



RESEARCH ON THE KINEMATIC CHARACTERISTICS OF AN IMPACT-SCRAPING DRILLING TOOL

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ABSTRACT

The formation structures of oil resources vary significantly across different regions, causing severe difficulties in deep exploration. Consequently, conventional drilling tools suffer from poor rock-breaking efficiency and low penetration rates. To improve efficiency and reduce premature cutter wear, this study proposes an impact-scraping drilling tool in which the axial impact and rotary cutting functions of the bit are separated but coordinated. This specialized tool utilizes the rotational energy of positive displacement mud motors, converting it into mechanical impact energy via a cam-roller system and a compression spring. It delivers periodic axial impacts directly to the rock layer, while the external bit independently performs standard rotary cutting. The tool's theoretical kinematics and dynamics were modeled using the Lagrange multiplier method alongside an equivalent spring damping model. Motion characteristics were numerically simulated under input rotational speeds of 120, 150 and 180 rpm. Additionally, a simplified bench experiment validated the drill bit's rock-impacting state. The findings clearly demonstrate that as rotational speed increases, the roller's effective stroke decreases, while the impact frequency and maximum axial impact force increase. Notably, the maximum impact force reached 19479 N at 180 rpm. The experimental impact force and frequency showed good agreement with the simulation results, with amplitude consistency above 88%. These findings confirm that the proposed impact-scraping drilling tool can generate stable periodic axial impact, enhance rock-breaking efficiency, and potentially reduce bit cutter failure.

Keywords: impact-scraping; kinetic characteristic; impact parameter; vibration.

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1. Introduction

After long-term and large-scale exploration, development and use, it has become increasingly difficult to increase the storage capacity of conventional oil and gas resources, and the contradiction between supply and demand has become increasingly prominent [1-2]. Against this backdrop, the global oil and gas industry has advanced technological progress in multiple industrial chains to cope with the supply-demand pressure: in the field of petroleum chemical materials, Babayev et al. [3] systematically summarized the synthesis routes, performance regulation laws and industrial application scenarios of dicyclopentadiene-based fuel hydrocarbons, providing a clear research framework for the development of high-performance special fuels; in terms of low-carbon development and enhanced oil recovery, Shakhverdiev et al. [4] discussed the development prospects of CO₂ enhanced oil recovery projects in the Russian Federation, and demonstrated the dual value of improving oil recovery and realizing carbon sequestration; in deep geological exploration, Yetirmishli et al. [5] adopted seismic tomography to study the deep crustal structure of

Azerbaijan and the Caspian region, and clarified the regional deep structural characteristics and hydrocarbon accumulation geological background, providing key geophysical support for deep oil and gas exploration. Meanwhile, oil and gas exploitation has gradually developed towards deep and ultra-deep wells. For the development of deep single-well formations, Abdullayev and Gamzaev [6] proposed a calculation method for pressure distribution in the elastic mode of single-well formation development, which improved the theoretical calculation model of single-well development dynamics, and can provide theoretical support for productivity evaluation and development parameter optimization of complex lithologic reservoirs. However, with an increase in depth, the hardness, plasticity and uneven distribution of the rock increase, and the drillability decreases, which leads to a slow rate of penetration (ROP) and low rock-breaking efficiency [7].

Rotary impact drilling technology is an effective method for improving ROP [8]. Rock breaking is mainly achieved by applying axial and torsional impacts to the drill bit [9], and periodic impact can cause large fragmentation of rock. Over the years, practical applications have demonstrated that rotary impact technology can effectively improve the ROP in hard formations [10], and the drilling rate is approxi-

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mately 40% faster than that of traditional drilling [11]. Rotary impact drilling with mud as an energy carrier can not only resist high temperature, high pressure, corrosion, and other extreme environments, but also does not need to stop drilling and replace the BHA when drilling through soft formations or formation changes. Timonin [12] proposed technical and economic evaluation criteria for the effective application of rotary percussion drilling tools, and successfully applied them to field tests [13]. Xin et al. [14] designed different rotary impact tools, all of which showed a significant increase in the drilling rate in field tests.

At present, considering the motion characteristics of rock in the process of rock breaking, there are few studies that analyze the relationship between the impact parameters and rock-breaking efficiency, and the theories are not mature [15]. Song et al. [16-17] proposed a new rotary-impact drilling method to address the shortcomings of traditional rotary-impact drilling, such as low impact frequency and insufficient utilization of impact energy. The authors analyzed some key parameters of the alternating shear stress and impact energy. The results show that an increase in the amplitude of the alternating shear stress aggravates the fatigue failure of the rock. The influence of parameters such as the impact frequency and, dynamic force on the impact energy was analyzed using a theoretical method [18]. A full-scale drilling test was conducted, and the results showed that the optimal frequency was higher than the resonant frequency of the rock. The impact energy increases with an increase in the amplitude of the dynamic load. Karpov et al. [19] optimized different types of rotary impact drilling processes based on rock volume failure energy criteria and fracture energy content. The estimation criteria for this value range were proposed to ensure drilling with minimal energy and bit wear.

The stick-slip vibration of polycrystalline diamond compact (PDC) bits causes low ROP and premature bit wear. To solve this problem, the combined percussion-rotary drilling approach (CPRDA) compound drill tool [20] was developed and tested in two hard-rock wells. Coupled impact produces a high frequency, high amplitude rotation speed, and low torque, which slows down the accumulation of cutting force and the occurrence of the stick-slip effect [21]. The results show that CPRDA can increase the ROP by 56.1~60.4 % and prolong the bit life. In terms of performance optimization of drilling bit components, Bashmur et al. [22] proposed a hybrid bionic strategy combined with CFD simulation to enhance the static characteristics of roller-cone bit bearings, providing a new technical approach coupling bionic engineering and fluid mechanics for the performance upgrade of key drilling tools. A rotary hydraulic impact-specific energy model of the PDC bit and roller cone bit (RCB) was established. Practical application results show that the efficiency

of using RCB with a hydraulic impactor is better than that of using PDC with hydraulic impactor [23]. According to the theory of rotary cutting rock and shock wave drilling, the prediction method of rotary drilling speed was established, and the measurement of rotary drilling rig parameters and core index was discussed [24]. Concerning the actual size of the PDC bit cutters and the load transferred to the formation, a corresponding numerical model was established for analysis. The results show that rotary impact drilling can improve the dynamic-to-static load ratio, which is conducive to improve the penetration depth and formation erosion volume fraction [25].

Considering the above factors, this paper proposes a type of impact-scraping drilling tool, which can separate the impact and rotary cutting actions. The impact system enables the impact bit to generate an axial impact that directly impacts the rock formation. Simultaneously, the external PDC bit is not affected by the impact and can normally complete the rotary cutting effect. The tool can significantly improve the ROP and reduce the cutter failure rate. By changing the rotation speed of positive displacement mud motors (PDM), the motion state of the roller, the impact between the drill bit and rock and the contact between the roller and cam are simulated. The experimental axial impact force was compared with the simulation data, and the motion characteristics of the tool were studied at different rotation speeds. This is of vital engineering and social significance to the development and improvement of compound drilling tools.

2. Impact-scraping drilling tool working mechanism

As shown in figure 1, the impact-scraping drilling tool is mainly composed of a universal joint, water cap, driving shaft, cam-roller system, shock shaft, compression spring, impact drill bit, etc. And the cam-roller system is shown in figure 2.

