Design and Analysis of Iron Roughneck Non-Clamp-Tooth Clamping Mechanism

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Abstract—During the makeup and breakout operation of iron roughneck, the performance of the clamp tooth holding the pipe string directly affects the drilling efficiency and the service life of the pipe string. In this paper, a cylindrical non-clamp-tooth Clamping body with 20CrMnTi material is designed to replace the upper shackle clamp tooth. In the up-shackle operation of the torque wrench, the clamping body contacts the outer wall of the string in the form of two parallel axis cylinders, the relationship between each influencing factor and contact stress is established, and the contact stress and deformation of the string are simulated by finite element method. This non-clamp-tooth clamping body can reduce the damage degree of the string, make its elastic deformation not exceed 1mm, and improve the service life of the clamping body.

1. Introduction
The running and pulling operation of the pipe string accounts for the most time of the drilling process, foreign oil equipment manufacturers used iron roughneck as makeup and breakout mechanism on the pipe string as early as the late 20th century, and used iron roughneck in rig equipment[1], it is also equipped with independent screwing mechanism and the punching mechanism to complete the makeup and breakout operation of the pipe string, which is widely used in a variety of drilling platforms and different drilling environments on land and sea[2].

Torque wrench is the iron roughneck actuator with the maximum torque and load. This paper mainly designs the caliper body part of the torque wrench. In the up-shackle operation of the torque wrench, the clamping body contacts the outer wall of the string in the form of two parallel axis cylinders, the purpose is to reduce the damage degree of the clamp tooth to the outer surface of the string and improve the service life of the clamping body.

2. Design of non-clamp-tooth clamping mechanism
The upper and lower caliper of the iron roughneck is driven by a single hydraulic cylinder, the hydraulic cylinder is placed at the tail of the upper and lower caliper, and it is connected to the caliper body through the thrust rod to achieve the clamping action of the pipe string[3]; when makeup and breakout mechanism of the torque wrench is working, the upper caliper adopts one-way torsion mode of single hydraulic cylinder, and the torsion hydraulic cylinder is placed on the tail of the upper caliper, after the lower caliper clamps the pipe string, the upper caliper is pushed to rotate relative to the lower
caliper. The structure of the iron roughneck non-clamp-tooth clamping mechanism designed by the research team of the author is shown in Figure 1\[4\].

![Diagram of the iron roughneck non-clamp-tooth clamping mechanism]

**Figure 1. The structure of the iron roughneck non-clamp-tooth clamping mechanism**

Through the research on the structure of the clamp tooth, the main design direction should be to reduce the damage degree of the clamp tooth to the pipe string and extend its service life. As can be seen from figure 2, the designed non-clamp-tooth clamping mechanism mainly consists of cylindrical clamping body. The springs at the upper and lower ends of the clamping body are kept in contact with the base by fastening bolts.

3. **contact analysis of non-clamp-tooth clamping mechanism and pipe string**

3.1. **Properties of Clamping Body and Pipe String Material**

The clamping body is the main vulnerable part during the makeup and breakout operation of the torque wrench. Under the premise of effectively protecting the pipe string from serious damage, the replacement frequency should be reduced as much as possible. The clamping body is selected to be manufactured by 20CrMnTi and its steel grade is G105\[5\]. The material properties of the clamping body and the string are listed in Table 1.

| Material | Yield/MPa | Tensile Strength/MPa | Elasticity Modulus/GPa | Poisson's Ratio |
|----------|-----------|----------------------|------------------------|-----------------|
| 20CrMnTi | 835       | 1080                 | 212                    | 0.289           |
| G105     | 724       | 793                  | 210                    | 0.28            |
| E75      | 516       | 690                  | 208                    | 0.28            |
| X95      | 655       | 724                  | 208                    | 0.28            |
| S135     | 931       | 1000                 | 211                    | 0.28            |

3.2. **Contact Stress**

At the contact line at the beginning of the contact between the clamping body and the pipe string, the compressive stress suffered by both of them is the largest, that is, the contact stress $\sigma_{H}$ after the force exerted on the clamping body and the string. For the linear contact between two cylinders, the formula of contact stress given in elastic mechanics is $^{(6)}$:

$$\sigma_{H} = \frac{F}{\pi L \left( \frac{1}{\rho_1} + \frac{1}{\rho_2} \right)} \sqrt{\frac{1-\mu_1^2}{E_1} + \frac{1-\mu_2^2}{E_2}}$$

F- total pressure acting on the contact surface, N;
L-the length of the initial contact line, m;
ρ1, ρ2-radius of curvature at the initial contact line between the clamping body and the pipe string, m;
μ1, μ2-poisson's ratio of the clamping body and pipe string material;
E1, E2-elastic modulus of the clamping body and pipe string material, Pa.

