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Finite Element Analysis of Static Strength of Tooth-flap Cardan Shaft

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Abstract

The cardan shaft is an important component of the screw drilling tool, which transmits the movement and force of the motor to the drill rod below, and the drill rod is transmitted to the drill bit to achieve the purpose of cutting and breaking the rock. At the same time, the cardan shaft is also a vulnerable part, which seriously affects the efficiency of the entire drilling system, so it is necessary to conduct a static strength analysis on it. Perform 3D modeling based on the drilling data in the field, and then import ANSYS Workbench to set the material parameters, set the contact type, mesh, apply load, obtain the displacement and equivalent stress cloud map, and then verify the grid independence. Solve the maximum torque that the universal joint shaft can withstand under WOB. It is found that there are several stress concentration points in the toothed cardan shaft in the article, and further optimization of the structure is needed; at the same time, the maximum torque that the toothed cardan shaft can bear under a certain drilling pressure is obtained, which is the cardan shaft selection Provide a strong basis. The static strength analysis of the tooth-flap universal joint shaft in the article is helpful to guide the practical application of the project.

Keywords

Tooth-flap cardan shaft; numerical analysis; grid independence; maximum torque.

1. Introduction

Screw drilling tool is a positive displacement downhole power drilling tool that can convert liquid pressure energy into mechanical energy[1]. It is a product introduced by Dyna-Drill CO in the mid-1980s in my country. At present, it has been widely used in the drilling, twisting, coring, reaming and workover operations of directional wells, cluster wells, and vertical wells in petroleum and geological exploration operations in China, thus largely replacing turbodrills. got a good result[2,3,4].

The cardan shaft, as the transmission part in the middle of the screw drilling tool, is an important component second only to the motor. Its function has two aspects: one is to convert the eccentric motion of the motor rotor into the concentric motion of the transmission shaft, and the other is to transmit the motor speed and Torque. Because of its compact structure and large torque transmission, the petal-shaped universal joint shaft has become the most common universal joint shaft used in the field[5,6]. With the increase of oil drilling speed and the development of special craft wells, the problem of screw drilling tool operating conditions and cardan shaft failure is becoming more and more serious. The cardan shaft is the most vulnerable component of drilling tools (according to statistics, cardan shaft failure accounts for about 1/3 of the common failure types of screw drilling tools), which often leads to accidents of drilling tools and unnecessary trips. The drilling operation has caused a great impact, so the working stability of the cardan shaft directly determines the working life of the screw drilling tool[7].

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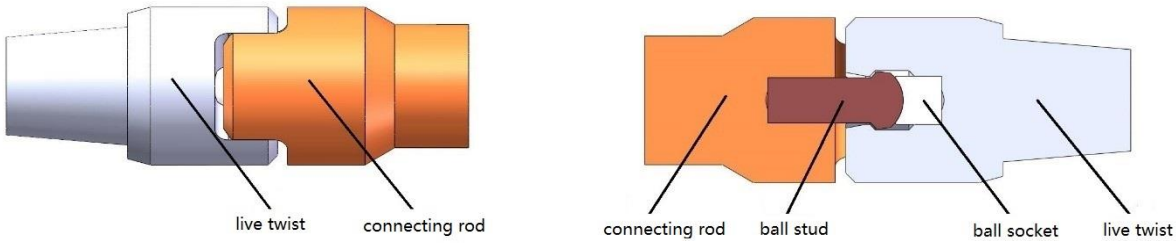
2. Cardan Shaft Composition

The basic structure of the toothed universal joint shaft is shown in Fig 1. One end of the connecting rod part is connected to the screw motor rotor to transmit the output movement and torque of the rotor. The live strand is connected to the drill string through threads, and the lower end of the drill string is connected to the bit part. There are rectangular tooth flaps at the end of the connecting rod and the live twist, and the contact surface of the tooth flap pair should be carburized or nitrided to ensure the service life. The ball seat and the ball head pin are respectively assembled inside the live strand and the connecting rod, and the lower end of the ball head pin is fixed inside the connecting rod by the pin. There is an eccentricity of the drill string at the connection between the screw motor rotor and the live strand. The ball strand pair formed by the ball seat and the ball head pin realizes the conversion of planetary motion into fixed axis rotation, and the torque is transmitted through the tooth flap pair.