The working mechanism of the impact-scraping drilling tool is as follows: When drilling normally, the weight on bit (WOB) and torque are transmitted to the bit by the outer shell of the drilling tool. The external drilling fluid enters the PDM, converts the hydraulic energy into mechanical energy of the rotor, and transmits it to the drive shaft through the universal joint and water cap. This causes the driving shaft to rotate, and drives the cam mechanism to rotate with it. The cam is matched with the roller such that the cam pushes the roller to rotate during the rotation process (refer to fig. 2a), and the roller base is connected to the lower shaft of the shock. The upward movement of the roller system will inevitably drive the lower shaft of the shock and the impact bit to lift, and compress the spring between the lower casing and the shaft of the shock, so that the spring can store energy. When the roller climbs to the highest point of the cam, the energy

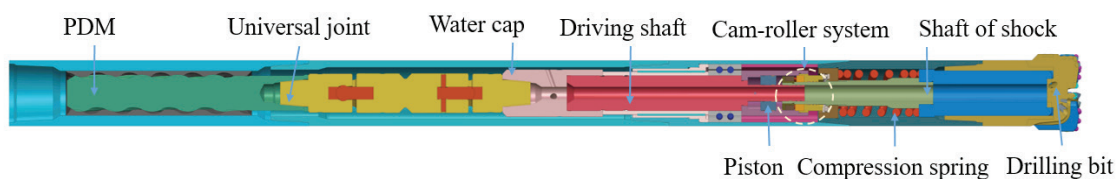


Fig. 1. Structure diagram of impact-scraping drill tool

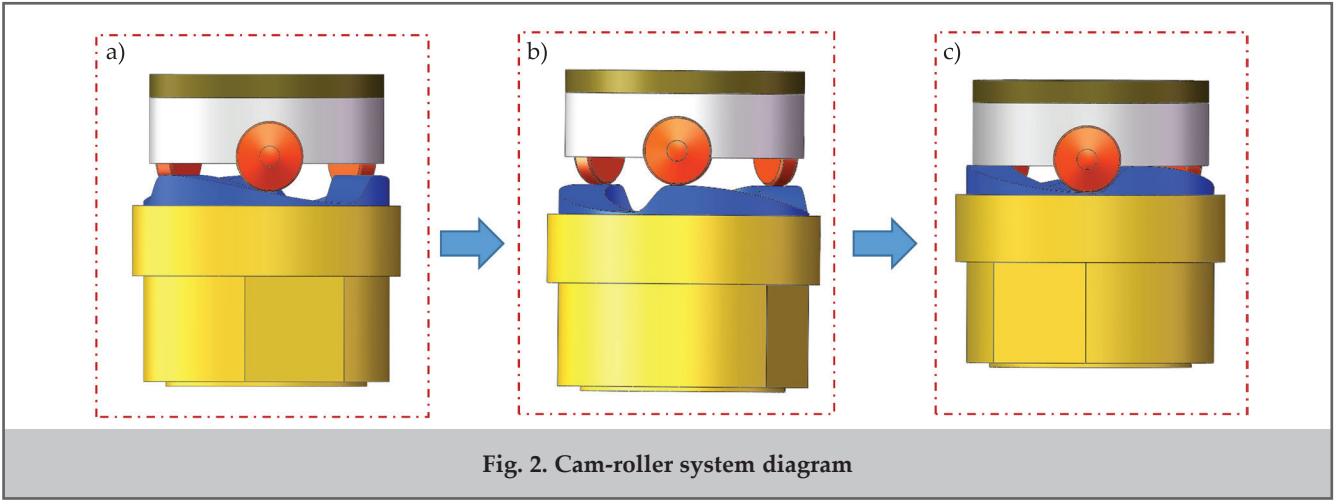


Fig. 2. Cam-roller system diagram

storage of the spring is at its maximum (fig. 2b). After passing this point, the roller accelerates to fall and collides with the lowest point of the cam to cause the spring to release energy to rebound (refer to fig. 2c). Under the double action of the cam-roller system and compression spring, the impact bit produces an impact load and acts directly on the rock formation at the bottom of the well. The PDM continues to transfer the rotational drive energy downward, thereby causing the cam to rotate. As the roller repeatedly rises and falls, the spring compresses and releases.

3. Theoretical modeling and simulation analysis of drilling tools

3.1. Basis theory

The kinematic characteristics of the virtual prototype were analyzed by considering the center-of-mass Cartesian coordinates of the rigid body G_i and the Euler angles reflecting the orientation of the rigid body as generalized coordinates. The generalized coordinates of the center of mass of the rigid body were $q_i = [x, y, z, \psi, \theta, \varphi]^T$. The generalized matrix of the entire mechanical system can be expressed as $q = \{q_1^T, q_2^T, q_3^T, \dots, q_m^T\}^T$ [26]. The Lagrange multiplier method was used to establish the entire mechanical system dynamics equation:

$$\frac{d}{dt} \left(\frac{\partial T}{\partial \dot{q}} \right)^T - \left(\frac{\partial T}{\partial q} \right)^T + f_q^T \rho + g_q^T \mu = Q \quad (1)$$

When the whole system is fully constrained:

$$f(q, t) = 0 \quad (2)$$

When the system is not holonomically constrained:

$$g(q, \dot{q}, t) = 0 \quad (3)$$

where T is the kinetic energy of the system; q is the generalized coordinate matrix of the system; Q is a generalized force matrix; ρ is a Lagrangian multiplier matrix with holonomic constraint; μ represents the Lagrangian multiplier matrix with non-holonomic constraint; and $\dot{q}$ represents the generalized velocity matrix.

Common methods for rigid body contact collision analysis include recovery coefficient method and equivalent spring damping method [27]. Compared with the recovery coefficient method, the equivalent spring damping method equates the collision process to a spring damping system. The contact force can be calculated and the contact collision

process can be simulated more realistically [28]. Therefore, the impact collision of the cam-roller system in this study was analyzed using the equivalent spring damping model. It is assumed that the collision process of the equivalent damped spring model does not occur instantaneously. When the two components collide and produce deformation, a contact force is generated. The force is related to the size and speed of the collision deformation [29].

The contact force between the two components was calculated according to the impact function model in the equivalent spring damping method, and the generalized expression is as follows:

$$F_n = K\delta + C\dot{\delta} \quad (4)$$

where F_n is the normal contact force in N; K is the Hertz contact stiffness in N/m; δ is the normal penetration depth of the contact point in m; C is the damping coefficient; and $\dot{\delta}$ is the normal relative velocity of the contact point.

The contact stiffness K can be obtained from Hertz contact theory:

$$K = \frac{4}{3} \left(\frac{1}{R_1} + \frac{1}{R_2} \right)^{-\frac{1}{2}} \left(\frac{1-\mu_1^2}{E_1} + \frac{1-\mu_2^2}{E_2} \right)^{-1} \quad (5)$$

where R_1 and R_2 are the radii of curvature of the two collision components from the point of collision, mm; E_1 and E_2 are the elastic modulus of the two colliding members, MPa; μ_1 and μ_2 are Poisson's ratios of the two colliding components, dimensionless.

The damping coefficient C was calculated using the modified hysteresis damping coefficient proposed by Lankarani et al [30]. The calculation formula is as follows:

$$C = \frac{3K(1-e^2)}{4u} \delta^n \quad (6)$$

where e is the coefficient of elastic recovery; n is the nonlinear elastic force index; u collision velocity, m/s.

3.2. Simplified geometric physical model

In order to shorten the calculation time, the model of the cam-roller impact system was simplified when a virtual prototype was established. A simplified geometric physical model is shown in figure 3. The drill tool was driven by the PDM to rotate the cam-roller impact system, and the input flow of the PDM was 63.5, 79 and 95 L/s (the conversion speeds were 120, 150 and 180 rpm) for simulation (table).

