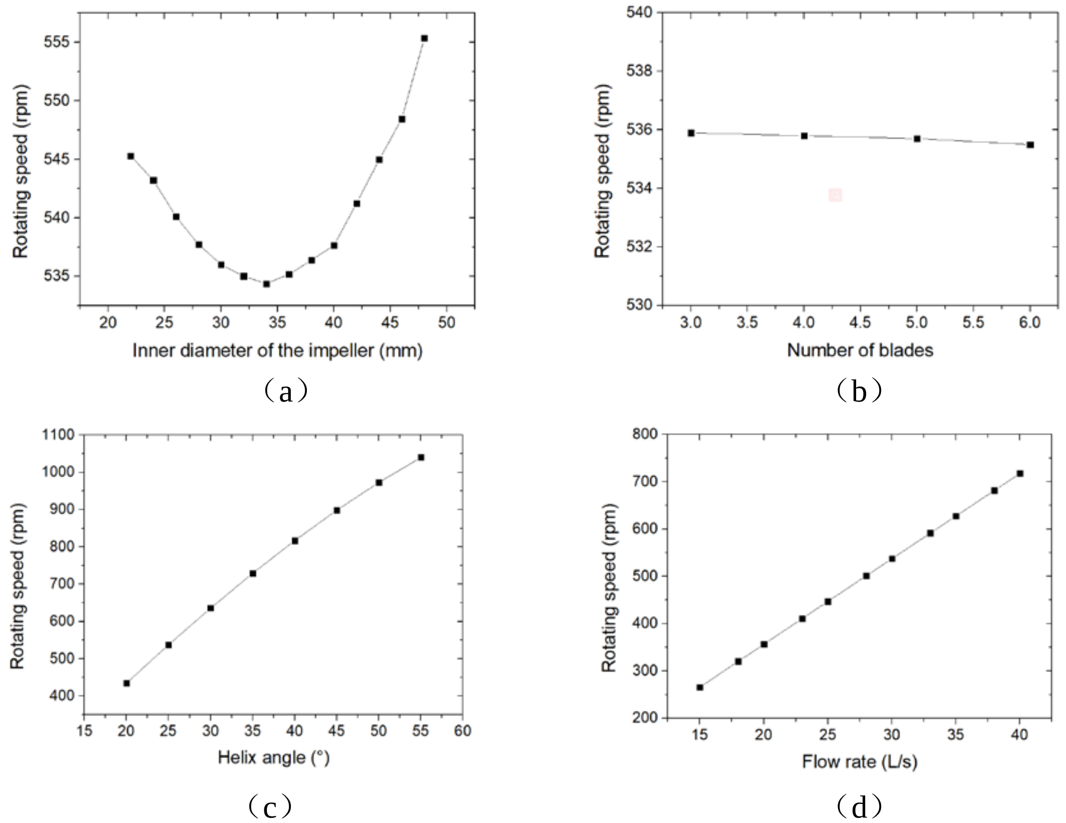


Figure.4 The Rotating Speed of the Impeller Varies with (a) Inner Diameter, (b) the Number of Blades, (c) the Helix Angle and (d) the Flow Rate of Drilling Fluid

According to the curves shown in Figure 4, as the inner diameter increases, the impeller rotation speed decreases first and then increases. On the other hand, the impeller rotation speed seems to be unaffected by the number of blades. Moreover, the rotation speed significantly increases with increasing helix angle and increasing flow rate.

4. NUMERICAL SIMULATION

4.1 Physical Model

For the impeller power system of the pulsed water jet generator, four kinds of inner diameter size, four kinds of blades, four kinds of helix angle impellers and six kinds of displacements were selected to complete the analysis of their effects on the torque, pressure loss and rotation speed, as shown in Table 1.