(a) 3D model (b) Section view

Fig. 1 The basic structure of the tooth flap universal joint shaft

3. Load Analysis and Its Failure Mode

During rotary drilling, the cardan shaft rotates while being affected by the motor torque, axial force, centrifugal force, rock breaking reaction force of the bit, stick-slip impact and other loads, and the force is complicated. Its motion trajectory is the center of the following end sphere. It is the inverted cone of the vertex[8], as shown in Fig 2. In order to simplify the calculation, the article mainly analyzes the force of the toothed cardan shaft under the axial thrust and torque of the motor rotor[9].

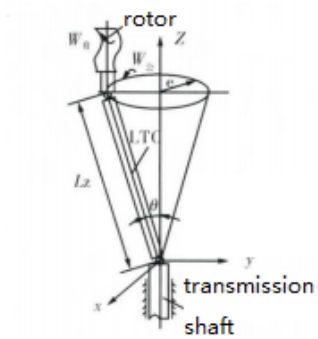


Fig. 2 Schematic diagram of the movement process of the tooth flap universal joint shaft

3.1. Axial Force

$$G = \Delta p e^2[2N(N-1)\pi + 16Nr^0 + \pi(r^0)^2] \quad (1)$$

Where: Δp Indicates the total pressure drop of the screw motor, e represents eccentricity, N represents the number of rotor heads, r^0 represents the equidistant radius coefficient.

The cardan shaft converts the planetary motion of the motor rotor into the rotational motion of the drive shaft and transmits torque. Therefore, the axial force of the cardan shaft can be investigated through the total axial force of the drive shaft, thereby avoiding complicated formula derivation and analysis process. During the drilling process, the axial forces acting on

the drive shaft are: rotor, cardan shaft, drive shaft and bit gravity, rotor axial force, drive shaft piston force, weight on bit. The axial component $Wcosa_B$ of the weight of the rotating parts inside the drilling tool changes according to the inclination angle of the wellbore.

The total axial force of drive shaft P_T:

$$P_T = G + P_D - P_B + Wcosa_B \tag{2}$$

Where: P_D represents the drive shaft piston force, P_B represents weight on bit, W represents the total weight of the rotor, cardan shaft, drive shaft and drill bit, a_B represents the inclination angle of the drilling site.

The piston force of the drive shaft is related to the structure of the drilling tool, the size of the water hole of the drill bit, the displacement of the drilling fluid and the relative density. Calculation formula of drive shaft piston force is:

$$P_D = \frac{\pi D^2 \Delta P_S}{4} \tag{3}$$

3.2. Maximum Contact Stress

Both the living hinge and the living hinge rod are rotating parts. Under the combined action of the axial thrust and torque of the motor rotor, the calculation formula for the maximum contact stress is as follows.

$$\sigma_H = \frac{959}{n_1 n_2}(Pk_d^2)^{1/3} = C_p k^{1/3} \tag{4}$$

$$C_p = \frac{959}{n_1 n_2}[(D_w k_d)^2]^{1/3} \tag{5}$$

$$k = \frac{P}{D_w^2} \tag{6}$$

Where: σ_H indicates the maximum contact stress, n_1, n_2 represents the coefficients related to the curvature of the contact point, k_d indicates the sum of the maximum and minimum curvatures of the rotating body passing through the contact point.

3.3. Main failure mode

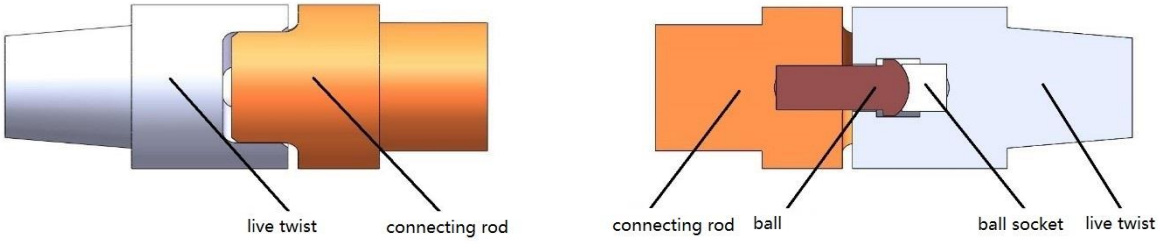
There are three failure modes of tooth flap universal joint shafts[10]: one is the failure of the root of the tooth flap; the second is the failure of the meshing side wear; the third is that the hydraulic piston of the rotor generates a lot of pressure (50~70 kN) , And the ball seat and the ball-end pin are in a state of sliding friction, so the wear is serious or even broken.