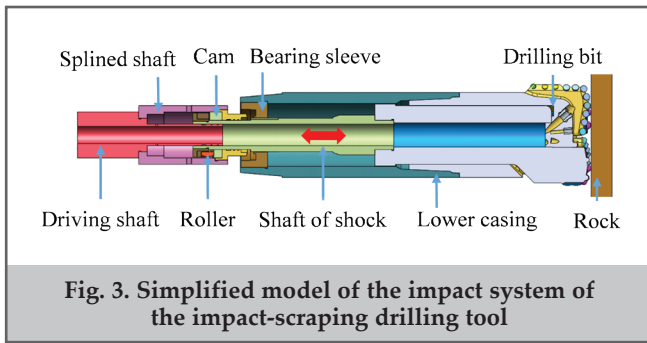


Fig. 3. Simplified model of the impact system of the impact-scraping drilling tool

Contact collision parameter		Table
Parameter name	value	Unit
Model material	Alloy steel	/
Density ρ	7801.0	kg/m ³
Modulus of elasticity E	2.07×10^{11}	N/m ²
Poisson's ratio μ	0.29	/
Contact stiffness K	1.5981×10^{10}	N/m
Penetration depth δ	10^{-4}	m
Force index c	1.8	/
Damping coefficient C	10^4	N·s/m
Coefficient of dynamic friction μ_s	0.1	/
Coefficient of static friction μ_d	0.3	/

3.3 Analysis of roller

The displacement and velocity of the roller during the working process were measured. Simultaneously, the simulation results reflect the impact of the roller motion on the next stage. By changing the flow rate, the rotor speed of the PDM was set to 120, 150 and 180 rpm. The drive shaft transmits the power of the PDM to the cam, which pushes the roller until it reaches the highest point of the cam and then accelerates down, and the cycle is repeated to achieve periodic movement. The displacement and speed change in the center position of the roller over time are measured, and the results are shown in figure 4.

The results show that when the rotation speed is 120, 150 and 180 rpm respectively, the displacement variation ranges of the roller are 0.2257-0.2369 m, 0.2257-0.2368 m and 0.2258-0.2368 m respectively. That is, as the speed increases,

the initial falling speed of the roller increases, which leads to a higher collision point between the roller and cam, so the effective push decreases and the movement frequency of the roller increases correspondingly.

The time domain and spectrogram of the roller speed under different speed conditions are presented in figures 5-7. The axial speed of the roller increases with an increase in the flow rate, and the instantaneous speed changes from about 0.6 m/s to about 0.7 m/s. That is, when the roller fell from the highest point of the cam, a large wave peak occurred every 0.167 s at 120 rpm, and the average peak speed was approximately 0.6 m/s. At 150 rpm, the period was 0.133 s, and the average peak speed was approximately 0.63 m/s. The period at 180 rpm was 0.111 s, and the average peak speed was approximately 0.69 m/s. With an increase in rotational speed, the faster the roller rotation speed, the greater the average peak speed, and the greater the impact frequency and impact force of the cam. The results indicate that the speed spectrum of the roller at different speeds is multi-peak structure. When the speeds were 120, 150 and 180 rpm, the main frequency was 5.9, 7.58 and 9.14 Hz, respectively. With an increase in the rotation speed, the dominant frequency of the roller speed increases, and the influence of the rotation speed on the frequency is evident.

3.4. Contact force analysis between roller and cam

The roller moves on the cam surface, and when the roller goes beyond the top of the cam, the roller will make a parabolic fall. There is no contact between the two until the impact is completed, and the roller continues to climb upward on the cam surface and forms a cycle. Figures 8-10 show the time-domain and spectrum results of the contact force between the roller and cam under different rotation speeds, and the curves all show obvious regularity. When the rotation speed was 120, 150 and 180 rpm, the maximum contact forces were 2204.3878, 3027.0221 and 3232.4545 N, respectively, and the average contact forces were 519.6847, 514.1789 and 479.4281 N, respectively. The non-contact times of the two components were 0.019, 0.016 and 0.013 s, respectively. The frequency of the contact force is 5.9 Hz (amplitude 176.5 N), 7.58 Hz (amplitude 167.6 N) and 9.14 Hz (amplitude 152.4 N), respectively. With an increase in the rotation speed, the non-contact time between the roller and cam gradually decreases, that is, the time of the return stage is reduced, the impact frequency generated per unit time is increased, and the amplitude decreases.

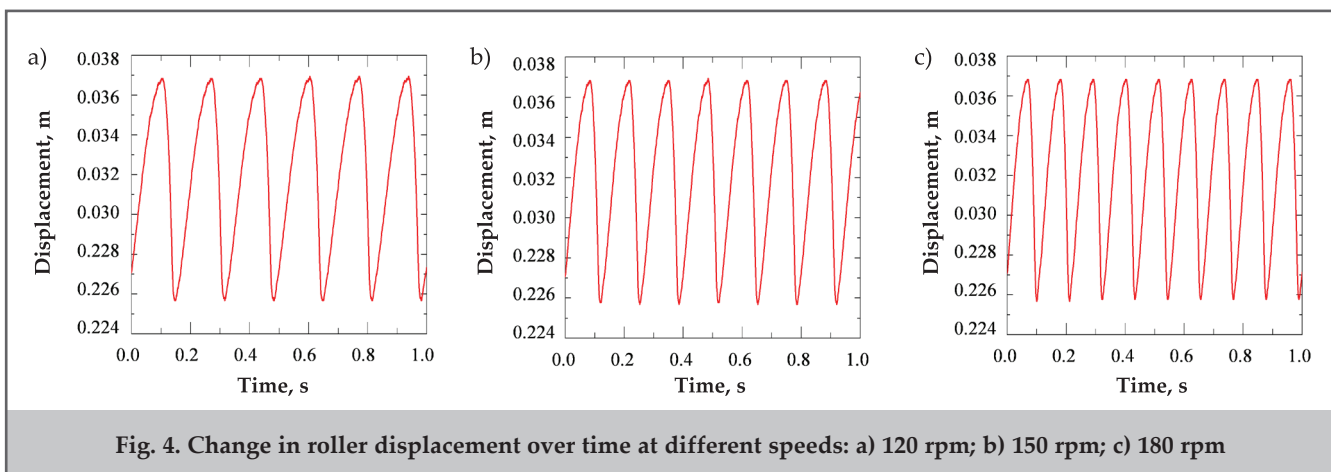


Fig. 4. Change in roller displacement over time at different speeds: a) 120 rpm; b) 150 rpm; c) 180 rpm