3.3. Sensitivity Analysis of Contact Parameters Between the Clamping Body and Pipe String

After analyzing the contact between cylindrical clamping body and pipe string, it can be known from formula (1) that after selecting the pipe string and the clamping body material, the factors affecting the contact stress between the clamping body and the outer wall of the pipe string joint are the A-clamping body radius ρ1, B-the outer radius of the pipe string ρ2, C-the initial contact line length L and D-the total pressure F acting on the contact surface, as shown in table 2, the best values of four factors can be selected by orthogonal experiment.

The optional orthogonal table corresponding to 4 factors and 4 levels is L16(4⁵), that is, 16 experiments are carried out, these four factors are respectively labeled as A, B, C, D, and there's a vacant column, labeled as E. In order to calculate the stress when the clamping body contacts the pipe string, design the orthogonal test, the orthogonal table and contact stress calculation results are shown in table 3. The results of mean stress and standard deviation are shown in Table 4.

| Table 2. Factor levels of contact stress |
|-----------------------------------------|
| levels | A-radius of the clamping body ρ1 (mm) | B-outer radius of the pipe string ρ2 (mm) | C-contact line length L (mm) | D-total pressure at the contact surface F (kN) |
|--------|---------------------------------------|----------------------------------------|-----------------------------|----------------------------------|
| 1      | 10                                    | 60.3/2                                 | 100                         | 50/2                             |
| 2      | 25                                    | 88.9/2                                 | 150                         | 75/2                             |
| 3      | 40                                    | 127/2                                  | 200                         | 100/2                            |
| 4      | 55                                    | 168.3/2                                | 255                         | 120/2                            |

| Table 3. orthogonal experimental table of contact stress |
|---------------------------------------------------------|
| test number | A | B | C | D | E (vacant column) | test results (MPa) |
|-------------|---|---|---|---|-------------------|--------------------|
| 1           | 1 | 1 | 1 | 1 | 1                 | 551                |
| 2           | 1 | 2 | 2 | 2 | 2                 | 335                |
| 3           | 1 | 3 | 3 | 3 | 3                 | 1028               |
| 4           | 1 | 4 | 4 | 4 | 4                 | 981                |
| 5           | 2 | 1 | 2 | 3 | 4                 | 944                |
| 6           | 2 | 2 | 1 | 4 | 3                 | 1171               |
| 7           | 2 | 3 | 4 | 1 | 2                 | 447                |
| 8           | 2 | 4 | 3 | 2 | 1                 | 596                |
| 9           | 3 | 1 | 3 | 4 | 2                 | 799                |
| 10          | 3 | 2 | 4 | 3 | 1                 | 583                |
| 11          | 3 | 3 | 1 | 2 | 4                 | 747                |
| 12          | 3 | 4 | 2 | 1 | 3                 | 474                |
| 13          | 4 | 1 | 4 | 2 | 3                 | 869                |
| 14          | 4 | 2 | 3 | 1 | 4                 | 431                |
| 15          | 4 | 3 | 2 | 4 | 1                 | 704                |
| 16          | 4 | 4 | 1 | 3 | 2                 | 728                |
Table 4. calculation table of mean and range of contact stress

| factors | Mean 1 (MPa) | Mean 2 (MPa) | Mean 3 (MPa) | Mean 4 (MPa) | range |
|---------|--------------|--------------|--------------|--------------|-------|
| A       | 723.8        | 789.5        | 650.8        | 683.0        | 7.73  |
| B       | 790.8        | 630.0        | 731.5        | 695.8        | 8.2   |
| C       | 799.3        | 614.3        | 613.5        | 720          | 9.5   |
| D       | 433.5        | 636.8        | 820.8        | 913.8        | 14.57 |

Figure 2 shows the broken line diagram of the mean contact stress of factors A, B, C and D under the four levels of orthogonal test. It can be seen from Figure 2 that the range of factor D is the largest, indicating that the total pressure F acting on the contact surface has the most significant influence on the contact stress, followed by factor C, B and A, which have the least influence on the contact stress.

