Pre-tightening Force Analysis of Different Bolt Models Based on Finite Element Simulation

Rongwang Chai [1, a], Zhilin Yu [2, 3, b], Hui Xu [4, c], Xu Huang [2, 3, d],
Kun Mao [5, e], and Wen Zhou [2, 3, f]

[1] Kaihua County Natural Gas Co. LTD, Quzhou, China;

[2] Quzhou Special Equipment Inspection and Testing Research Institute, Quzhou, China;

[3] Quzhou Key Laboratory of New Energy Equipment Safety and Intelligence Research, Quzhou, China;

[4] Hangzhou Special Equipment Inspection Science Research Institute, Hangzhou, China;

[5] Faculty of Information Engineering, Quzhou College of Technology, Quzhou, China.

[a] 306895961@qq.com, [b] 327011861@qq.com, [c] qztj_xh@163.com, [d] 412198501@qq.com,
[e] mk1201@126.com, [f] dester19831219@126.com

Abstract. This article adopts three common bolt modeling methods among scholars and uses Workbench to simulate and analyze the three models under the same preload force, obtaining different simulation results. The advantages and disadvantages of the three modeling methods were analyzed by comparing the simulation results with the theoretical calculation results. It was proposed that for components that may fail due to fatigue at the root of the thread, refined modeling in the fatigue life calculation process is necessary, which provides a reference for theoretical simulators.

Keywords: Bolt; Preload; Finite element analysis.

1. Introduction

Bolt connection is a common mechanical connection method widely used in equipment such as flange installation, fan blade fixation, engines, and wind turbines. During the assembly process of a threaded connection, bolts are subjected to axial tension pre-tightening force by applying external torque. Pre-tightening force can improve the structural load-bearing capacity, improve the internal stress distribution of components under stress, enhance the reliability, anti-loosening ability, and fatigue strength of bolt connections, strengthen the tightness and rigidity of connections, and prevent sliding of bolt connections under transverse loads[1]. If the pre-tightening force is insufficient, it is easy to cause premature failure of the equipment, loosening of bolts, and leakage of the flange. Therefore, research on bolt pre-tightening force and reliability has become a hot topic of concern for scholars [2-6]. In the pre-tightening force simulation analysis, some scholars adopt simplified modeling methods for bolts [7-8], while others adopt refined modeling methods [9-11].

However, relevant scholars have not yet researched the impact of different modeling methods on the pre-tightening force and reliability analysis of bolts. This study establishes a theoretical calculation and simulation comparison in response to the above issues. It analyzes the differences in bolt preload force under different models based on three different bolt modeling and calculation methods, providing a reference for theoretical simulators.

2. Finite element model of bolted connections

2.1 Solid Model.

In this study, we established a three-dimensional model based on the actual size of the bolt, simplified and refined models of the M12×1.75 bolt and nut, and a bolt pre-tightening sensor model, as shown in Figure 1-3. The pre-tensioning sensor is made of carbon steel, and the bolts, nuts, and washers are made of 304 austenitic stainless steel, which is the default material library in the workbench.

