A stress amplification model to evaluate the stress concentration of tool joint under complex loads

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Abstract: The more and more severe drilling conditions make the load conditions of tool joint extremely complex. Under the complex load conditions, the stress concentration of tool joint is serious, so failure of tool joint often occur in the oilfield. However, few researchers have studied the stress concentration of tool joint under complex loads. In this paper, the stress distribution of an API tool joint, NC38, was analyzed with 3D elastic-plastic finite element method under complex loads. The stress concentration factors of the tool joint were calculated, and the influence of tensile and bending load on it was analyzed in detail. The results show that the existing model cannot effectively reflect the fact that the stress concentration of tool joint will increase with large tensile load in the crooked hole section. Based on the 3D finite element analysis, a stress amplification model is proposed and used to reflect the failure risk of tool joints under the combined action of axial tension and bending moment. The results show that this model is consistent with the fact that the failure risk of tool joint will increase with large tensile load in the crooked hole section.

1. Introduction
A great number of drill pipes and drill collars are connected by tool joints to form a drill string. Conic threaded connection is the typical and widely used tool joint in the oil and gas industry due to its advantage of well assembly, but it is also the main form of drill string failure. In the upper vertical section of an ultra-deep well or an extended reach well, the drill string has to withstand a huge tensile load, and the failure risk of tool joint increases sharply if large borehole curvature exists.

In order to reduce the failure risk of tool joint, researchers have done many works, with the experimental method, theoretical method and numerical simulation method. Experiment method is intuitive, but it is time-consuming and expensive, and the complex loads are hard to be applied [1-2]. The theoretical method has the advantage of parameterization of the model and understanding the mechanism. However, the highly nonlinear mechanics of thread interactions under complex loads cannot be described due to the stringent assumptions [3-4]. Hence, the finite element method (FEM) has been widely used. Baryshnikov and Baragetti [5] applied a 2D FEM to calculate allowable loads for API tool joints under the make-up torque, torsion and tension. Han, S. et al. [6] analyzed the
fatigue of drill string threaded connections with 2D FEM and fatigue tests, and found that the last engaged thread (LET) subjected to the highest preload and cyclic stress due to stress concentration.

Because the helix angle was neglected in 2D FEM, the make-up torque can not be applied directly. Then more attentions have been paid to 3D FEM recently [7-8]. Shahani et al. [7] analyzed the contact stress distribution on the tool joint by using 3D FEM, and given the location and the value of maximum stress concentration factor (SCF) of the tool joint. Toshimichi Fukuoka et al. [8] applied an analysis with 3D FEM and found that the axial load along engaged threads shows a different distribution pattern from those obtained by 2D finite element analysis and elastic theory. The 3D FEM investigations mentioned above have significant meaning in understanding the mechanical mechanism of the tool joint, but most of the materials used in these analyses are elastic. In fact, severe stress concentration will appear in some local region of tool joint and plastic deformation will occur.

In order to evaluate the mechanical behavior of tool joint, the widely used concept of stress concentration is applied. Tafreshi [9] analyzed the stress characteristics of tool joint under axial, bending and torsion loads with a 2D finite element model, and axisymmetric solid elements with nonlinear, asymmetric deformation with fourier interpolation have been employed in the case of bending. Bahai [10] proposed a 2D parametric model to study the variation of stress concentration factor (SCF) in threaded connections under preloading, axial and bending loads. However, there were no works to analyze the effect of bending moment on the tooth load distribution by the 3D elastic-plastic FEM, and the stress concentration of tool joint under complex loads have never been studied and reported.

In the present study, the mechanical properties of tool joint under various cases of preloading, tensile, bending and combinations of them (40 cases) are studied with the 3D elastic-plastic FEM. The SCF at the root of the tooth of pin are calculated and the influences of external loads on it are analyzed. Because the existing SCF model cannot effectively reflect the fact that the failure risk of tool joint will increase with large tensile load in the crooked hole section. So, in order to reveal the influence of the bending and tensile load on the failure risk of tool joint, a stress amplification model is proposed to describe the stress concentration features of tool joint under complex loads.