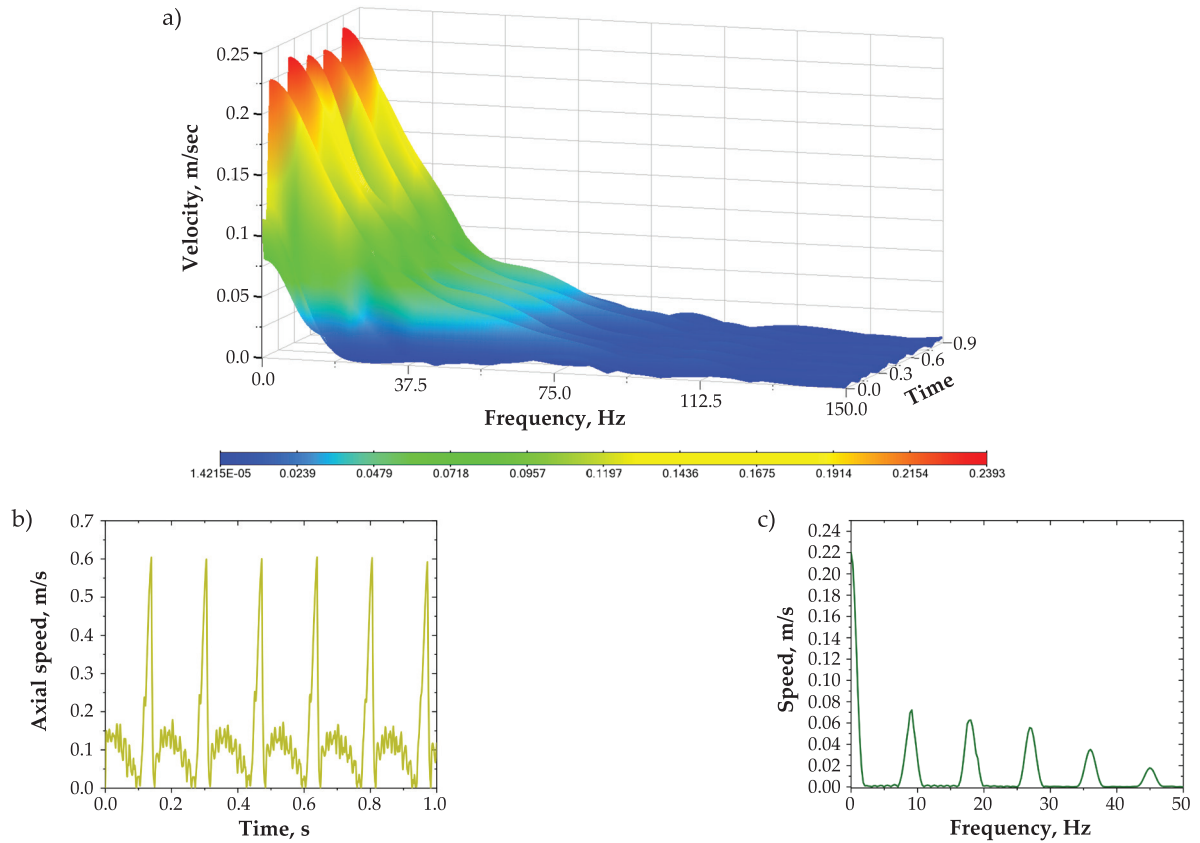


Fig. 5. Time domain and frequency domain diagrams of roller speed (120 rpm): a) 3D spectrogram of roller speed; b) time domain diagram of roller speed; c) frequency domain diagram of roller speed

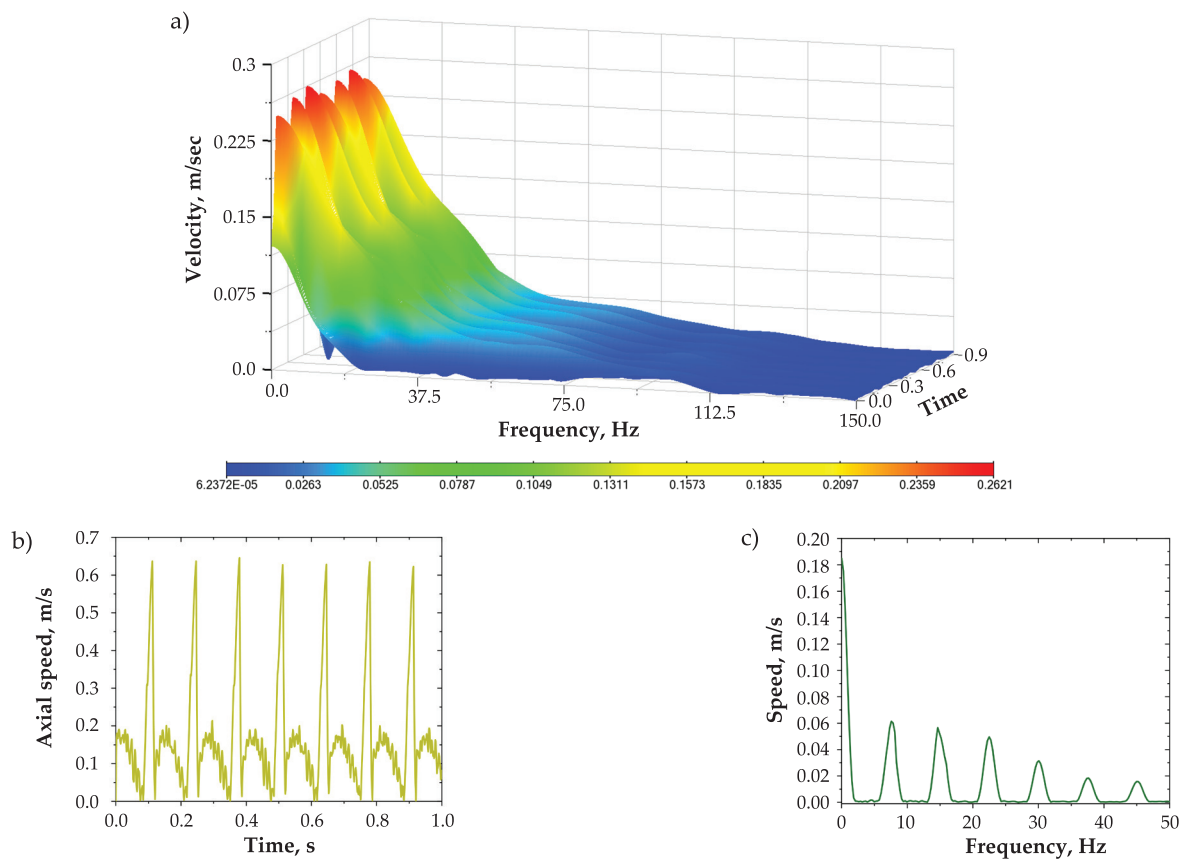


Fig. 6. Time domain and frequency domain diagrams of roller speed (150 rpm): a) 3D spectrogram of roller speed; b) time domain diagram of roller speed; c) frequency domain diagram of roller speed

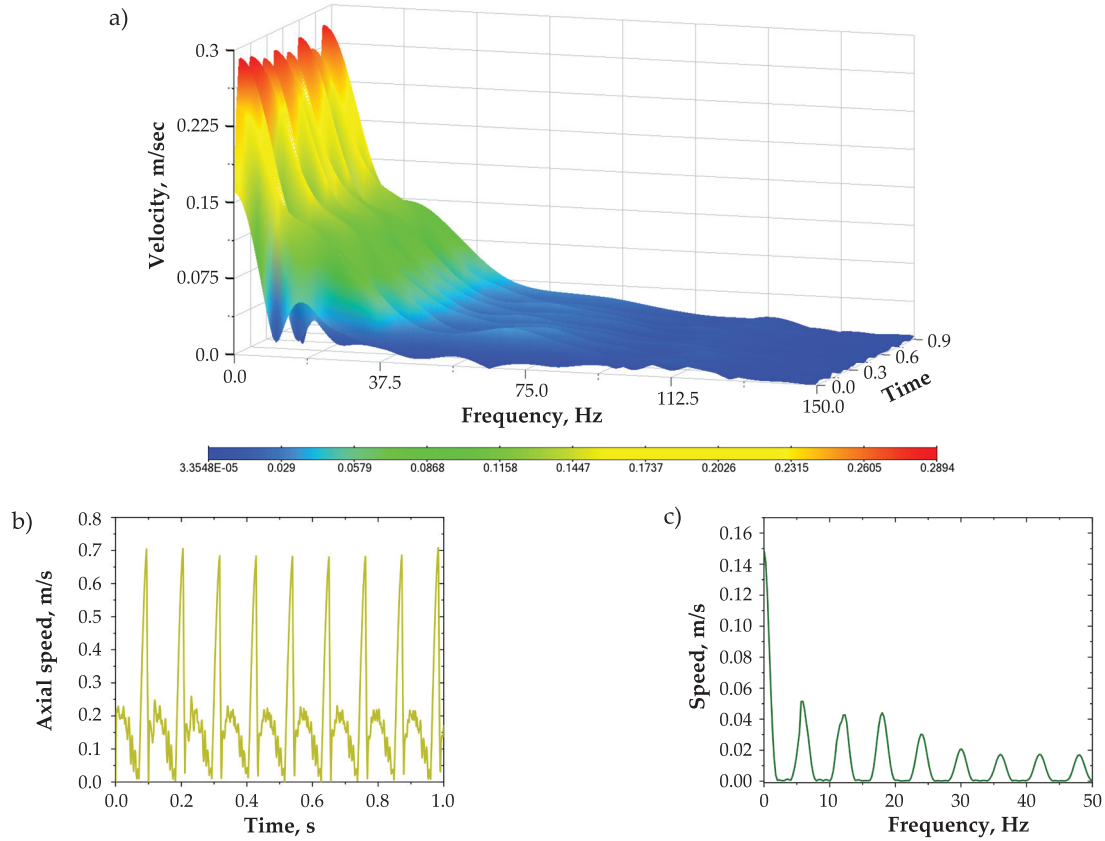


Fig. 7. Time domain and frequency domain diagrams of roller speed (180 rpm): a) 3D spectrogram of roller speed; b) time domain diagram of roller speed; c) frequency domain diagram of roller speed