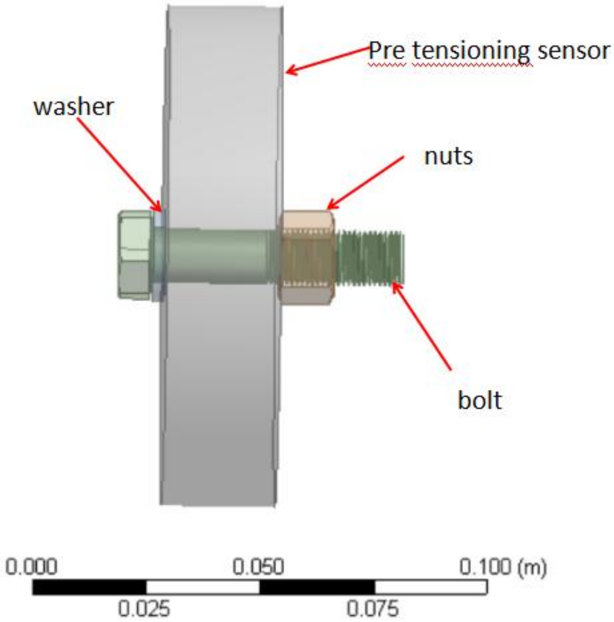


Fig. 1 Model of bolted connections

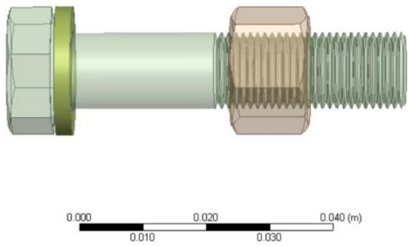


Fig. 2 Refined bolt model

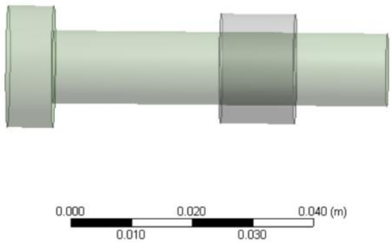


Fig. 3 Simplified bolt model

2.2 Grid meshing.

In this simulation, the mesh of the pre-tensioned sensor and washer by sweep method uses quadrilateral mesh for bolts and nuts. The mesh size of the pre-tightened sensor is 2mm. In order to simulate the contact between bolts and nuts, the mesh size of bolts, nuts, and washers is refined to 0.5mm, with 229319 elements and 924733 nodes. The mesh is shown in Figure 4.

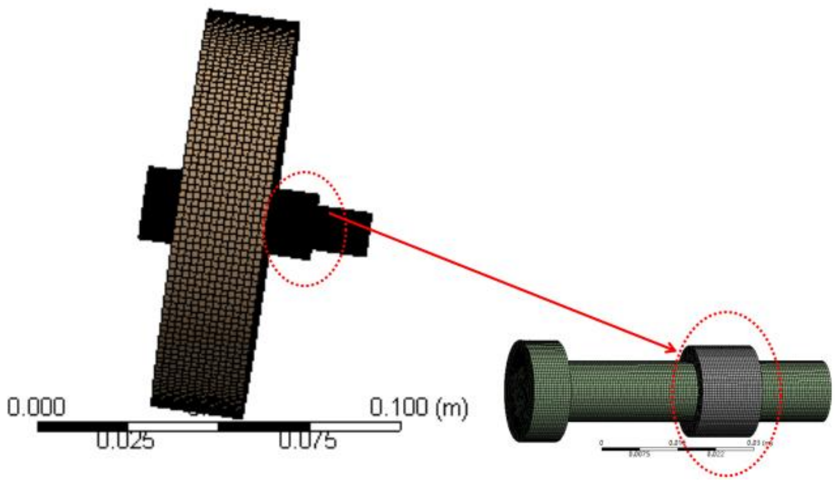


Fig. 4 Grid division diagram of simplified bolt model structure.

2.3 Setting.

This study is divided into three models. The first model is a simplified bolt model, which uses bonded contact between the bolt and nut. The second model is also a simplified bolt model, but uses no separation contact between the bolt and nut, where the bolt contact geometry correction is input and the thread parameters of the bolt are modified in the contact geometry modification. The third model uses the GB/T5782 bolts and corresponding nuts from the Solidworks toolbox to create a refined model. The remaining contact settings for the three models are shown in Table 1, with a friction coefficient of 0.15.

Calculation of bolt preload force is a crucial aspect of this study, particularly in its practical application to real-world engineering scenarios. Under operating conditions, the load on the bolt is generated by the preload stress, the weight of the component, and external forces. In practical engineering applications, the preload stress of the bolt is calculated using the following equation 1 [12]..

$$F_k \le k\sigma_s A_{eff} = 25474.3\text{N} \tag{1}$$

Where k is the coefficient of yield strength, 0.6-0.7 for alloy steel bolts, and 0.5-0.6 for carbon steel bolts, σ_s is the yield strength of bolt material, and A_{eff} means an effective cross-sectional area of bolts.

Where

$$A_{eff} = \pi \frac{d_e^2}{4} \tag{2}$$

Herein, d_e is the effective diameter of the bolt. This simulation analysis adopts a pre-tension force of 24000N, aligning with the engineering situation. Its setting is shown in Figure 5.

Table 1. Contact setting of three different models.