4. Cardan Shaft Tooth Static Strength Check

Since the tooth flap universal joint shaft often bears the circumferential or axial load and the combined load of the two during operation, the flap teeth of the tooth flap universal joint shaft are the weak link of the universal joint shaft, which affects the service life of the universal joint shaft. key. The flap tooth is subjected to a combination of pressure and torsion during the rotation process, and the calculation involves geometric nonlinearity, material nonlinearity and state nonlinearity. It is difficult to obtain a reliable solution by theoretical analysis. Therefore, the finite element analysis software Workbench software is used to perform the calculation of the flap tooth Finite element strength check and fatigue life analysis[11].

4.1. 3D model establishment

In order to obtain accurate analysis results, a complete modeling of the toothed cardan shaft must be carried out. Use SolidWorks software to carry out 3D solid modeling. In order to simplify the model during the modeling process, the fixing pins of the ball head pin and the connecting rod and the pin holes on the connecting rod body are omitted. Twisting for division processing, while eliminating unnecessary rounding and chamfering. The model established is shown in Fig 3.



(a) Simplified 3D model (b) Simplified 3D model section

Fig. 3 Simplified model composition structure of tooth flap cardan shaft

4.2. Workbench finite element static strength analysis

Select the statics analysis module in the Workbench software, and the flap tooth material is 40CrNiMoA structural steel. The elastic modulus of the material is $E = 2.09 \times 10^{11}$, the density is 7870kg/m^3, the yield strength is 830MPa, the tensile strength is 950MPa, and the Poisson's ratio is $\mu = 0.295$[12]. In order to reduce the amount of computer calculations, tetrahedral and hexahedral hybrid meshing is adopted, the contact surface size is controlled to 5mm, the surface size of the live twist and connecting rod fillet is controlled to 5mm, the number of nodes is 56834, the number of cells is 37219, and the average grid is orthogonal The quality reaches 0.74138. Set the lower end of the ball stud to be in binding contact with the inside of the connecting rod, the others are in frictional contact, and the friction coefficient is set to 0.1.

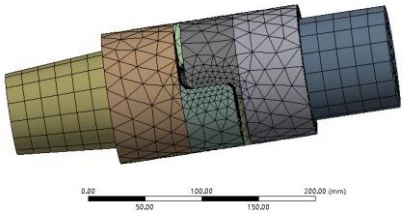
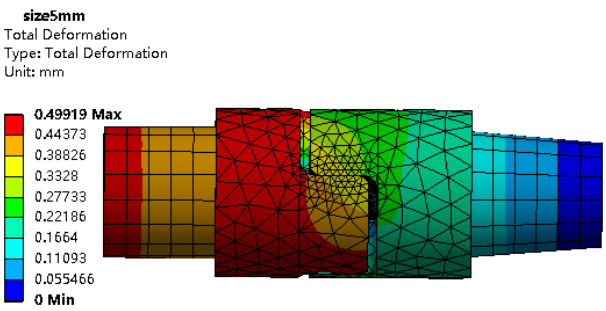
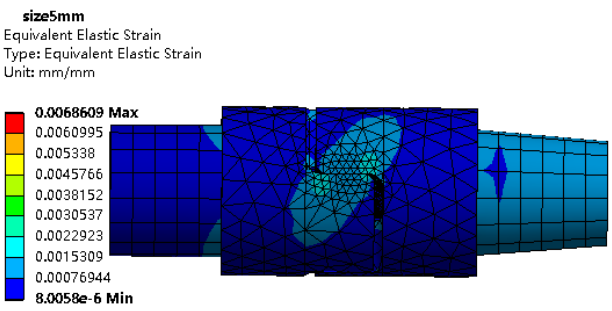


Fig. 4 Finite element model of tooth flap

Fully fix and constrain one end of the live twist to restrict its rotation in the direction of the vertical axis. Apply pressure and torque loads to the ends of the connecting rods. According to theoretical calculations and drilling parameters used in the field, take the torque T=11000kN·m and the axial force F=79.512kN of the cardan shaft, and convert it to the pressure p=12.45MPa when applied to the end face of the connecting rod. The total displacement cloud image and equivalent strain cloud image can be obtained by numerical solution. And the equivalent stress cloud diagram is shown in Fig 5.