2. The bending moment of the tool joint under tension and the model of SCF

Most of the drill pipe failures are related to the severity of the dog leg. This understanding has been presented in the LandMark software, WellPlan, and API RP 7G, i.e. American Petroleum Institute Recommended Practice 7G [11].

The model of the fatigue ratio used in the LandMark software is presented as follows [12]:

$$R_F = \left(\frac{\sigma_{BK} + |\sigma_{BD}|}{\sigma_{FL}}\right)$$  \hspace{1cm} (1)

where, $R_F$ is the fatigue ratio index. When $R_F$ is of a large value, the actual buckling stress and bending stress of the drill string may be exceeded or closed to the fatigue strength limit, and the drill string is in a state prone to fatigue. Conversely, when $R_F$ is small, the stress in drill string at this time is not easy to cause fatigue failure. $\sigma_{BK}$ is the buckling stress (Pa), and is caused by buckling only when the buckling occurs. $\sigma_{FL}$ is the fatigue limit (Pa), and it is obtained by the rule of Goodman [13].

When the drill string is under compression,

$$\sigma_{FL} = \sigma_{FEL}$$  \hspace{1cm} (2)

When the drill string is under tension,

$$\sigma_{FL} = \sigma_{FEL} \left(1 - \frac{F_{AB}}{F_{AY}}\right)$$  \hspace{1cm} (3)

where $\sigma_{FEL}$ is the fatigue endurance limit of the drill string material (Pa). $F_{AB}$ is the axial tensile force of the drill string (N). $F_{AY}$ is the axial force of the drill string corresponding to the yield limit (N). $\sigma_{BD}$ is a bending stress considering the amplification of local bending effect, calculated as follows [12],

$$\sigma_{BD} = \frac{EDCB_M}{2}$$  \hspace{1cm} (4)
where $E$ is the elasticity modulus (Pa). $C$ is a wellbore curvature (rad/m). $D$ is the outer diameter of the drill pipe (m). $B_M$ is the bending stress magnification factor, which is defined as the maximum ratio of the curvature of the pipe body divided by the curvature of the hole axis. This factor can be applied as a multiplier on the bending stress calculations in a work string that has tool joints with bigger outside diameters (OD) than that of the pipe body. $B_M$ is useful because when a drill string with bigger tool joint OD than the body’s OD is subjected to either a tensile or compressive axial load, the maximum curvature of the drill pipe will exceed that of the hole axis curvature (shadow part of Figure. 1). The drill pipe sections conform to the wellbore curvature primarily through contact at the tool joints [12].

As a general practice, the WellPlan has taken the effect of bending into consideration. As shown in Figure. 1, it can be observed that bending stress becomes concentrated close to the tool joint when the pipe is in tension. However, this result only applies to pipe body, but the amplification effect of bending stress on the assessment of tool joint has never been concerned. In fact, the tool joint will also be exposed to serve bending under the action of large axial tension load in the crooked hole section [14].

As a general practice, the SCF can also be used to study the effect of bending moment on the failure of tool joint. SCF is defined as the ratio of the maximum von Mises stress at the root of the tooth, $\sigma_c$, to the nominal stress, $S_n$, which is defined as the stress at the section of critical thread and can be expressed with the loads carried by the body of the pin at the section of the critical thread divided by cross-sectional area at that location [8].

$$SCF = \frac{\sigma_c}{S_n}$$  \hspace{1cm} (5)

3. The 3D stress distribution characteristics and the SCF of the tool joint

3.1 The 3D elastic-plastic finite element model of the tool joint

This paper presents conclusions based on data gathered from full scale finite element numerical simulation analysis of API NC38 tool joint. According to API Spec7 [15], the specifications of the tool joint are listed in Table 1.