Model parts	Bolt and washer	Washer and pre tensioned sensor	Nuts and pre tensioned sensor	Nuts and bolt
Model1	Frictional contact	Frictional contact	Frictional contact	Bonded contact
Model2	Frictional contact	Frictional contact	Frictional contact	No separation contact
Model3	Frictional contact	Frictional contact	Frictional contact	Bonded contact

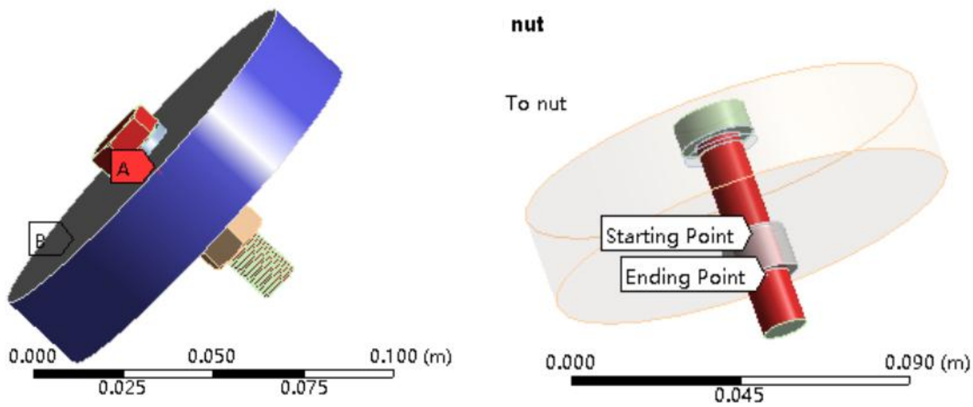


Fig. 5 The Setting of different models.

3. Theoretical analysis

The tensile strength of a bolt refers to the dangerous cross-section of the bolt, mainly the minimum cross-section of the bolt, usually the strength of the bare stem of the bolt. When pre-tightening bolt connections are assembled, the nut needs to be tightened. Under the action of tightening torque, the bolt is subjected to tensile stress caused by the pre-tightening force and shear stress caused by the friction torque rotation on the thread, resulting in the bolt being subjected to a combination of tensile and rotational stress. Therefore, when conducting a strength analysis of pre-tightened bolts, both tensile stress and shear stress should be considered [13]..

The tensile stress of the bolt under the action of a pre-tightening force is shown as follows:

$$\sigma = \frac{4F_k}{\pi d_e^2} \tag{3}$$

The bolt shear stress under pre-tightening force is shown as follows:

$$\tau = \frac{4F_k}{\pi d_e^2}\tan(\phi + \rho_v)\frac{2d_1}{d_e} \tag{4}$$

Where

$$\sigma_v = \arctan \mu_v \tag{5}$$

Where σ is tensile stress, F_k is pre-tightening force, d_e is the effective diameter of the bolt, ϕ is thread angle rise, σ_v is the equivalent friction angle of the thread, and μ_v is the equivalent friction factor of the thread.

According to reference [14], it is known that

$$\tau \approx 0.5\sigma \tag{6}$$

According to the von-Mises equivalent stress calculation formula, it can be concluded as follows:

$$\sigma_e = \sqrt{\sigma_1^2 + \sigma_2^2 - \sigma_1\sigma_2 + 3\tau^2} \approx 1.5\sigma \tag{7}$$

Finally, the stress of the von-Mises screw under bolt pre-tightening is calculated as 212.3 MPa, and the maximum stress at the thread is calculated as 318.4 MPa.

4. Result analysis of finite element

Under the action of pre-tightening force, the von Mises stress distribution of bolts with standard threads is shown in Figure 6, with a maximum stress value of 374.9 MPa, occurring at the connection between the nut and screw. In addition, the stress concentration of the thread is mainly concentrated at the root of the first few threads, as well as at the incomplete thread where the first thread is screwed in. The contact stress of the thread teeth gradually decreases with the increase in the number of threads, which is consistent with the distribution characteristics of the thread pair load. This is consistent with the previous research results in reference [10].