(a) The total displacement cloud of the toothed universal joint shaft



(b) Equivalent strain cloud diagram of tooth flap universal joint shaft

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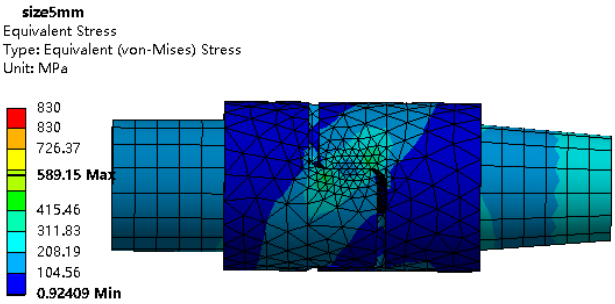


(c) Equivalent stress cloud diagram of tooth flap universal joint shaft

Fig. 5 Finite element analysis results of tooth flap universal joint shaft

4.3. Grid Independence Verification

When there is no problem with model establishment, material property setting, contact setting, and boundary condition setting, the calculation result of the model is largely determined by the quality of the mesh. Therefore, when the global mesh setting is unchanged, the contact network The grid and rounded grid sizes are set to 5mm, 3mm, 2mm, 1.5mm to analyze and compare the equivalent stress to determine the best grid size.

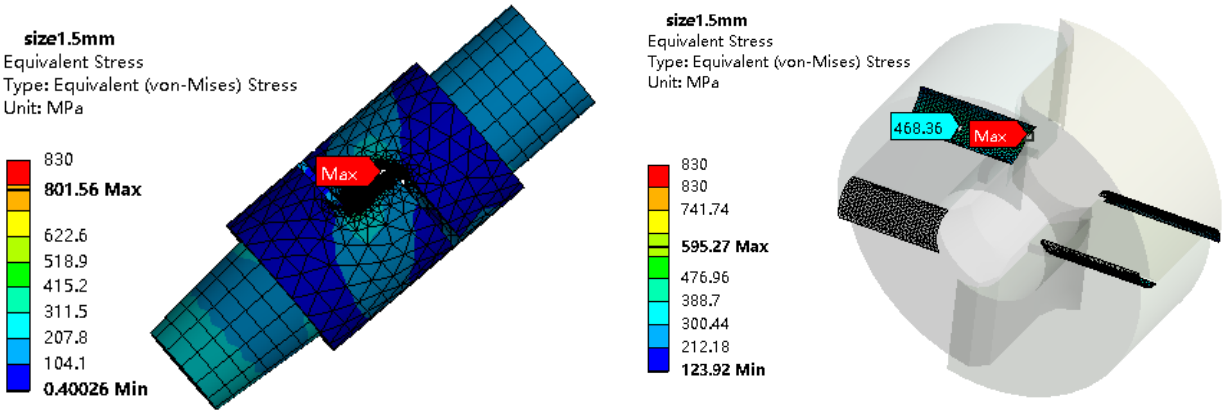
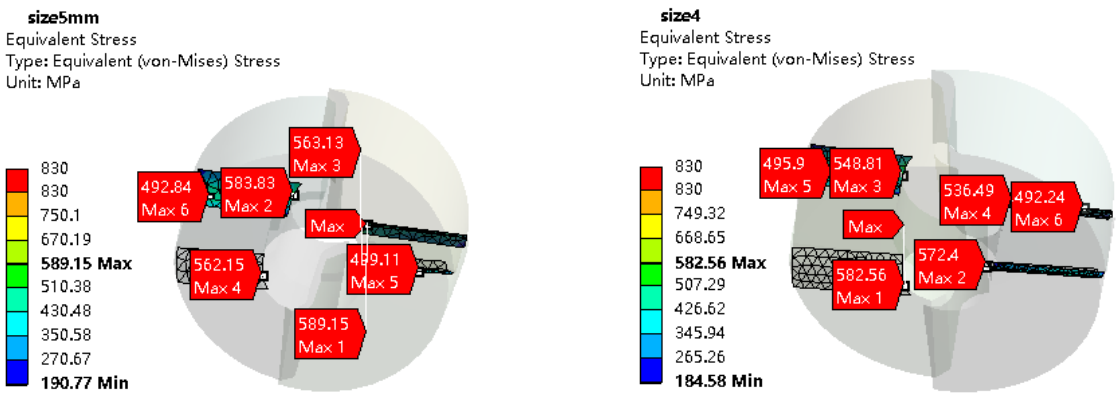