| Specifications of the tool joint          | Value       |
|-----------------------------------------|-------------|
| Outside diameter /mm                    | 120.7       |
| Pitch diameter of thread at gauge point /mm | 96.7       |
| Inner diameter /mm                      | 68.3        |
| Minimum length of box threads /mm       | 104.8       |
| Diameter of flat on pin /mm             | 98.8        |
| Thread teeth profile                    | NC38        |
| Length of pin /mm                       | 101.6       |
| Thread form                             | V-0.038R    |
| Depth of box /mm                        | 117.5       |
| Pitch /mm                               | 6.35        |
| Box counterbore /mm                     | 103.6       |
| Taper                                   | 1:6         |
The finite element model of API NC38 is shown in Figure 2. By using the elastic plastic material [16], the mechanic properties of the tool joint are analyzed. The meshes around thread portions are refined to ensure the numerical accuracy, while coarse meshes are used in other regions to ensure the computation efficiency. The finite element model consists of 339,700 nodes and 336,000 elements. The element type chosen for calculation is C3D8I. The friction coefficient of the contact surfaces has been assumed to be 0.08 [11].

To avoid the numerical fluctuation, the loads are applied to the model smoothly, as shown in Figure 3. According to API RP 7G[11], the preload, 14.73 kN\cdot m, is applied. Then axial tension is applied, followed by bending moment, where the bending is applied in critical direction[17].

### 3.2 Stress distribution on the threads

#### 3.2.1 Load cases

In order to reveal the stress characteristics of tool joint, it is necessary to take into account the complex loads, such as make-up torque, axial tension and bending moment. To accomplish this, 40 cases of various loading presented in Table 2 are applied to the model.

| Load case | Load combination | Load case | Load combination |
|-----------|------------------|-----------|------------------|
|           | Make-up torque/kN\cdot m | Tension /kN | Bending /kN\cdot m | Make-up torque/kN\cdot m | Tension /kN | Bending /kN\cdot m |
| 1         | 0                 | 400       | 0                | 21                 | 14.73       | 400               | 7.98             |
| 2         | 0                 | 600       | 0                | 22                 | 14.73       | 600               | 7.98             |
| 3         | 0                 | 800       | 0                | 23                 | 14.73       | 800               | 7.98             |
| 4         | 0                 | 1000      | 0                | 24                 | 14.73       | 1000              | 7.98             |
| 5         | 0                 | 1200      | 0                | 25                 | 14.73       | 1200              | 7.98             |
| 6         | 14.73             | 400       | 0                | 26                 | 14.73       | 400               | 11.40            |
| 7         | 14.73             | 600       | 0                | 27                 | 14.73       | 600               | 11.40            |
| 8         | 14.73             | 800       | 0                | 28                 | 14.73       | 800               | 11.40            |
| 9         | 14.73             | 1000      | 0                | 29                 | 14.73       | 1000              | 11.40            |
| 10        | 14.73             | 1200      | 0                | 30                 | 14.73       | 1200              | 11.40            |
| 11        | 14.73             | 400       | 3.42             | 31                 | 14.73       | 400               | 13.68            |
| 12        | 14.73             | 600       | 3.42             | 32                 | 14.73       | 600               | 13.68            |
| 13        | 14.73             | 800       | 3.42             | 33                 | 14.73       | 800               | 13.68            |
| 14        | 14.73             | 1000      | 3.42             | 34                 | 14.73       | 1000              | 13.68            |
| 15        | 14.73             | 1200      | 3.42             | 35                 | 14.73       | 1200              | 13.68            |
| 16        | 14.73             | 400       | 5.70             | 36                 | 14.73       | 400               | 17.10            |
| 17        | 14.73             | 600       | 5.70             | 37                 | 14.73       | 600               | 17.10            |
| 18        | 14.73             | 800       | 5.70             | 38                 | 14.73       | 800               | 17.10            |
| 19        | 14.73             | 1000      | 5.70             | 39                 | 14.73       | 1000              | 17.10            |
| 20        | 14.73             | 1200      | 5.70             | 40                 | 14.73       | 1200              | 17.10            |
3.2.2 3D stress distribution on the threads subjected to preload. Tightening of the tool joint is important for the reliability of the drill string. Interference in connections due to tightening can generate contact pressure to ensure the seal integrity and prevent down hole breakout or loosening. Figure 4 indicates the distribution of axial force on the teeth of an NC38 tool joint under the action of preload. It is observed that the first pair of threads sustained the main part of the load, and the results are consistent with those of Shahani [8].