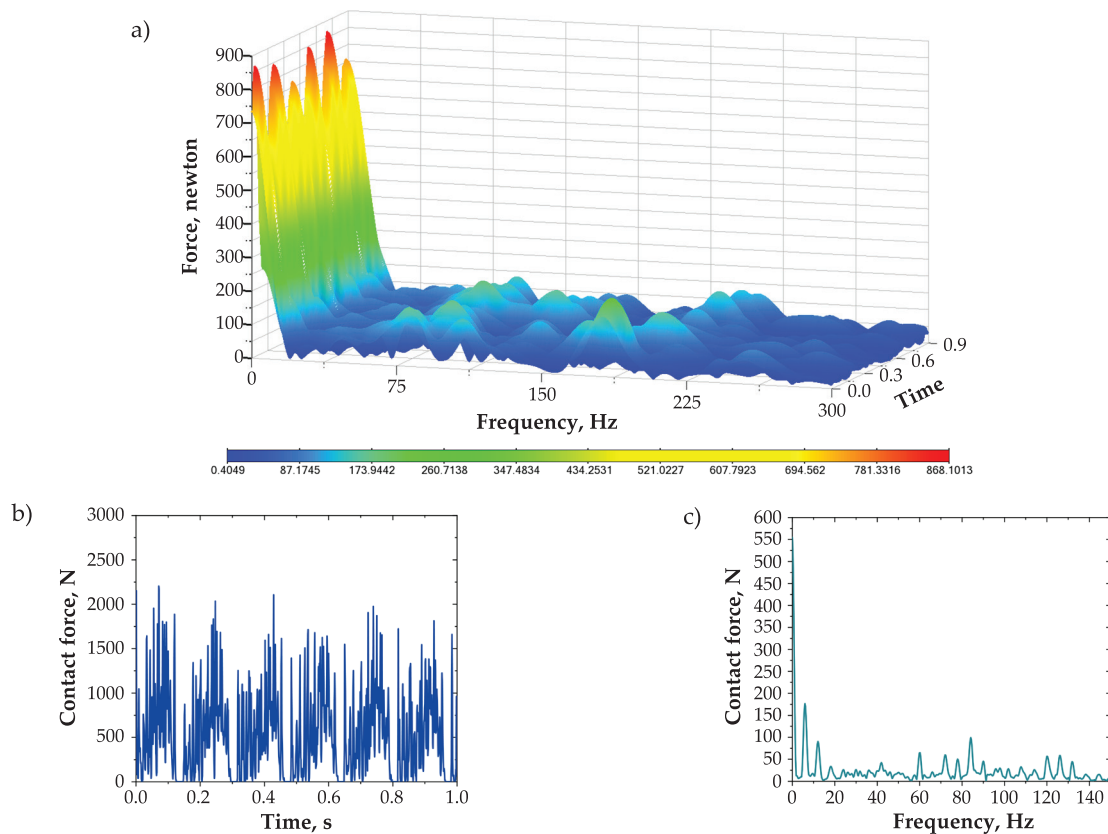


Fig. 8. Time domain and frequency domain diagram of cam-roller contact force (120 rpm): a) 3D spectrogram of cam-roller contact force; b) time domain diagram of cam-roller contact force; c) frequency domain diagram of cam-roller contact force

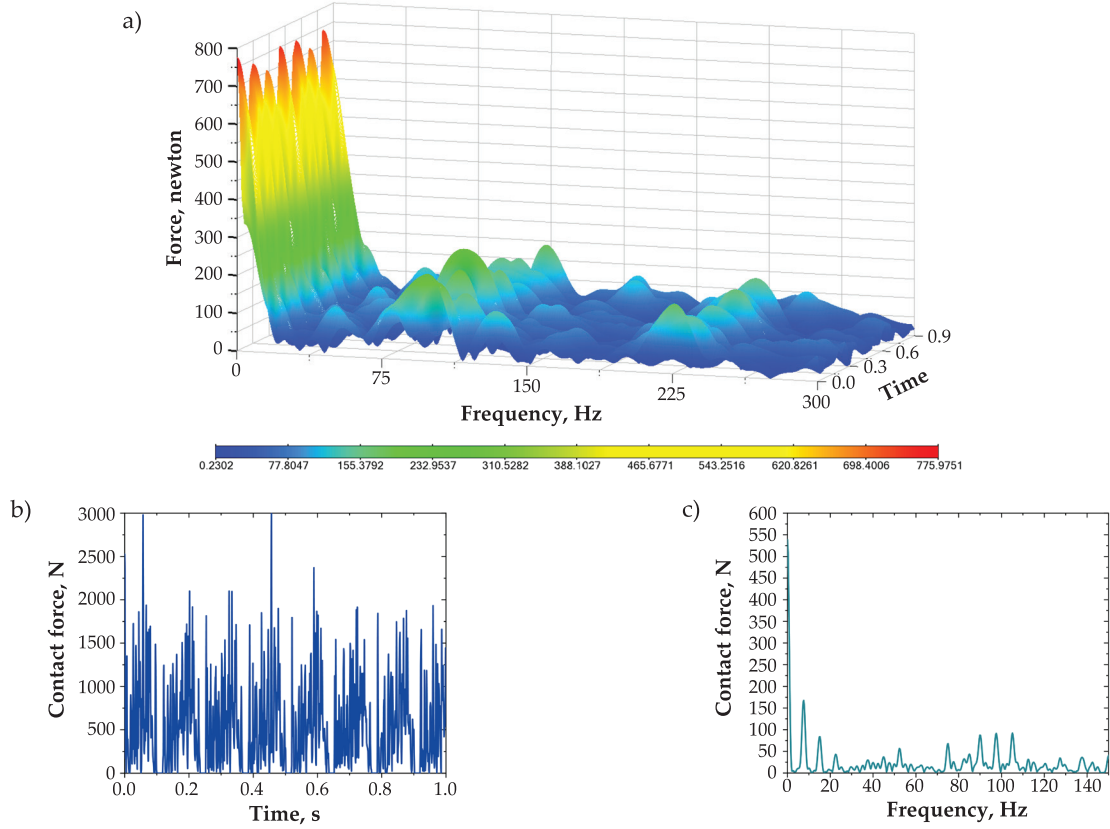


Fig. 9. Time domain and frequency domain diagram of cam-roller contact force (150 rpm):
a) 3D spectrogram of cam-roller contact force; b) time domain diagram of cam-roller contact force;
c) frequency domain diagram of cam-roller contact force