Table 1. Different numerical simulation parameters of impeller

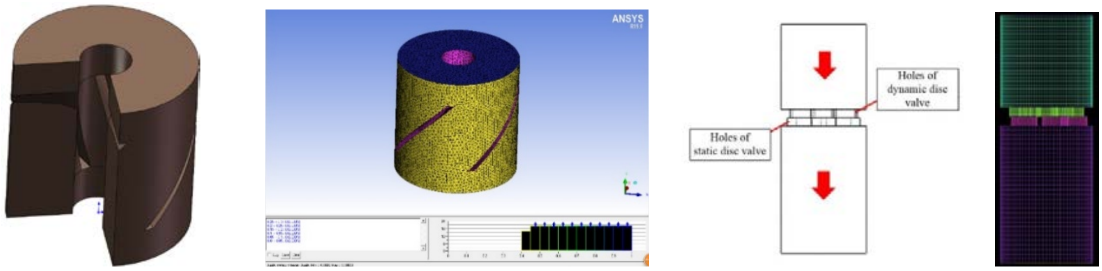
No	**Inner diameter(mm)**	**Number of blades**	**Helix angle(°)**	**Displacement ($L\cdot s^{-1}$)**
1	28	3	25	15
2	34	4	35	20
3	40	5	45	25
4	46	6	55	30
5				35
6				40

For the pulse generation system, six kinds of rotation speeds, four kinds of displacements, five kinds of disc valve holes and four kinds of disc valve apertures are selected to complete the simulation analysis of the pulse characteristics, as shown in Table 2.

Table 2. Different numerical simulation parameters of disc valve

No	**rotation speed （rpm）**	**displacement （L/s）**	**Number of holes**	**Diameter of holes （mm）**	**Drilling fluid**
1	100	20	2	16	pure water
2	150	25	3	20	pure water
3	200	30	4	24	pure water
4	300	35	5	28	pure water
5	400		6		
6	500				

According to the above selected different impeller power system and pulse generating system, a three-dimensional model (shown in Figure.5) was built using SolidWorks, and imported into ICEM for meshing. Due to the complex structure of the impeller, the unstructured mesh was used for mesh division, and the structure of the pulse generating system was relatively regular, the mesh with better quality was used for mesh division. The number of meshes was 220,000 and 180,000 respectively, the mesh quality was greater than 0.6, and the orthogonal mesh quality was greater than 0.7, which was consistent with the mesh quality required for numerical simulation.



(a) (b) (c) (d)

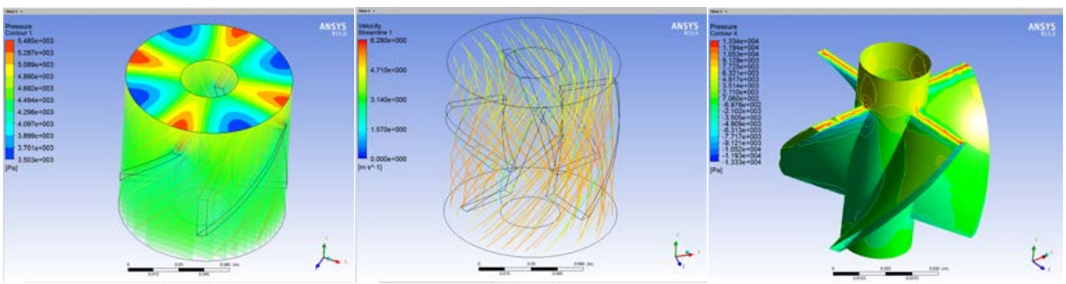
Figure.5 Geometric Model and Structure Division of Impeller and Pulse Generating Mechanism

The Realizable k-ε model is selected, which was suitable for the description of rotating shear flow. The SIMPLE algorithm is a semi-implicit separation method and it belongs to the pressure correction method based on the finite volume method. It is the most widely-used flow field calculation method in engineering.

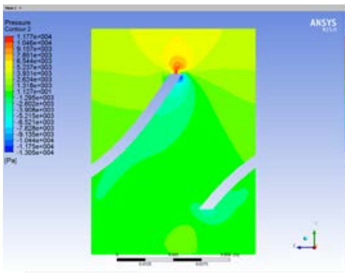
The inlet boundary is set as the velocity inlet while the outlet port is directly set as the atmosphere pressure. There is no slippery near the wall surface, and the wall function method is selected for the fluid phase. The clean water is used as the fluid phase, and the initial velocity of the fluid at the inlet boundary is given. Set the corresponding turbulent dissipation rate and turbulent kinetic energy. For the numerical simulation of the dynamic mechanism, the 6DOF (Six Degrees of Freedom) model in the dynamic mesh model is used to make the impeller passively rotate until the rotational speed is stable. For the numerical simulation of the pulse generating mechanism, the holes of the disc valves are given a certain rotational speed for active rotation.