Fig. 6 Stress concentration point

Fig 6 shows the stress concentration part encountered due to structural needs during the analysis process. We do not consider its influence in the grid correlation. In the actual application process, the wear of the contact surface will eventually lead to pitting corrosion and indirectly cause the tooth flap fracture. Failure, so here we mainly discuss the equivalent stress of the fillet surface. From the stress concentration at one end of the rounded surface, the first six points with the largest stress are read in this paper, and the results are shown in Figs 7 and 8.



(a)size5mm (b)size4mm

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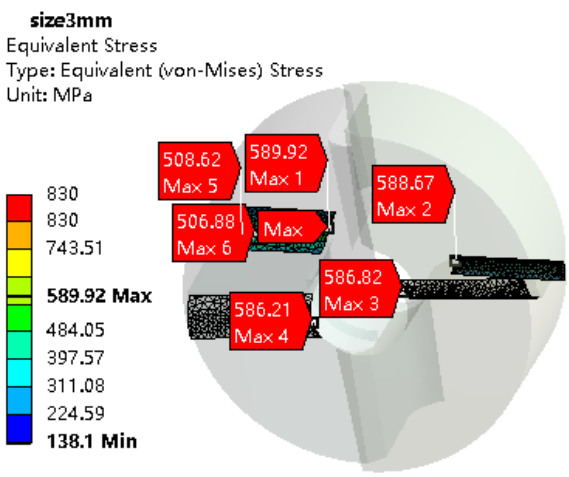
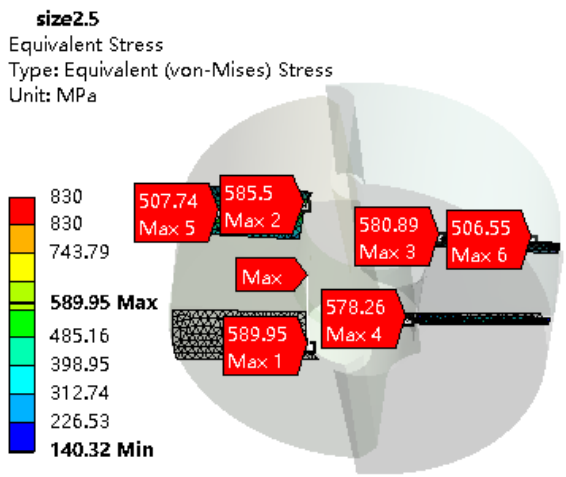
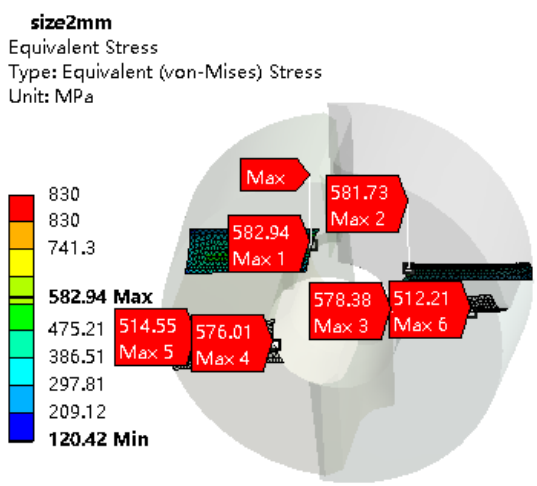