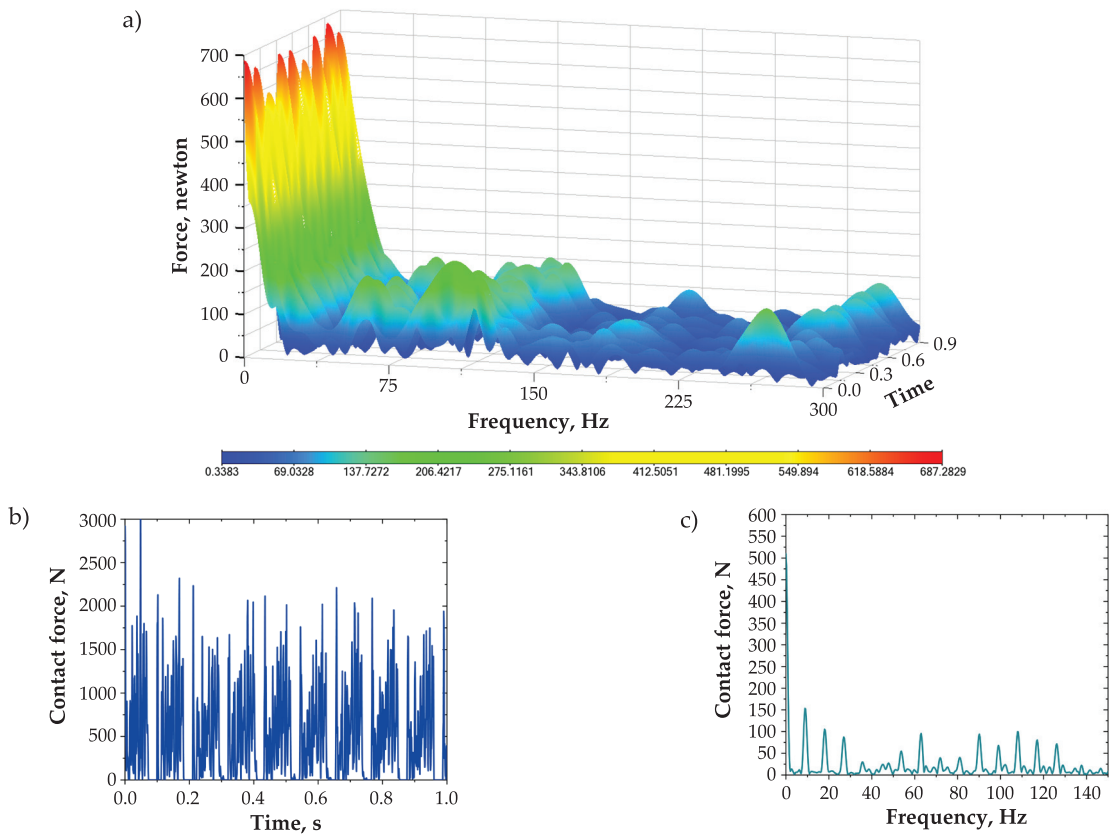


Fig. 10. Time domain and frequency domain diagram of cam-roller contact force (180 rpm):
a) 3D spectrogram of cam-roller contact force; b) time domain diagram of cam-roller contact force;
c) frequency domain diagram of cam-roller contact force

3.5 Impact analysis between drilling bit and rock

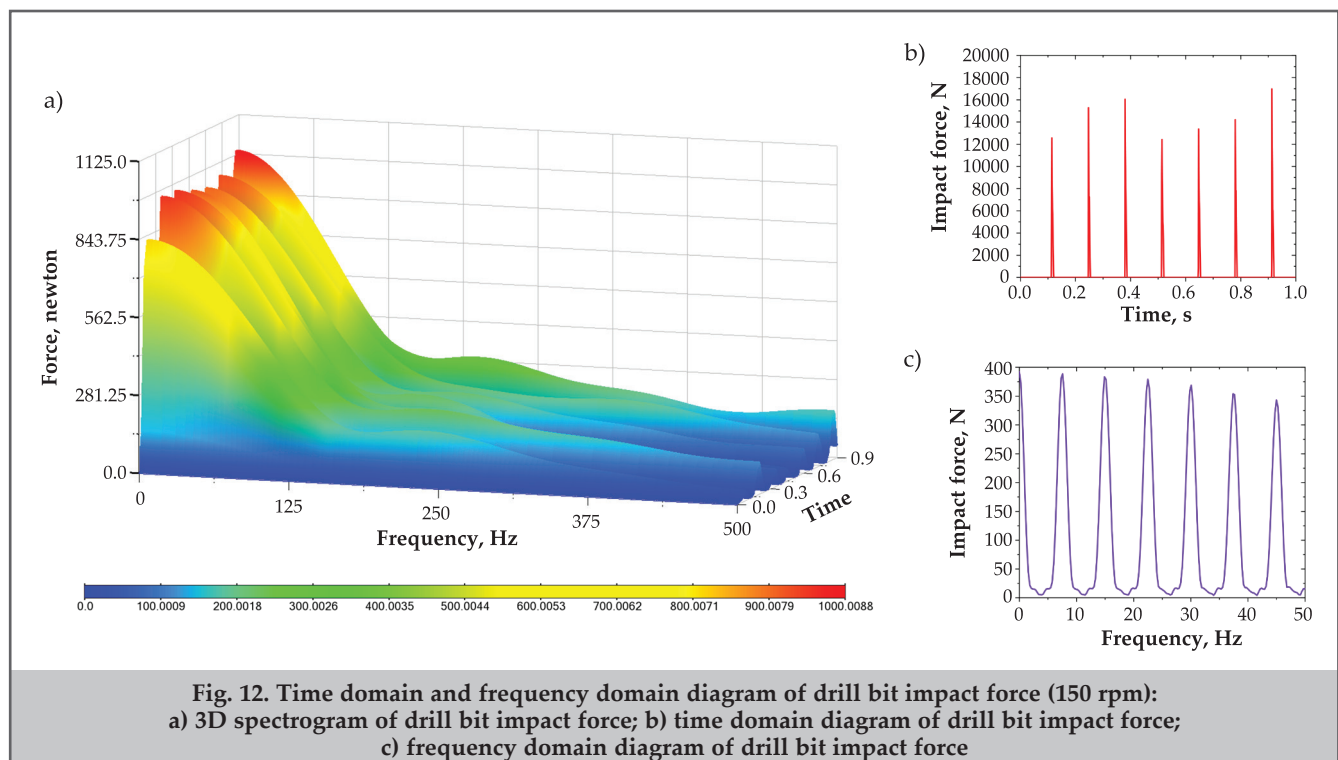
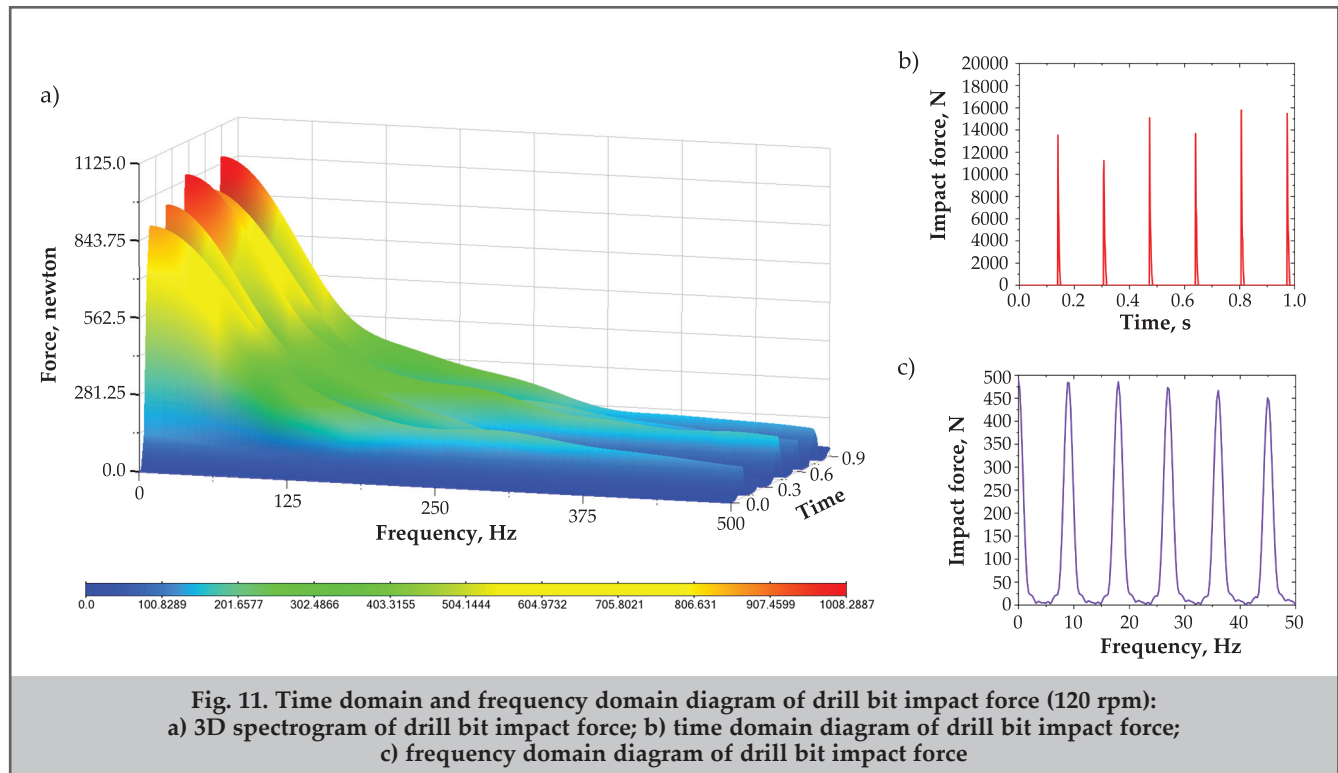
The drilling bit can provide periodic impact to the rock. Figures 11-13 show the time domain and spectrum curves of the axial impact force of the drilling bit under different rotation speeds. The axial impact frequency is stable, and the impact force spectrum at different rotation speeds is multi-peak, and its impact frequency increases with the increase of rotation speed. When the rotation speed was 120, 150 and 180 rpm, the maximum axial impact forces were 15786.2131, 16984.1656 and 19479.6913 N, respectively. The

