Analysis of Over-Torque Failure Characteristic of Tool Joints

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Abstract. Increasingly demanding drilling conditions and progressive drilling technology put forward new requirements for the torsion performance of tool joints. Over-torque failure of tool joints in downhole occurs frequently, resulting in huge economic losses. It is of great significance to analyze the mechanism of over-torque failure of the tool joint. This mechanism can help to selection of the tool joint and improve the design of the tool joint. In this paper, three dimensional elastic-plastic finite element models of single-shoulder tool joint and double-shoulder tool joint are established. Based on ABAQUS, the stress characteristics of tool joints under working torque are analyzed, and the mechanism of over-torque failure of API single-shoulder tool joints and double-shoulder tool joints is revealed. The result shows that there is a serious stress concentration phenomenon in both single-shoulder tool joints and double-shoulder tool joints, which restricts the performance of tool joint. The stress distribution of the tool joint should be homogenized in the design of the new type of tool joint, so as to improve the performance of the tool joint.

1. Introduction

With the decrease of shallow oil resources, drilling and production conditions become more and more harsh, drill string failure accidents often occur. As the weak part of the drill string, tool joints often fail. The cost of a well in the western oilfields of China is about tens of millions of yuan, and some even hundreds of millions of yuan. The wellbore scrap caused by the failure of drill string will cause huge losses [1]. The statistical analysis of drill string failure shows that 65% of the failure accidents occur in the tool joints [2]. Once the failure occurs, the drilling tool will need to be replaced, leading to the delay of the drilling cycle. In serious cases, failure will cause the entire well to be scrapped, causing huge economic losses to the oil field.

In deep wells, ultra-deep wells, and extended reach horizontal wells, the drill string is subjected to a large torque. Especially under extreme working conditions, such as when the drill string gets stuck, it is usually used to move the drill tool by lifting and lowering the drill string. The emergence of top drive equipment makes it possible to rotate the drill string while lifting and lowering the drill tools, which is often accompanied by larger axial tension and rotating torque during operation. Therefore, the tool joint is required to have higher torsion performance [3].

Designing tool joint with higher torsion performance is important for reducing drill string failures. Due to the addition of an secondary shoulder, the double-shoulder tool joint has changed the mechanical characteristics of the structure. Compared with the single-shoulder tool joint in API
standard, the double-shoulder tool joint has higher torsional performance and has been widely used. However, due to insufficient understanding of the force mechanism of tool joint, the over-torque failure of tool joint still occurs frequently. By analyzing the over-torque failure mechanism of tool joint, it can help to select the tool joint and improve the design of tool joint.

In this paper, three-dimensional elastic-plastic finite element model of tool joints are established. We use ABAQUS to calculate the stress characteristics of tool joint under the action of torque, and then compare the mechanism of the over-torque failure of the single-shoulder tool joint and the double-shoulder tool joint. The results obtained in calculation are of great significance for improving the design of tool joint.

2. Establishment of three-dimensional finite element model of tool joint

The structure diagrams of single-shoulder tool joint and double-shoulder tool joint are shown in Fig. 1 and Fig. 2. Double-shoulder tool joint has one more shoulder than single-shoulder tool joint. This paper compares the mechanical characteristics of each contact surface of two different types of tool joints, and then the change of torsional performance of tool joints is studied.

![Figure 1. The structure diagrams of single-shoulder tool joint.](image1)

![Figure 2. The structure diagrams of double-shoulder tool joint.](image2)

The finite element model was established with the 4-3/4” tool joint as the research object. The basic dimensions of tool joint are shown in Table 1. In order to simulate the stress characteristics of the tool joint under the action of torque accurately, the spiral angle of the thread should be considered and the three-dimensional finite element analysis should be carried out. The three-dimensional finite element models of single-shoulder tool joint and double-shoulder tool joint are established respectively, the meshing situation is shown in Fig. 3.

| Basic dimensions of tool joint |
|--------------------------------|
| Outer diameter [mm] | 120.65 |
| Inside diameter [mm] | 50.8 |
| Base diameter [mm] | 89.69 |
| Female button boring [mm] | 98.84 |