![Figure 4. Load distribution due to preloading](image)

3.2.3 3D stress distribution on the threads subjected to combination of preload and axial load. Figure 5 presents the tooth load of tool joint under different axial tensile loads in the presence of preloading. It can be found that the teeth near to the shoulder have higher load bearing contribution than the other teeth, while the load on each tooth increases along with the increase of axial tension.

![Figure 5. Tooth load under case6,7,8,9,10](image)

![Figure 6. Tooth load under case7,12,17,22,27,32,37](image)

| Table 3. Percentage of load distribution on the threads |
|--------------------------------------------------------|
| Load case | 1 | 2 | 3 | 4 | 5 | 6 | 7 | 8 | 9 | 10 | 11 |
|-----------|---|---|---|---|---|---|---|---|---|----|----|
| 6         | 24.0 | 14.2 | 10.2 | 8.6 | 7.9 | 7.1 | 6.5 | 5.7 | 5.3 | 5.0 | 5.6 |
| 7         | 23.3 | 14.0 | 10.1 | 8.6 | 7.8 | 7.1 | 6.6 | 5.8 | 5.4 | 5.3 | 6.0 |
| 8         | 22.7 | 13.9 | 10.0 | 8.5 | 7.8 | 7.1 | 6.6 | 6.0 | 5.6 | 5.5 | 6.3 |
| 9         | 22.3 | 13.8 | 9.8  | 8.4 | 7.7 | 7.1 | 6.7 | 6.1 | 5.7 | 5.7 | 6.7 |
| 10        | 21.8 | 13.5 | 9.7  | 8.3 | 7.7 | 7.1 | 6.7 | 6.2 | 5.9 | 6.0 | 7.1 |

From Table 3, we can observe that the stress concentration of the first three teeth near the shoulder is serious, especially the first teeth bear much more than the other teeth. In the presence of axial tension load, the load portion of the first three pairs is about 45.0% to 48.8% of the total load while the bearing ratio of the three tooth pairs from the pin end is just about 15.8% to 19.0%.

3.2.4 3D stress distribution on the threads subjected to bending moment. The contact moment distribution on each thread of the tool joint at different hole curvature under an axial tension of 600kN is shown in Figure 6, and the detailed data are listed in Table 4. The results show that the load is mainly borne by the engaged teeth at both ends, whether the bending moment is loaded or not. The first pair of teeth undertook the maximum load, which was more than 20% of the total load. It also can
be observed that the action of bending moment will further aggravate the stress concentration of the tool joint, and the tooth load increases with hole curvature.

It is known that under the action of tensile load, a greater dog leg will lead to a larger bending moment. In the actual drilling process, “dog leg” and its induced bending moment is inevitable. Therefore, it is necessary to take the bending moment into consideration in the process of exploring the mechanical properties of the tool joint under complex loads.

### Table 4. Percentage of load distribution on the threads/ (%)

| Load case | Bending /kN•m | 1 | 2 | 3 | 4 | 5 | 6 | 7 | 8 | 9 | 10 | 11 |
|-----------|----------------|---|---|---|---|---|---|---|---|---|----|----|
| 12        | 3.42           | 28.7|17.6| 9.5| 7.1| 4.9| 4.0| 1.9| 3.8| 2.7| 2.1 | 17.7|
| 17        | 5.70           | 27.6|17.4| 9.3| 6.8| 4.9| 4.6| 2.2| 4.2| 2.4| 3.4 | 17.3|
| 22        | 7.98           | 26.2|16.6| 9.0| 6.6| 4.9| 5.0| 2.7| 4.4| 3.3| 4.6 | 16.7|
| 27        | 11.40          | 24.3|15.6| 8.6| 6.3| 5.0| 5.3| 3.4| 5.1| 3.9| 6.2 | 16.3|
| 32        | 13.68          | 23.2|14.9| 8.3| 6.1| 5.0| 5.4| 3.8| 5.3| 4.9| 6.9 | 16.1|
| 37        | 17.10          | 21.8|14.2| 8.1| 6.0| 5.1| 5.6| 4.3| 5.8| 5.3| 7.9 | 16.0|

3.3 The stress concentration factors of the tool joint

As shown in Equation (5), SCF is defined as the ratio of the maximum von Mises stress at the root of the tooth to the nominal stress at the critical cross section of the tool joint. For a specific tool joint and load conditions, a relative accurate SCF can be obtained from the results of 3D FEM model.