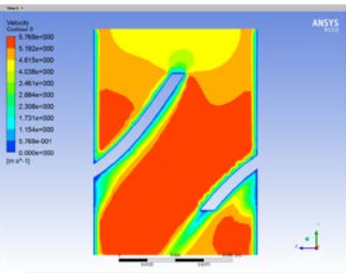
4.2 Simulation Results and Discussion

4.2.1 Numerical Simulation Analysis of Impeller Power Mechanism



(a) (b) (c)

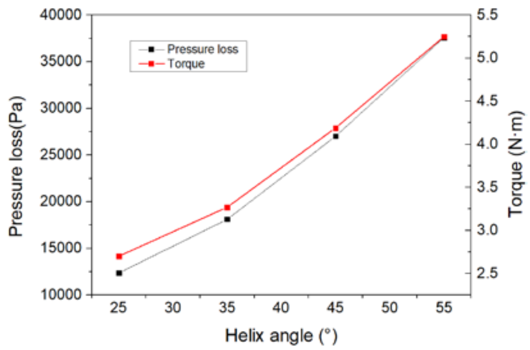




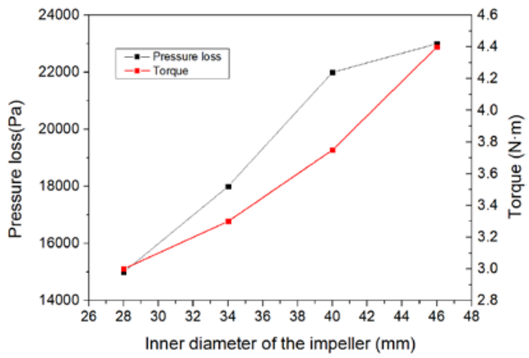
(d) (e)

Figure.6 Pressure and Velocity Distributions Near Impeller(a) Three-dimensional Pressure Distribution, (b) Flow Diagram, (c) Surface Pressure Distribution of Impeller, (d) Side View of Pressure Distribution, and (e) Side View of Velocity Distribution

It is found that the pressure is distributed like a petal at the entrance of the impeller through numerical simulation, as shown in Figure.6 (a), which should be closely related to the presence of the blade and increased pressure on the upper end surface of the blade. The flow rate of the drilling fluid also becomes larger as it enters the fluid channel between blades, and gradually remains substantially unchanged. At the same time, it can be seen from the figure that the flow rate of the drilling fluid becomes smaller or even zero near the wall surface of the blade, in accordance with the theoretical analysis of the influence of the wall on the flow velocity of the drilling fluid. The pressure exerted by the drilling fluid on the upper side of the blade is relatively large, so the resulting torque forces the impeller to rotate, converting the kinetic energy of the drilling fluid into the mechanical energy of the impeller.



(a）



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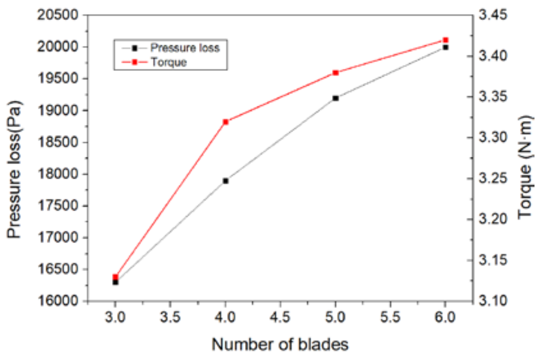

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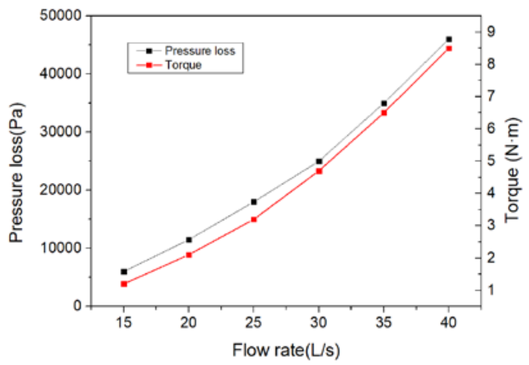


