Study on strength and sealing of a bolted flange joint under complex working conditions

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Abstract. Bolted flange joints are widely used in petrochemical equipment and pipelines, while the problem of its leakage is a serious threat to the environment and safety. In order to study the strength and sealing of the bolted flange connection, a three-dimensional finite element model of the bolted flange joint with a flexible graphite spiral wound gasket is developed by using APDL in ANSYS. Considering the nonlinear compression rebound constitutive relation of gasket material, the indirect coupling method of temperature and structure is used to simulate the whole process of the bolted flange joint from installation to maintenance. The stresses of the flange, bolt and gasket under different operated conditions are investigated in detail. The results show that the stress distribution of the bolted flange joint is not uniform along the circumference due to the tightening sequence of bolts. The stress of each part of the bolted flange joint changes slightly under pressure condition, but the stress distribution is greatly affected by high temperature and creep at high temperature for 1000 hours. And the possibility of leakage of the bolted flange connection is increased with the application of transient cooling and heating up and additional bending moment and torque. After the application of various loads in turn, the bolted flange joint is likely to leak on the tension side.

1. Introduction

Bolted flange joint is a kind of detachable sealing structure, which is widely used in petrochemical vessels and pipelines because of its convenient installation and disassembly. Its working conditions are complex, often affected by the preload, medium pressure, temperature and the additional bending moment and torque caused by the additional displacement of the pipe caused by the settlement of the bracket, which will have different degrees of impact on the strength and sealing performance of the bolted flange joint, and will cause sealing failure in severe cases. The main failure form of bolted flange joints is leakage, which will cause environmental pollution and even serious safety accidents, so
it is of great significance to study the strength and sealing of bolted flange joint under various complex working conditions.

Many scholars have done a lot of research on the strength and sealing of flange joint under single load and some combined working conditions at home and abroad. Ju H and Linb Z [1] used three-dimensional nonlinear finite element modeling and experimental method to study the influence of elastic interaction in the process of tightening bolted flange joint, and evaluated the dispersion of bolt pre tightening force in the process of tightening considering the nonlinear non-elastic characteristics of gasket. Aklinechache [2] considered the analytical solution of the creep relaxation of bolt flange and gasket, focused on the stress relaxation caused by the creep of flange and bolt, and established a three-dimensional finite element model with ANSYS to verify it. The research shows that the analysis results of bolt stress relaxation and gasket contact stress change are consistent with the finite element results. Sayed A. Nassar [3] studied the influence of the joint action of elastic interaction and creep relaxation between bolts on the integrity and sealing of bolted flange joint. The results showed that the influence of elastic interaction and creep on bolted flange joint was related to the gasket material used.

However, the failure of bolted flange joint is often the result of long-term accumulation of various working conditions. And it is difficult to understand the change of stress and displacement of flange, gasket and bolt under the joint action of various working conditions for a long time. Therefore, based on the previous research, this paper makes a deeper study, considering from the pre tightening condition of bolt tightening sequence when flange connection is installed, to the pressure and temperature rise conditions when it is put into operation, to the creep condition of stress relaxation of flange and bolt material caused by long-term high temperature, and finally to the stop and start-up condition after troubleshooting the equipment. With the help of APDL in ANSYS, the strength and tightness of flange connection under complex working conditions are studied, and the whole process of flange connection from installation to operation and maintenance is simulated, which provides theoretical basis for the practical application of flange connection design, installation and maintenance.

2. Finite element model

2.1. Structural and material parameters

In this paper, the oil and gas inlet flange of an aromatics reforming unit is taken as the research object, and the bolted flange connection model is established according to HG/T20623-2009 and HG/T20631-2009 standards. In order to consider the influence of bolt tightening sequence and additional load on bolted flange joints, APDL in ANSYS is used to model the whole model of bolted flange joint. In addition, considering that there is a certain gap between the bolt and the flange, and the air in the gap has a large thermal resistance when the flange joint heats up, the air layer model of bolt hole is established in the thermal analysis to consider the influence of air on its temperature field, and the air layer model is deleted in the structural analysis. The bolted flange joint model is shown in Fig.1. The flange material is 2.25Cr1Mo, the bolt and nut materials are 25Cr2MoVA and 35CrMo respectively, and the gasket is flexible graphite spiral wound gasket.
2.2. Load and boundary conditions of thermal analysis

In the thermal analysis, the temperature of the inner surface of the flange is 497°C, the heat transfer coefficient of the outer surface of the bolt flange joint contacting with the air is 32W/m²°C, and the heat transfer coefficient of the outer surface of the middle part of the bolt contacting with the air is 20W/m²°C. The flow and heat transfer coefficient of the outer surface of the upper and lower flange which is not in contact with the gasket and the outer surface of the gasket are 10W/m²°C, and the ambient air temperature is 25°C.