The stress distribution of tool joint and the corresponding SCF at the root of the threads of the pin are displayed in Figure. 7 and Figure. 8. We can observe that the maximum value of SCF is located at the root of the first tooth of the pin and is much higher than that of other teeth. This can clearly demonstrate that the bending load resulting from the dog leg and large tension load is the main reason of the tool joint failure at the root of the critical tooth, i.e. the first tooth of the pin. Besides, it is not hard to see that the values of SCF increase with the borehole curvature.

The SCFs of the critical tooth under different axial tensions and different borehole curvatures are shown in Figure. 9. It is found that the SCF on the critical tooth of the tool joint increases with the increase of borehole curvature and decreases with the increase of axial tension.
4. Discussion and stress amplification factor

It is worth emphasizing that the SCF becomes smaller with the increase of axial tension, which is not consistent with the fact that a large tensile load will observably increase the failure risk of the tool joint. This shows that the empirical Equation (5) has some limitations. In fact, SCF is promoted to reflect the stress concentration of the tool joint under a given axial tension, but cannot show the fact that the tool joint with high failure risk when a large axial tension is applied. With the increase of the axial tensile load, the nominal stress $S_n$ of the tool joint body has a faster growth than the maximum local stress $\sigma_c$, and then the SCF of the tool joint decreases gradually. In order to reflect the effect of tension load on failure risk adequately, a stress amplification factor model is proposed here,

$$F_{cr} = \frac{\sigma_c}{S_1}$$

where $F_{cr}$ is the stress amplification factor. $\sigma_c$ is the maximum von Mises stress at the root of the critical tooth (MPa). $S_1$ is the maximum stress value of the tool joint under make-up torque recommended by the API RP 7G. For the S135 drill pipe, $S_1$ has the value of 496.44 MPa.

According to Equation (8), the values of stress amplification factor on the critical tooth of the tool joint can be calculated based on 3D FEM and the results are shown in Figure. 10. We can see that the stress amplification factor gradually increases as the tension load and the borehole curvature increase, and can be used to reflect the fact that the tool joint will have a high risk in a curved wellbore when the axial tension is of a large value. That is to say, the stress amplification factor can more intuitively reflect the stress concentration features of the tool joint under complex loads.

The results described in Figure. 11 can be expressed as follow:

$$F_{cr} = 0.0002F + 0.021k + 1.1979$$

where, $F$ is the axial tension load (KN); $k$ is the borehole curvature (°/30m).

It can be seen from Equation (7) that the stress amplification factor of the tool joint has a linear positive correlation with the borehole curvature and the axial tensile load. So, this model of stress amplification factor consistent the fact that the failure risk of tool joint will increase with the increase of tensile load in the crooked hole section. In practical drilling, the combined effects of axial tension and borehole curvature, especially in the upper borehole section of an ultra-deep well, must be taken into consideration to decrease the failure risk of the tool joint.

5. Conclusions

Under complex load, the stress concentration seriously affects the durability of tool joint. In this paper, a stress amplification model is proposed to describe the stress concentration characteristics of tool joints under complex loads, and the effects of bending and tensile loads on the failure risk of tool joints are revealed. The following conclusions are drawn from this study:
1) The critical tooth (the first thread of pin for NC38) of tool joint bears the largest load, and the stress concentration factor increases significantly with the action of bending moment.

2) The widely used SCF approach can reflect the fact that the SCF on critical tooth increases gradually with the bending moment at a given axial tensile load, but cannot describe the fact that a large failure risk will exist under the large tensile loads in the condition of a given bending moment.

3) Based on the 3D finite element analysis, the stress amplification factor is proposed to reflect the failure risk of tool joints under combined action of axial tension and bending moment.

4) The results of the stress amplification factor is consistent with the fact that the failure risk of tool joint will increase with the increase of tensile load in the crooked hole section.

5) The stress amplification factor obtained in this paper is only applicable to nc38 joints. For other types of joints, stress amplification factors can be calculated according to the method in this paper.

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