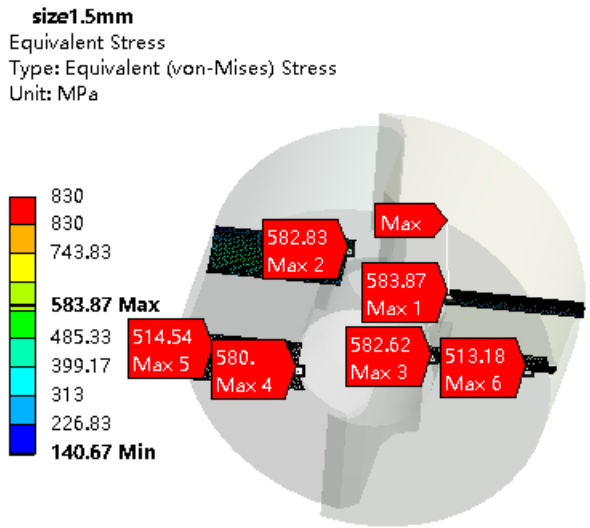
(c)size3mm



(d)size2.5mm



(e)size2mm



(f)size1.5mm

Fig. 7 Grid independence verification

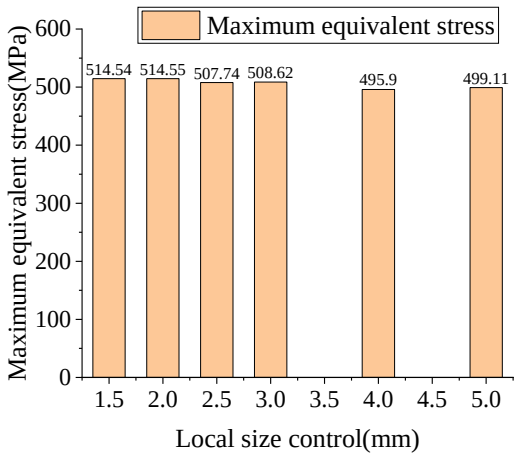


Fig. 8 Maximum equivalent stress of different contact mesh sizes

Due to limited computing resources, the local size can only be controlled to 1.5mm. The six points with the largest equivalent stress in Fig 7 are the stress concentration points at the four fillets and the maximum equivalent stress points on the two fillet planes. Therefore, the normal fillet can be obtained by excluding the point where the stress at the end of the fillet is the