2.3. Loads and boundary condition for structural analysis

In the preloading condition, 280KN is taken in the finite element calculation of preload of single bolt based on waters formula. According to ASME PCC-1-2010, a single tool loading method was selected, 28 bolts are numbered clockwise. The pressure working condition is to apply the internal pressure of the medium after the bolt is pre tightened, in which 1.35MPa is applied to the inner wall of the flange, and 14.85MPa is applied to the free end face of the upper flange. In the heating condition, the indirect coupling method of thermal structure is adopted, and the temperature of the element nodes in the temperature field is read and applied to the finite element model as the body load. The implicit creep method is used to simulate the creep of flange and bolt material for 1000 hours. The creep equation of bolt material at 450°C is shown in formula (1), and the creep equation of flange material at 565°C is shown in formula (2) [4], where $\dot{\varepsilon}$ is creep strain rate and $\sigma$ is stress.

$$\dot{\varepsilon} = 9.86 \times 10^{-18} \sigma^{2.99} \quad (1)$$

$$\dot{\varepsilon} = 1.733 \times 10^{-26} \sigma^{9.732} \quad (2)$$

In the condition of stop and start-up, the overall temperature of flange connection reduces to the ambient temperature, and then increase to the working temperature. On this basis, use MPC184 element to establish a rigid beam at the end of the upper flange and apply the bending moment $4 \times 10^8$ N·mm and torque $6 \times 10^5$ N·mm. In the whole process of structural analysis, the bottom of the lower flange is restrained by fixed displacement.

3. Results analysis and evaluation

3.1. Assessment of stress intensity

Due to the consideration of the sequence of tightening bolts, the stress distribution of bolted flange joint is uneven along the circumference. The variation rules of the maximum

![Figure 1. Finite element model of a bolted flange joint.](image)
stress of flange and bolt and the minimum stress of gasket under complex working conditions are shown in Fig.2.

3.1.1. **Evaluation of the flange strength.** After all bolts are tightened, the maximum stress of the flange is 265.065MPa, which occurs at the neck of the lower flange corresponding to the No.27 bolt. The internal pressure and the membrane stress at the pipe end reduce the stress of flange, and the maximum stress is reduced to 226.5MPa, which is located at the corner of the bolt hole of the lower flange corresponding to the No.28 bolt. It is the discontinuity of the structure and the sequence of tightening bolts that produce local high stress. The maximum stress of flange under temperature load increased to 654MPa. Because the expansion of bolt is greater than the expansion of flange under high temperature, the stress at the discontinuity of the outer structure of the lower flange hole corresponding to the No.1 bolt is the largest.

After creep of flange and bolt for 1000 hours, the stress of bolted flange joint is relaxed, and the maximum stress of flange drops sharply to 161.37MPa. At this time, the maximum stress occurs on the left and right sides of the bolt hole of the lower flange corresponding to the No.28 bolt. The temperature stress of the bolted flange joint is generated by transient cooling to room temperature and then rising to the working temperature, and the maximum stress on flange is increased to 311.7MPa. After the additional bending moment and torque are applied, the stress of the upper flange increases obviously, and the maximum stress is 419.3MPa on the tensile side of the top flange as shown in Fig.3.

![Figure 2. Maximum stress of flange and bolt and minimum stress of gasket.](image1)

![Figure 3. Stress distribution of flange under complex working conditions.](image2)

According to JB 4732-1995 [5], the third strength theory is used to evaluate the stress of flange. When \( P_m \leq S_m \), \( P_l + P_b \leq 1.5S_m \) and \( P_l + P_b + Q \leq 3S_m \) are satisfied at the same time, the strength of flange can be considered safe. Under various complex working conditions, the stress linearization curve along the flange thickness direction is shown in Table 1. The evaluation results show that the upper and lower flanges can still meet the requirements of strength under various conditions.