(d)

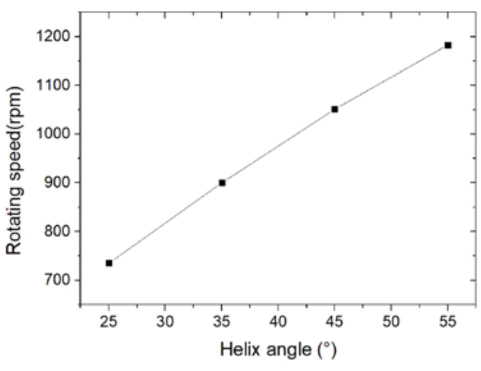
Figure.7 Changes of Pressure Loss and Torque with (a) Helix Angle of the Blade,(b)Inner Diameter of Impeller,(c) Number of Blades,(d) Flow Rate of Drilling Fluid

Figure 7 is a summary of the changes of pressure loss and torque as a function of blade helix angle, impeller inner diameter, number of blades, and drilling fluid displacement. It can be seen from Figure. 7(a) that both the pressure loss and the torque increase with the increase of the helix angle of the impeller. When the helix angle is increased from 25° to 55°, the torque is increased by 94% and the pressure loss is increased by 204%. The final circumferential velocity of the drilling fluid becomes larger as the helix angle become larger. According to the momentum conservation, the torque acting on the impeller is greater.

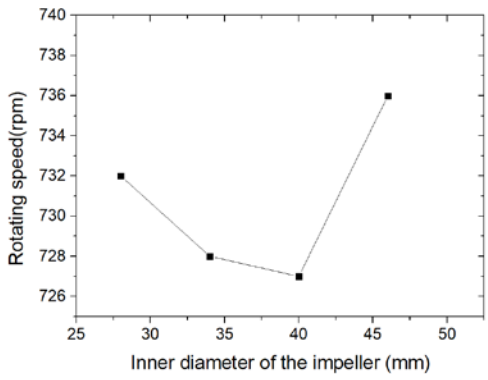
According to Figure. 7(b), both the pressure loss and the torque increase as the inner diameter of the impeller increases. When the inner diameter was increased from 28 mm to 46 mm, the torque and pressure loss is increased by 47% and 53%, respectively. The average radius of the drilling fluid acting on the impeller increases as the inner diameter of the impeller increases, so the same pressure produces a greater torque. At the same time, as the flow area becomes smaller, the flow rate increases and the pressure loss increases.

It can be seen from Figure. 7(c) that both the pressure loss and the torque increase with the increase of the number of blades, but the variation range is relatively small, which are 23% and 9%, respectively. As the number of blades increases, the flow area and pressure loss increase due to the blade having a certain thickness.

It can be seen from Figure. 7(d) that the pressure loss and torque increase with the increase of the drilling fluid displacement, and the increment speed becomes more and more obvious. With the displacement increases from 15L/s to 40L/s, the pressure loss and torque get increased by 660% and 608%, respectively.



(a)



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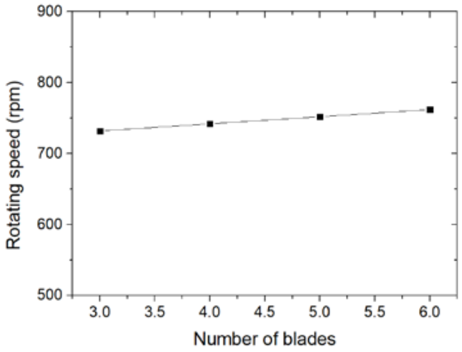

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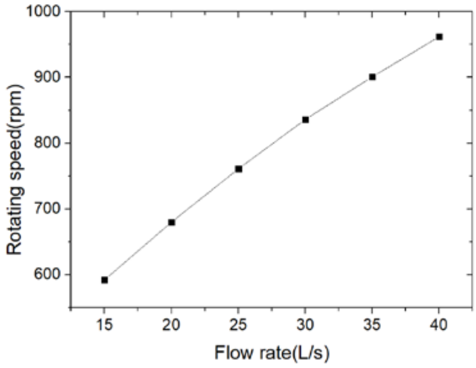