tool produces 6, 7.5, and 9 shocks within 1s, with average impact forces of 14135.3971, 14411.2865 and 13189.9282 N respectively. The main frequencies are 5.9, 7.6 and 9.14 Hz, respectively. With an increase in rotational speed, the average axial impact force decreased.

4. Verification experiment

4.1. Experimental test

A verification experiment was conducted to study the motion state, impact performance, and design rationality of the impact-scraping drilling tool. The platform and related



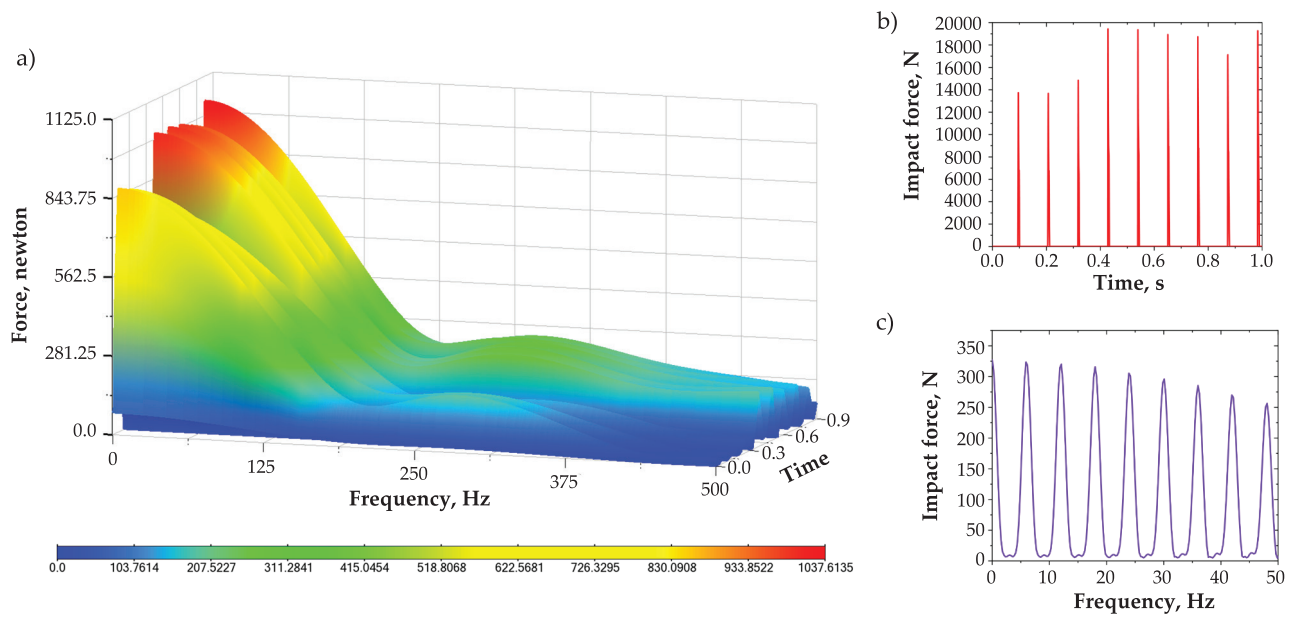


Fig. 13. Time domain and frequency domain diagram of drill bit impact force (180 rpm):
a) 3D spectrogram of drill bit impact force; b) time domain diagram of drill bit impact force;
c) frequency domain diagram of drill bit impact force

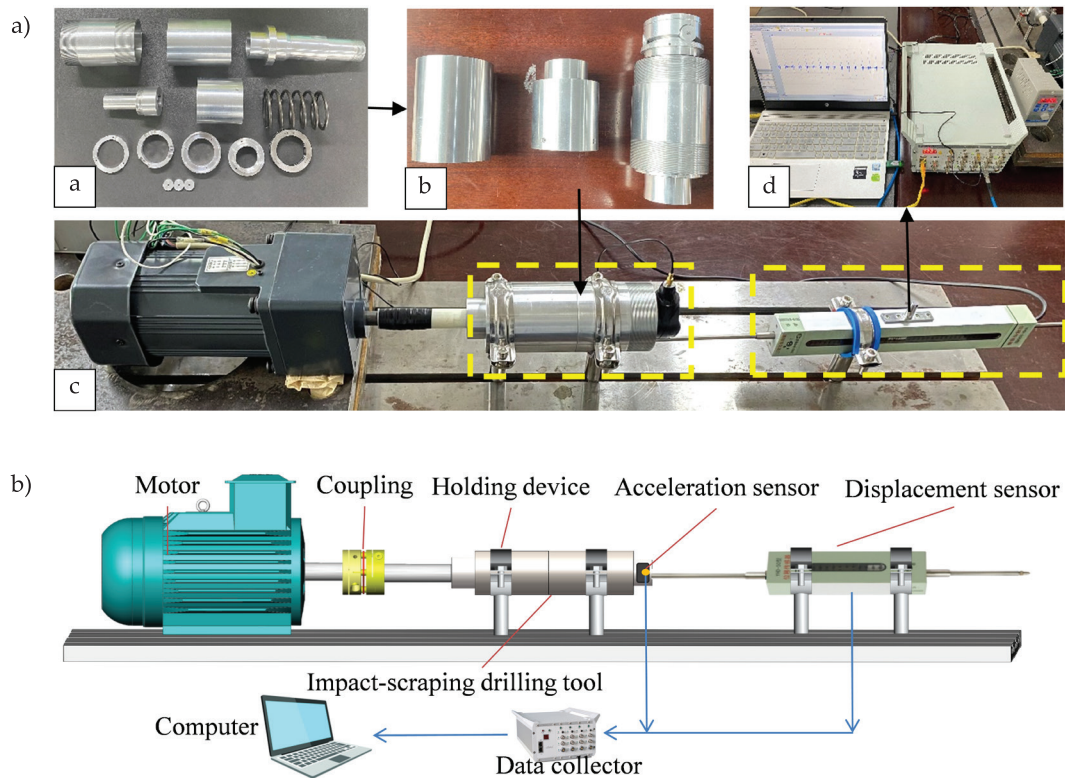


Fig. 14. Experimental apparatus

measurement equipment required for the experiment are shown in figure 14. The experiment used a steel pipe bracket to fix the upper and lower shell and displacement sensor, and a motor was used to provide power. Meanwhile, reasonable installation positions of tools and sensors should be considered to reduce the influence of environmental parameters on the experimental results.