When factors A and B are fixed, the three-dimensional diagram of the change of contact stress with factors C and D is shown in Figure 3. It is instructive to select the appropriate length of clamping body when the pressure is constant.

When factors C and D are fixed, the three-dimensional diagram of the change of contact stress with factors A and B is shown in Figure 4. It is instructive to select the appropriate length of clamping body when the pressure is constant. It is instructive to select the appropriate clamping body radius when the drill pipe size is fixed.
Figure 4. Three-dimensional diagram of the variation of contact stress with factors A and B

When factors B and C are fixed, the three-dimensional diagram of the change of contact stress with factors A and D is shown in Figure 5. It is instructive to select a reasonable pressure when the clamping body radius is fixed.

Figure 5. Three-dimensional diagram of the variation of contact stress with factors A and D

It can be seen from the contact stress value obtained from the combination of all factors in Table 3 that the selected drill pipe steel grade is G105, and the results of experiment number 1, 2, 7, 8, 10, 12, 14 and 15 meet the strength requirements. The clamping body radius $\rho_1$, the outer radius of the pipe string $\rho_2$, and the contact line length $L$ were selected from the same group, calculate and analyze the change of safety coefficient of drill pipe of steel grade G105, E75, X95 and S135 with the increase of pressure, as shown in Figure 6: the difference of safety coefficient of drill pipe of different steel grade, this is conducive to select the appropriate strength of drill pipe under specific working conditions, as well as to save costs and improve the utilization rate of drill pipe.

Figure 6. Safety factor of drill pipe of different steel grade
4. Finite element simulation of the contact between the clamping body and the pipe string

When calculating the deformation amount of the clamping body and the string, the axial stretch or compression of the string caused by the torsion process can be ignored, and think of one end of the string as a fixed constraint. In the process of clamping the string with torque wrench, the clamping body is only restricted by contact with the string.

The clamping body has a force of 120kN when clamping the 5ʺ pipe string, the diameter of the clamping body is 110 mm, and the length is 150 mm, it is consistent with the 15th set of data in the orthogonal experiment, and the maximum stress is found in the qualified group. Both ends of the pipe string are fixed with constraints and the clamping body is fixed with axial constraints due to the constraints of the upper and lower shell of the torque wrench. The clamping body is distributed symmetrically on both sides of the pipe string, and take 400mm specimen length for deformation and stress analysis. The contact model between the clamping body and the pipe string was meshed with 4-node quadrilateral elements. After several trial calculation, the mesh size of the clamping body was determined to be 7mm, the mesh size of drill pipe was determined to be 7mm, the drill pipe contact position was set as a small rectangle, and the mesh size was set as 0.5mm, at this time, the number of elements and nodes was 103,282 and 177,048.

Figure 7. Total deformation nephogram when clamping 5ʺ pipe string

Figure 8. Total stress nephogram when clamping 5ʺ pipe string
Figure 9. deformation nephogram when clamping 5" pipe string

Figure 10. stress nephogram when clamping 5" pipe string

It can be seen from Figure 7- Figure 10 that the deformation and stress of the clamping body and the pipe string under the action of the total pressure are mainly concentrated near the contact line. The maximum deformation of the string is about 0.12mm, and the maximum stress on the clamping body is 868MPa, which is close to the results of the 15th group of orthogonal experiments. At this point, the drill pipe with the highest steel grade S135 should be selected, with a yield strength of 931MPa. By using the same method, the other three pipe strings with diameters are analyzed, and it is found that the total deformation of the pipe string and the clamping body under the action of force is about 0.07mm at most, and the maximum stress is 631MPa, so the G105 drill pipe should be selected.

5. conclusion

1) In this paper, a new non-clamp-tooth clamping mechanism is designed. The clamping body contacts the outer wall of the pipe string in the up-shackle operation on the torque wrench in the form of two parallel axis cylinders. This non-clamp-tooth clamping mechanism can reduce the damage degree of the pipe string and make its elastic deformation less than 1mm, which can meet the engineering needs.

2) The factors that affect the contact stress of the clamping body and the outer wall of the pipe string joint include the clamping body radius, the outer radius of the pipe string, the the initial contact line length and the total pressure acting on the contact surface. In this paper, the relevant data in the design are given and analyzed to prove that the parameter selection is reasonable.

3) The finite element method was used to simulate the contact stress and deformation of the clamping body when clamping the pipe string, and the consistency of the two stress calculation methods was verified.

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