![Table 1. Basic dimensions of tool joint.](image3)
3. Material properties
Performing a tensile test on the material of the tool joint to obtain the relationship between the nominal stress and the nominal strain of the material. According to formulas (1), (2), (3), nominal stress and nominal strain can be transformed into real stress, real strain and plastic strain [4].

\[ \varepsilon = \ln (1 + \varepsilon_{\text{nom}}) \]  
\[ \sigma = \sigma_{\text{nom}} (1 + \varepsilon_{\text{nom}}) \]  
\[ \varepsilon^\text{pl} = \varepsilon' - \sigma' / E \]

where \( \varepsilon \) is the real strain, \( \sigma \) is the real stress, \( \varepsilon_{\text{nom}} \) is the nominal strain, \( \sigma_{\text{nom}} \) is the nominal stress, \( \varepsilon^\text{pl} \) is the plastic strain, \( \varepsilon' \) is the total real strain, and \( E \) is the Young modulus.

Since the test data contains thousands of data points, the elastic-plastic material model should not contain all the data points. Because the force measured by the test has slight up and down fluctuations, the stress-strain curve appears slightly zigzag. This easily leads to the difficulty of calculation convergence [5]. Select the appropriate number of representative data points in the material test data and let it form a smooth stress-strain curve, as shown in Table 2.

According to the test, the plastic strain of the tool joint material failure is 0.192, which is used as the material failure criterion in the three-dimensional elastic-plastic finite element analysis. The text of your paper should be formatted as follows:

| Real stress [Mpa] | Plastic strain | Real stress [Mpa] | Plastic strain | Real stress [Mpa] | Plastic strain | Real stress [Mpa] | Plastic strain |
|-------------------|----------------|-------------------|----------------|-------------------|----------------|-------------------|----------------|
| 758               | 0              | 920.8             | 0.016          | 1017.6           | 0.045          | 1079.8           | 0.075          |
| 790.1             | 0.0005         | 940.1             | 0.021          | 1030.9           | 0.05           | 1086.1           | 0.08           |
| 818.3             | 0.001          | 953.7             | 0.025          | 1043             | 0.055          | 1091.2           | 0.085          |
| 878.5             | 0.006          | 971.3             | 0.03           | 1054.3           | 0.06           | 1095.1           | 0.097          |
| 891.3             | 0.009          | 987.6             | 0.035          | 1063.6           | 0.065          |                   |                |
| 905.2             | 0.012          | 1003.3            | 0.04           | 1072.4           | 0.07           |                   |                |

4. Three-dimensional Finite Element Analysis of Tool Joint
In order to be closer to the service condition of tool joint, we apply axial tension to the tool joint firstly, and then apply enough working torque to the tool joint until the tool joint fails. Working conditions are shown in Table 3. The loading curve is smooth to avoid numerical oscillation [6].

| Load of each analysis step         | Single-shoulder tool joint | Double-shoulder tool joint |
|-----------------------------------|----------------------------|---------------------------|
| Make-up torque [kN•m]             | 17.33                      | 26.93                     |
| Axial tension[kN]                 | 1500                       | 1500                      |
| Torque[kN•m]                      | 40                         | 50                        |

4.1. Single-shoulder tool joint
Under the working torque, the mechanical characteristics of a single-shoulder tool joint are shown in Fig. 4, Fig. 5 and Fig. 6. The calculation results show that when the working torque exceeds a certain value (Referring to Fig. 6, the torque value is approximately 14 kN•m, which is consistent with the conclusion of Brock, J.N. [7]), the pre-tightening state of the tool joint will be broken. This will lead to downhole make-up of too joints, which increases the failure risk of tool joint. In Fig. 4, the contact area on the thread is sharply increased and the contact pressure on the shoulder surface remains substantially unchanged. This is because the threaded teeth are not yet fully contacted under the make-up torque and the axial tension(Especially the middle thread), and the shoulder surface is already in full contact.In Fig. 5, it can be seen that when the working torque exceeds a certain value, the contact
force on the primary shoulder and the thread increases sharply, and when it rises to a certain extent, it suddenly drops to zero, this means that the joint of single shoulder drill tool has failed. Fig. 6 shows the working torque loading curve. It can be seen that the applied curve starts to fluctuate when the working torque reaches a certain value. This fluctuation reflects the male and the female engaging surface reaching equilibrium under the action of the working torque, then generating the sliding, then balancing again, then sliding again, and finally destroying. At this point, the material has entered the yield state. Since the yield limit under this condition is relatively stable, the ultimate working torque of the single-shoulder tool joint under this working condition is 24.22kN •m.