(d)

Figure.8 Rotating Speed Changes with (a) Helix Angle of the Blade,(b) Inner Diameter of Impeller,(c) Number of Blades,(d) Flow Rate of Drilling Fluid

Figure 8 is a graph showing the changes of the rotating speed with the impeller helix angle, the inner diameter of the impeller, the number of blades and the displacement of the drilling fluid. It can be seen from those curves that the rotary speed of the impeller significantly increases with increasing helix angle and increasing flow rate. However, increasing the number of blades seems to have few effects on improving the rotary speed. Moreover, there exists a minimum rotary speed along with increasing inner diameter of the impeller, which may be related to the interaction of the inertia and the torque.

These numerical simulation results corresponds well with the theoretical results shown in Figure 4. As for the small difference in the data, the numerical simulation process does not take into account the blocking effect of the dynamic disc valve on the drilling fluid, so that more energy from the drilling fluid is converted into mechanical energy of the impeller.

4.2.2 Numerical Simulation Analysis of Pulse Generating Mechanism

In a pulse generation cycle, the pressure mapping and velocity mapping at different times are shown in Figure. 9 and 10, respectively. It can be seen from the figure that when the position of the holes of the dynamic disc valve is shifted away from the holes of the static disc valve, the pressure in the holes of the dynamic disc valve rapidly increases, and the speed drops sharply. Again, when the position of the holes of the dynamic disc valve coincides with the position of the holes of the static disc valve, the pressure is reduced and the speed tends to be high. The continuous rotation of the dynamic disc valve causes the overcurrent area to change periodically, resulting in fluctuations in pressure and speed. According to the numerical simulation results, the influence of each parameter on the amplitude, frequency and pressure loss of pressure pulse is analyzed.

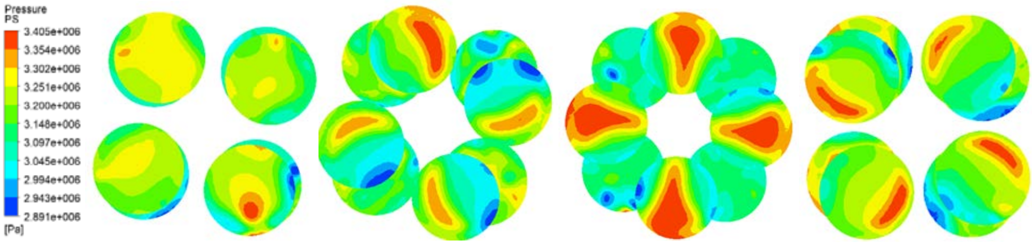


Figure.9 Pressure Distribution of the Pulse Generation Mechanism

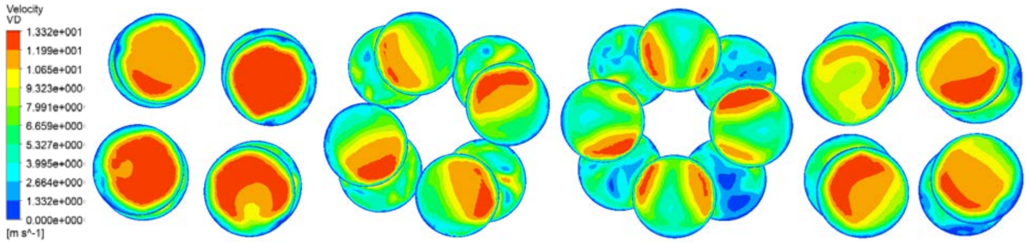
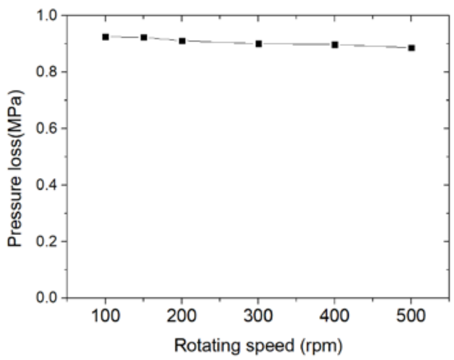
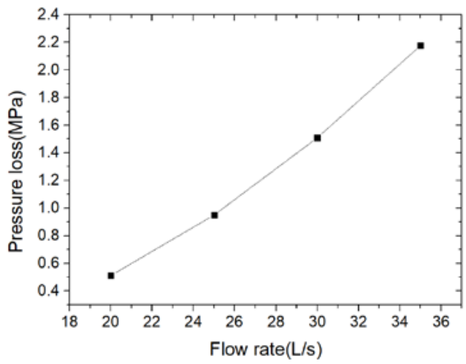