4.2. Data analysis

Through the experiment, the motion state of the impact-scraping drill was analyzed. The working performance of the drill under different rotation speeds was studied, and a corresponding simulation analysis was performed. The experimental and numerical results were compared and analyzed.

The obtained axial impact acceleration and displacement

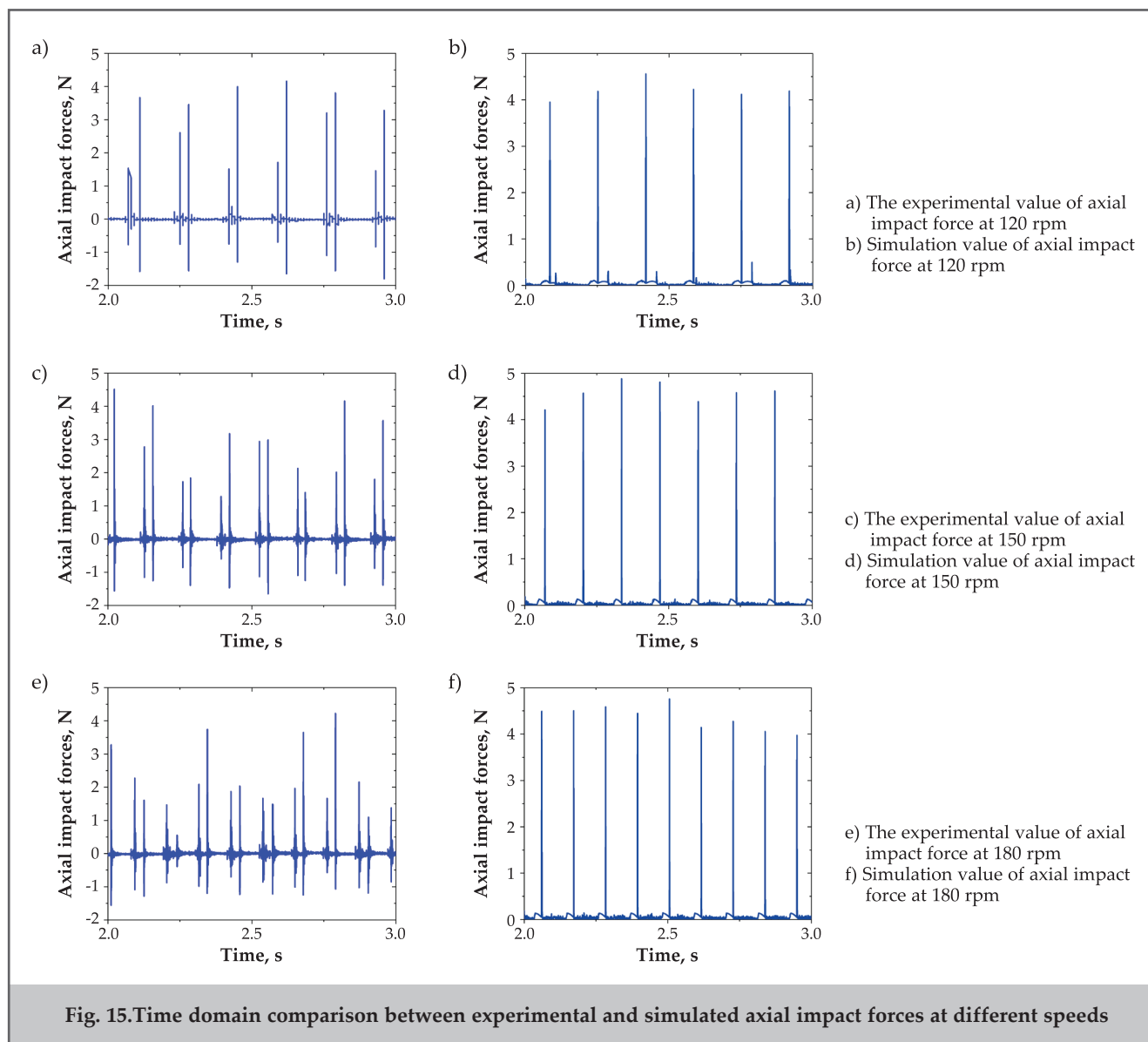
data were processed, and stable 2s data were selected. The experimental data of the axial acceleration were converted according to formula (7), where the m value is 0.4038kg, and the chart of the axial impact load variation over time is shown in figure 15.

$$F=ma \quad (7)$$

where F denotes the axial impact load, N; a denotes the axial acceleration of the test, m/s^2 ; m denotes the mass of the parts, kg.

As can be seen from figure 15, under different rotational speed conditions, the axial impact force presents obvious periodic fluctuations, indicating that the impact-scraping drill tool has a periodic axial impact function. When the input speed is 120, 150 and 180 rpm, the average axial impact force obtained in the bench experiment reaches about 4N-5N, which is slight-

ly smaller than the numerical analysis (5N-6N), but all of them show a decreasing trend with the increase of the rotation speed. The axial impact times tested under different rotation speeds were consistent with the numerical analysis results. The unilateral extreme values of the impact force are 4.16, 4.51 and 4.22 N, respectively, which is about 0.45N different from the numerical analysis results. The impact force amplitudes reached 91.2, 92.4 and 88.7 % of the numerical analysis results, respectively. From the analysis data, the experimental results were in good agreement with the numerical results. The slight difference between the experimental and numerical analysis results is directly related to the installation of the experimental bench and sensor, inaccuracy of the drive speed, and measurement error, which is caused by the difference between the ideal numerical analysis boundary conditions and the actual experimental boundary conditions.



Conclusions

In this study, an impact-scraping drilling tool is proposed, that provides the rotational driving energy of the PDM when working. Through the cam and spring, the hydraulic energy is transformed into the impact force of the drill bit, and the drill bit produces periodic impact to realize the rock breaking effect of the drill tool's impact and rotary cutting action, which are both separated and coordinated.

1. The displacement and speed of the roller at different speeds were analyzed, and the time domain and frequency-domain diagrams of the rotation speeds were obtained. The results show that with an increase in the PDM speed, the effective stroke of the roller decreases, and its speed increases. The contact force and impact frequency between the roller and cam at different speeds were obtained through analysis.
2. With an increase in rotational speed, the maximum contact force and impact frequency increase, and the average contact force decreases. The cam converts the contact force into the impact force between the drill bit and the rock. The results showed that as the speed increased, the impact force gradually increased and the average impact force gradually decreased. The maximum impact force was 19479 N, and the minimum average impact force was 13190 N, both of which were obtained at 180 rpm.
3. The simplified experimental results showed that the changes in the impact force and impact frequency were consistent with the simulation results. With an increase in speed, the maximum impact force increases and the average impact force decreases, which can improve the rock breaking effect of the tool and reduce the failure rate of the bit.

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