The distribution of von Mises stress on the single-shoulder tool joint at the failure time is shown in Fig. 7. It can be seen from the figure that a fault with a zero stress appears at the root of the first fully engaged thread near the shoulder at the time of failure. This is because the stress concentration at the root of the first thread is the most obvious [8]. Compared with other regions, the elements in this region are the first to fail. After the failure of the element, the element is removed, and then the stress concentration is transferred to the adjacent element, gradually expanding, and finally the fracture surface as shown in Fig. 7 appears.

The distribution of equivalent plastic strain on single shoulder tool joint at the failure time is shown in Fig. 8. According to the test, the plastic strain of the tool joint material failed to be 0.192. It can be seen from the figure that at the time of failure, the single shoulder tool joint has a relatively obvious equivalent plastic strain in the front threaded area near the shoulder. This area is a dangerous area, and the field practice is also consistent with this [9].
4.2. **Double-shoulder Tool Joint**

Under the working torque, the mechanical characteristics of a double-shoulder tool joint are shown in Fig. 9, Fig 10 and Fig 11. Fig. 9 shows that when the working torque exceeds a certain value (referred to Fig. 11, the torque value is approximately 12.5 kN•m), the secondary shoulder that has been separated under the axial tension is resumed contact. It can be seen from Fig. 10 that the contact forces on the primary shoulder, the secondary shoulder and the thread begin to rise sharply. When the working torque continues to be applied to 42.76 kN•m (refer to Fig. 11), it can be seen from Fig. 10 that the contact force on the secondary shoulder suddenly drops to zero, indicating that the secondary shoulder first failed. It can be seen from Fig. 11 that the working torque loading curve also suddenly drops, and then maintains at a certain level and generates fluctuations, eventually failure occurred.

The distribution of von Mises stress on the double-shoulder tool joint at the failure time is shown in Fig. 12. It can be seen from the figure that the male fails in the two areas: the area near the root of the first engaging thread and secondary shoulder area. According to Fig. 12, it can be seen that the secondary shoulder has failed first, and finally the root of the first thread has gradually formed a fracture surface. The location of the fracture surface coincides with the joint of a single-shoulder tool joint.

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**Figure 8.** Equivalent plastic strain cloud diagram.

**Figure 9.** Contact area

**Figure 10.** Contact force

**Figure 11.** Working torque

**Figure 12.** Von Mises stress cloud diagram.
The distribution of equivalent plastic strain on the double-shoulder tool joint at failure time is shown in Fig. 13. It can be seen from the figure that the double-shoulder tool joint has a relatively obvious equivalent plastic strain in two areas at the time of failure. They are the first few threaded areas near the primary shoulder and secondary shoulder area. These two areas are dangerous areas for double-shoulder tool joint, and it is confirmed by field observation [10].

![Figure 13. Equivalent plastic strain cloud diagram.](image)

5. Conclusions

(1) Three-dimensional nonlinear finite element analysis can help us understand the stress characteristics of tool joint under complex loads and analyze the failure mechanism of tool joint.

(2) The single-shoulder tool joint has a higher stress in the first few threaded areas near the primary shoulder under the working torque. This area is prone to plastic deformation and is a dangerous area where failure is likely to occur. This single-shoulder tool joint has an ultimate working torque of 24.22 kN*m under this condition.

(3) The double-shoulder tool joint has high stress in the first few threaded areas near the primary shoulder and secondary shoulder area under the working torque. These areas are prone to plastic deformation and are dangerous areas where failure is likely to occur. This double-shoulder tool joint has an ultimate working torque of 42.76 kN*m under this condition, which is 76.5% higher than that of the single-shoulder tool joint.

(4) The results show that there are obvious stress concentration phenomena in the joint of single-shoulder tool joint and double-shoulder tool joint, which is the most fundamental reason restricting the performance of the joint. The stress distribution of tool joints should be homogenized to improve the performance of tool joints.

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