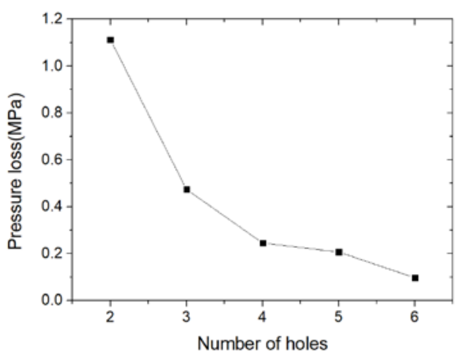
Figure.10 Velocity Distribution of the Pulse Generation Mechanism

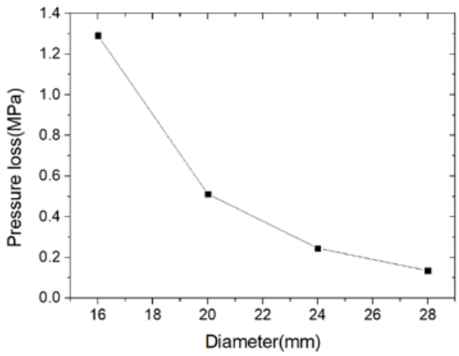
Figure 11 shows the relationship between the pressure loss and the parameters of the pulse generating mechanism. As shown in Figure. 11 (a), the rotation speed does not cause a change in the pressure loss since the rotation speed does not affect the flow area. As shown in Figure. 11(b), when the displacement increases, the flow rate of the drilling fluid increases, and the pressure loss gradually increases. As shown in Figure. 11 (c) and Figure. 11 (d), the pressure loss is gradually reduced due to the increased flow area, which is caused by an increase in the number of holes and the diameter of the holes.





(a) (b)



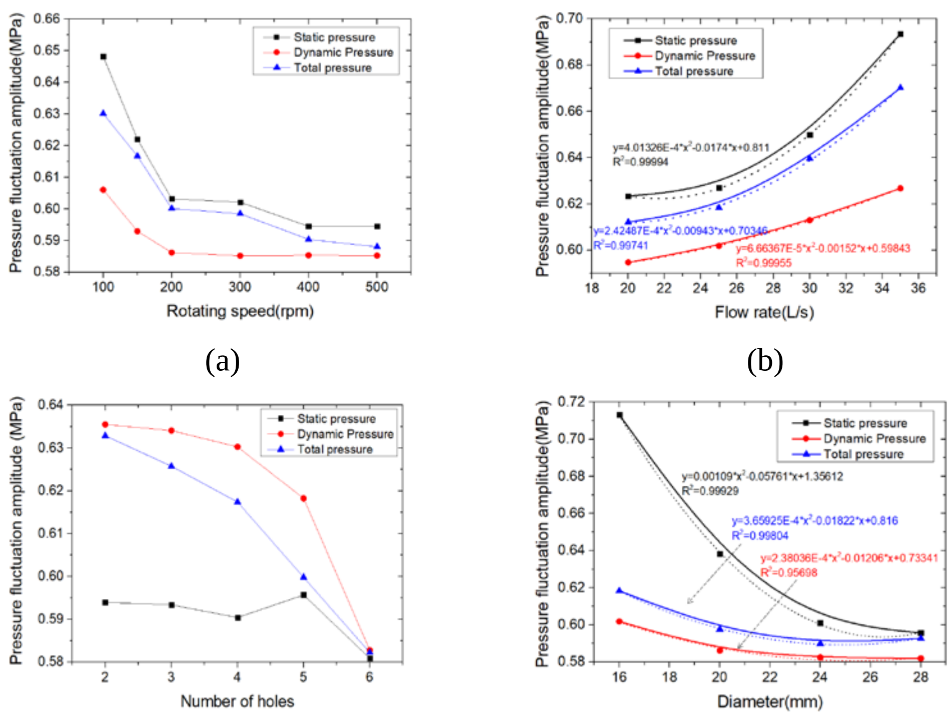


(c) (d)