**Table 1. Classification and evaluation of the stress of flange.**

| Project       | Upper flange | Lower flange | Allowable stress | Results of assessment  |
|---------------|--------------|--------------|------------------|------------------------|
| \( P_m \)    | 29.39        | 27.99        | \( S_m=117 \)    | Meet the requirement   |
| \( P_l + P_b \) | 87.69        | 105.2        | \( 1.5S_m=175.5 \) | Meet the requirement   |
| \( P_l + P_b + Q \) | 85.36        | 107.4        | \( 3S_m=351 \)   | Meet the requirement   |
3.1.2. Evaluation of bolts strength. In the process of working condition accumulation, the maximum stress of bolt reaches the maximum value after loading internal pressure. The stress distribution of bolts is uneven because of the bending deformation of bolts caused by the deflection of the flange. In general, the stress of bolts gradually decreases from the inside to the outside in the radial direction. Along the axial direction, from the middle to both sides, the stress decreases first, then increases and then decreases. The stress concentration at the bolted flange joint of nut and bolt reaches the maximum value due to the structural discontinuity. As shown in Fig.4, after the additional bending moment and torque, the stress on the surface of the No.28 bolt contacting with the lower flange is the largest. Since the maximum stress of bolts under all working conditions occurs on No.28 bolt, the strength of No.28 bolt is evaluated, and the stress linearization curve drawn along its middle cross-section path is shown in Fig.5. The basic allowable stress intensity of bolts at high temperature is 196Mpa. According to Fig.5, \( P_m = 140.5 \text{MPa} \leq S_m \), \( P_1 + P_2 = 210.8 \text{MPa} \leq 1.5S_m \), \( P_1 + P_2 + Q = 196.1 \text{MPa} \leq 3S_m \), so the bolts meet the strength requirements of various combinations of working conditions.

![Figure 4. Stress distribution of bolt under complex working condition.](image)

![Figure 5. Stress linearization curve along the cross-section of No.28 bolt.](image)

3.2. Sealing evaluation

3.2.1. Stress of the gasket. The stress distribution of the gasket under complex working conditions is shown in Fig.9, which increases gradually from the inside to the outside along the radial direction and fluctuates slightly along the circumferential direction. The external stress reaches the maximum at the corresponding position between No.27 and No.28 bolts, and the minimum value of the internal stress appears at the corresponding position with No.24 bolt. It can be seen from Fig.2 that the minimum stress of the gasket decreases gradually with the accumulation of working conditions, and the minimum stress of the gasket decreases to the minimum value of 33.7MPa after the action of additional bending moment and torque. The sealing performance of the gasket is determined by the minimum stress on the contact surface. It can be seen from Fig.6 that the stress on the inner side of the tension side of the gasket is the smallest after continuous accumulation of various working conditions, so the possibility of leakage at this location is the greatest. However, since the minimum stress of the gasket is still much larger than the minimum gasket sealing specific pressure, no leakage has yet occurred.

3.2.2. The deflection angle of the flange. Under different combined working conditions, the flanges deflect to different degrees, which results in uneven compression of gasket. The 2010 ASME VIII-1 stipulated that the deflection angle of the integral flange does not exceed 0.3°. In this paper, the flange deflection angles of the four positions of the flange circumferential
direction of 0°, 90°, 180° and 270° under different working conditions are obtained by dividing the difference between the axial displacement of the flange's inner and outer edges by the difference of the flange's inner and outer radius. The results are shown in Fig. 7. The maximum deflection angle of the flange reaches the maximum under pressure, which is 0.1221° and less than 0.3°. So the bolted flange joint meets the sealing requirements in terms of the deflection angle of the flange.

Figure 6. The stress distribution of the gasket under complex working conditions.  
Figure 7. The deflection angle of the flange in circumferential direction.

4. Summary

In this paper, the effects of complex operated conditions on the strength and sealing of the bolted flange joint are studied based on a three-dimensional finite element model. The investigation considered the nonlinear compression rebound constitutive relation of the gasket material and the simulation is completed by the software of the APDL in ANSYS. Some conclusions are obtained as below.

The stresses are larger at the hole and neck of the flange corresponding to No.1, No.27 and No.28 bolt, and the stress on the inside of No.28 bolt is always the largest. Therefore, more attention should be paid to these places during the inspection and maintenance of the bolted flange joint. The stress of the gasket decreases with the continuous application of loads. Furthermore, the additional bending moment and torque have a great influence on the stress distribution of the gasket, which increases the possibility of leakage on the tension side of the gasket. The non-uniformity of the stress distribution of the gasket along the radial direction is aggravated at larger deflection angle of the flange. And this phenomenon will lead to the decrease of the effective sealing width and even the collapse of the outer side of the gasket. These conclusions can provide a theoretical reference for the practical application of bolted flange joints.

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