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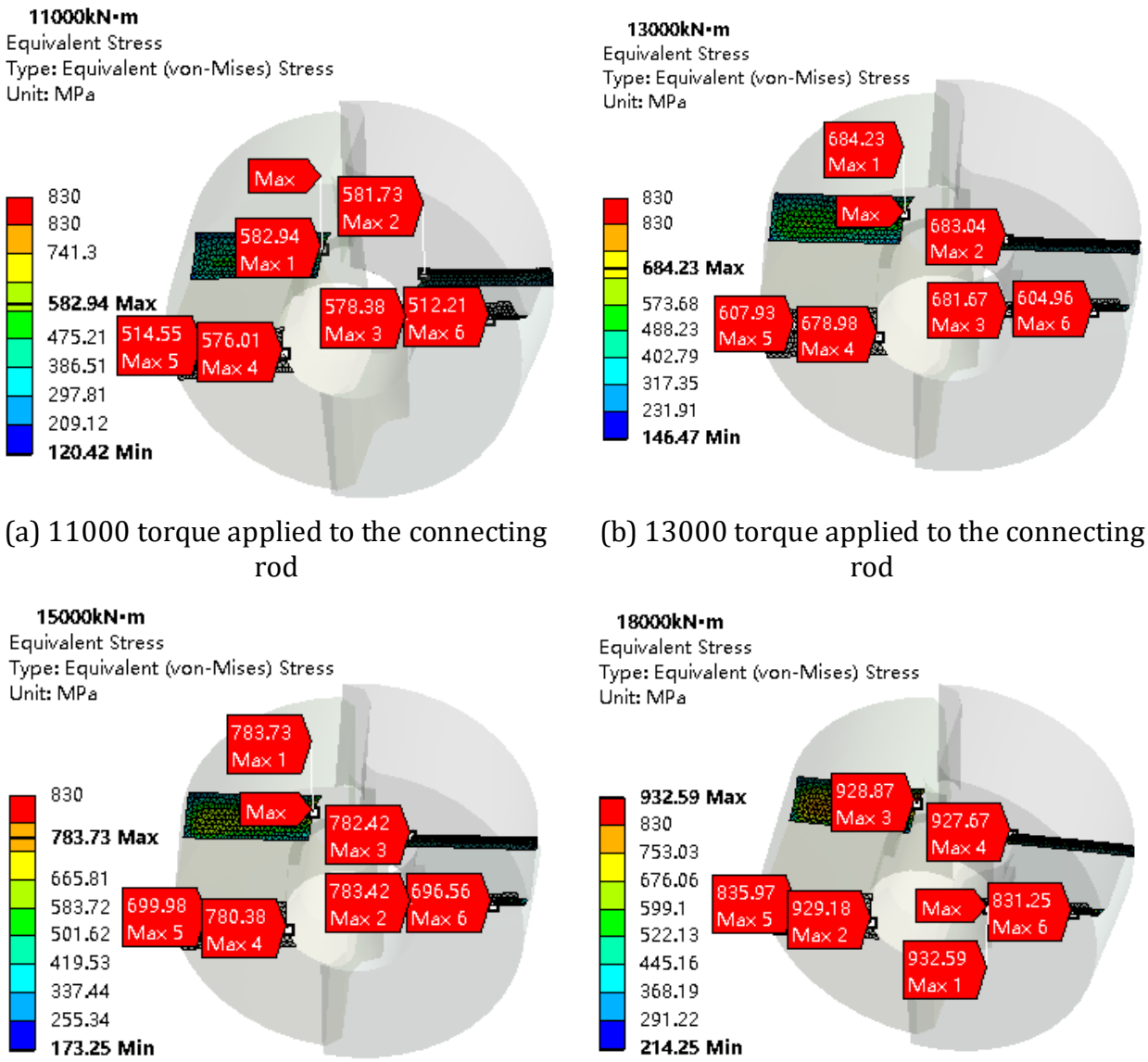
maximum. The point of maximum equivalent stress on the surface. It can be seen from Fig 7 that as the mesh size increases, the equivalent stress of the fillet is increasing as a whole, showing a slowly increasing relationship. From the increasing trend, the local size is controlled at 2mm. At that time, the accuracy of the result was higher, and it was finally decided to use a 2mm grid for the subsequent calculations.

4.4. Maximum axial torque

When the WOB is constant, the torque received by the universal joint shaft determines its performance and life. If it is operated for a long time exceeding the rated torque, the universal joint shaft is extremely vulnerable to damage, so it is necessary to The maximum bearing torque of the joint shaft under a certain weight-on-bit is explored, and the torque is increased to 13000kN·m, 15000kN·m, and 18000kN·m respectively to compare whether the static strength of the universal joint shaft meets the requirements. The results are shown in Fig 9.



(a) 11000 torque applied to the connecting rod

(b) 13000 torque applied to the connecting rod

(c) 15000 torque applied to the connecting rod

(d) 18000 torque applied to the connecting rod

Fig. 9 Comparison of static strength of universal joint shafts with different torques

It can be seen from Fig 9 that the axial pressure remains constant. With the increase of the circumferential torque, the static strength of the cardan shaft is continuously increasing. The influence of the stress concentration point is eliminated. When the torque increases to

18000kN·m, the circle The equivalent stress at the corner reaches 835.97MPa and exceeds the yield limit of the material 830MPa, and the material begins to undergo plastic deformation. In practical engineering applications, the maximum torque needs to be controlled below 18000kN·m.

5. Summary

Through the analysis, the following points are obtained: 1. There are many stress concentration points in the toothed universal joint shaft, which will affect the analysis to a certain extent, and the structure needs to be optimized in the future; 2. The grid independence verification affects the fillet part It is not large, and has a greater impact on the stress concentration part; 3. Under the condition of constant weight on bit, with the increase of torque, the equivalent stress of the tooth flap universal joint also increases. In the text, the weight on the drill is 12.54MPa. The maximum torque of the shaft can reach 18000kN·m, and the applied torque in actual application should be lower than the maximum bearing torque.

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