Figure.11 Pressure Loss Changes with (a) Rotating Speed, (b) Flow Rate, (c) Number of Holes, (d) Diameter of Holes

The pulsation amplitudes of dynamic pressure, static pressure and total pressure are counted and plotted in Figure.12. When the rotation speed increases, the pulsation amplitude gradually decreases, and the pulse effect also decreases. The increase of the rotation speed causes the overcurrent area to alternate. When the frequency of the pulsation is higher than the propagation speed of the pulse effect, the pulse effect is weakened, resulting in a slow response of the fluid to the pressure and a decrease in the pulsation amplitude.

Figure. 12(b) shows that as the displacement increases, the flow rate increases, the pulse effect is gradually enhanced, so the pulsation amplitude increases. According to the curve, the relationship between the pulsation amplitude and the displacement can be well fitted by quadratic equations, whose coefficients of correlation are close to 1.



(a) (b) (c) (d)

Figure.12 Pressure Fluctuation Amplitude Changes with (a) Rotating Speed, (b) Flow Rate, (c) Number of Holes, (d) Diameter of Holes

Figure. 12(c) and Figure. 12(d) respectively describe the influence of the number and the diameter of the holes on the amplitude of the pulsation. When the diameter of the holes is constant, increasing the number of holes brings increased flow area and thus result in a decrease in the pulsation amplitude. On the other hand, when the number of holes of the disc valve is constant, increased diameter of the holes of the disc valve also causes an increase in the flow area, thereby causing a decrease in the amplitude of the pulsation. The effect of the diameter of the holes on the amplitude of the pulsation can also be well fitted using a quadratic function.

As shown in Figure 13, the effects of each parameter on the pulse frequency are also drawn out. Along with increasing rotation speed, the pulse frequency linearly increases. However, the displacement and the diameter have no significant effects on the pulse frequency, which is showed Figure 13 (b) and Figure 13 (d). The pulse fluctuation frequency generated by the pulse generating device is mainly affected by the variation law of the flow area. Since the displacement and the diameter of the holes do not affect the change frequency of the flow area, there is no influence on the pulse frequency of the pulse generating device. The higher the rotation speed, the higher the frequency of change of the overcurrent area, resulting in an increase in the frequency of the fluctuation. As shown in Figure.13(c), when the number of holes of the disk valve increases, the frequency of the pulse fluctuation increases, because the relative distance of the disk valve hole becomes smaller as the number of holes increases. As the dynamic disk valve rotates, the overcurrent area changes more and more quickly, thereby causing an increase in the pulse frequency.

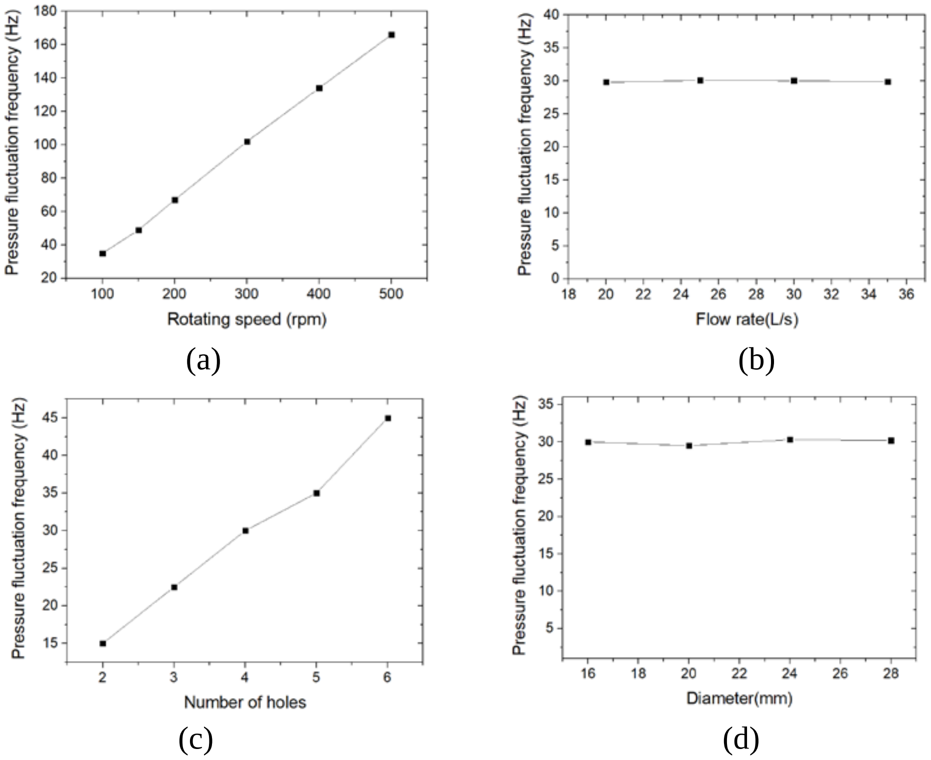


Figure.13 Pressure Fluctuation Frenquency Changes with (a) Rotating Speed, (b)Flow Rate, (c) Number of Holes, (d) Diameter of Holes

5. CONCLUSION

In this paper, a disc-type pulsed jet generating device was proposed, which used the impeller to drive the dynamic disc valve to rotate, and modulates the continuous drilling fluid flow into a pulse jet to act on the bottom hole. The theory combined with the numerical simulation analysis method is used to analyze the influence of the structural parameters of the power mechanism and the pulse generating mechanism. By comparing the cases set in this article, we found that:

(1) The rotation speed of the impeller varies from 500 to 1200 rpm. In general, the rotation speed significantly increases with increased helix angle and increased flow rate. However, it seems to be little affected by the number of blades and it should has a minimum value when the inner diameter of the impeller falls in a range of 20 to 50 mm.

(2) Both the pressure loss and the torque increase with the impeller helix angle, the inner diameter of the impeller, the number of blades and the displacement of the drilling fluid. Specifically, when the helix angle is increased from 25° to 55°, the torque is increased by 94%, and the pressure loss is increased by 204%. When the displacement is increased from 15L/s to 40L/s, and the pressure loss and torque are increased by 660% and 608%, respectively.

(3) For the pulse generating mechanism, the change of the rotation speed does not cause the change of the pressure loss; As the displacement increases, the pressure loss increases linearly; And as the number of holes and the diameter of the holes increase, the flow area increases and the pressure loss gradually decreases.

(4) The pulsation amplitude increases with increased flow rate and decreased values of rotation speed, the hole diameter and the number of holes. Especially, the effects of the flow rate and the hole diameter on the pulsation amplitude can be well descripted by quadratic functions.

(5) The pulse frequency significantly increases with increased rotation speed and increased number of holes. Nevertheless, there is no significant effects of the displacement and the hole diameter on the pulse frequency.

(6) To achieve a high pulse amplitude and a low pulse frequency, the disc-type pulse jet generating tool should have a small impeller helix angle, an inner diameter of the impeller within (30mm, 40mm), a small number of disc holes, and a small hole diameter. Considering the minimization on the press loss through the tool, the number of disc holes is suggested to be 4 while the hole diameter should be about 24 mm.

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NOMENCLATURE

D_1 Outer diameter of the impeller, mm;
D_0 Diameter of the sleeve, mm;
H Thickness of the blade, mm;
R Radius of the contact face, mm;
K Coefficient of friction;
ρ Fluid density, $kg\cdot m^{-3}$;
Q Flow rate, $L\cdot s^{-1}$;
$\overline{D}$ Middle diameter of the impeller, m;
C_2 Axial velocity, $m\cdot s^{-1}$;
C_z Actual axial velocity, $m\cdot s^{-1}$;
u_2 Circumferential velocity , $m\cdot s^{-1}$;
β Helix angle, rad;
ψ Excretion coefficient;
z Thickness of blade, m;
c Number of blades;
b Radial length of blade, m;
m Mass, kg;
μ Friction coefficient;
M Torque, $N\cdot m$;
M_b Friction torque, $N\cdot m$;
N Rotating speed, rpm.