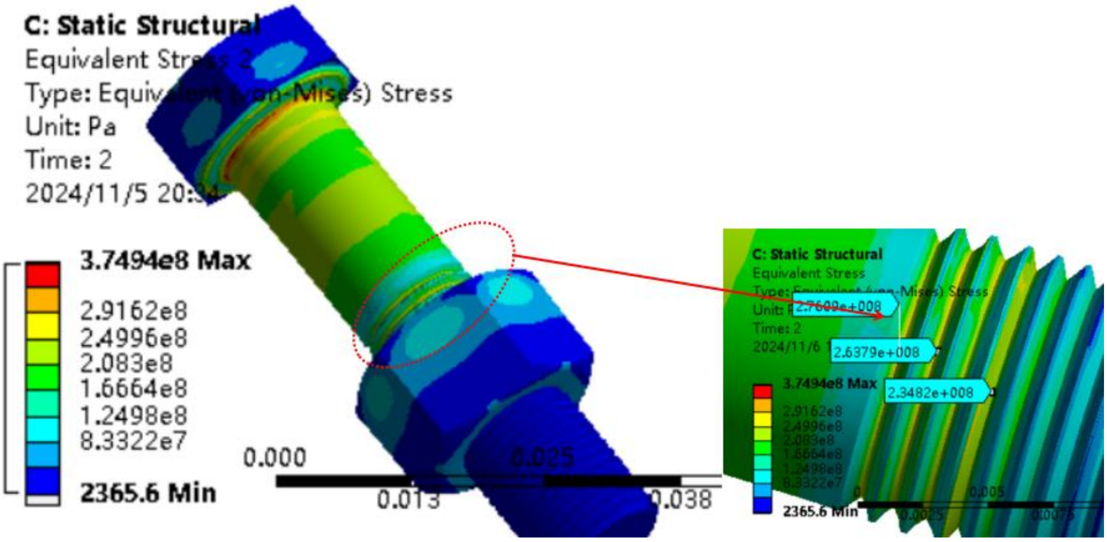
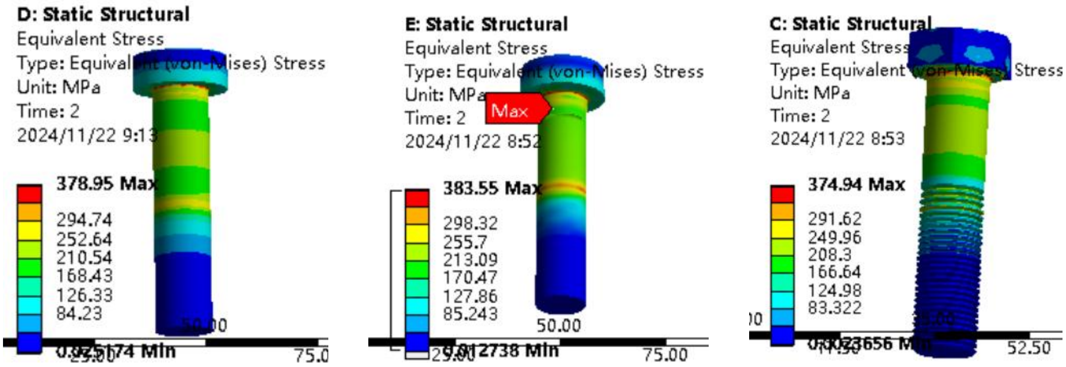


Fig.6 Von-Mises stress distribution diagram of bolts under pre tightening force.

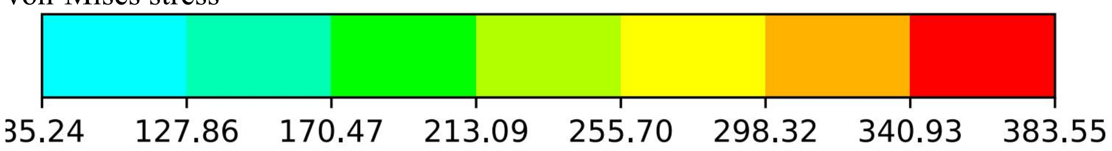
In order to compare the stress conditions of the three models under the same preload force, it can be seen from Figure 7 that the von-Mises stress at the screw position for the three modeling methods are 211.86MPa, 217.18MPa, and 212.85MPa. The maximum error from the theoretical value 212.3 is only 2.3%, indicating that the numerical simulation method is consistent with the theoretical solution and the model settings meet the requirements.



model 1 model 2 model 3

Fig. 7 Von-Mises stress distribution cloud map of three bolt models under preload force.

This study set up I, II, III, IV, V, and VI six surface monitoring devices at positions 0, 2.6, 25, 25.5, 27.6, and 39.8mm away from the nut, representing the connection between the nut and bolt, the position of the washer, the first thread position, the first thread root, the first thread position of the nut and bolt connection, and the last thread position of the nut and bolt connection. Record the stress and strain of different models at different cross-sections, as shown in Figure 8.



number	cross section I	cross sectionII	cross sectionIII	cross sectionIV	cross sectionV	cross sectionVI

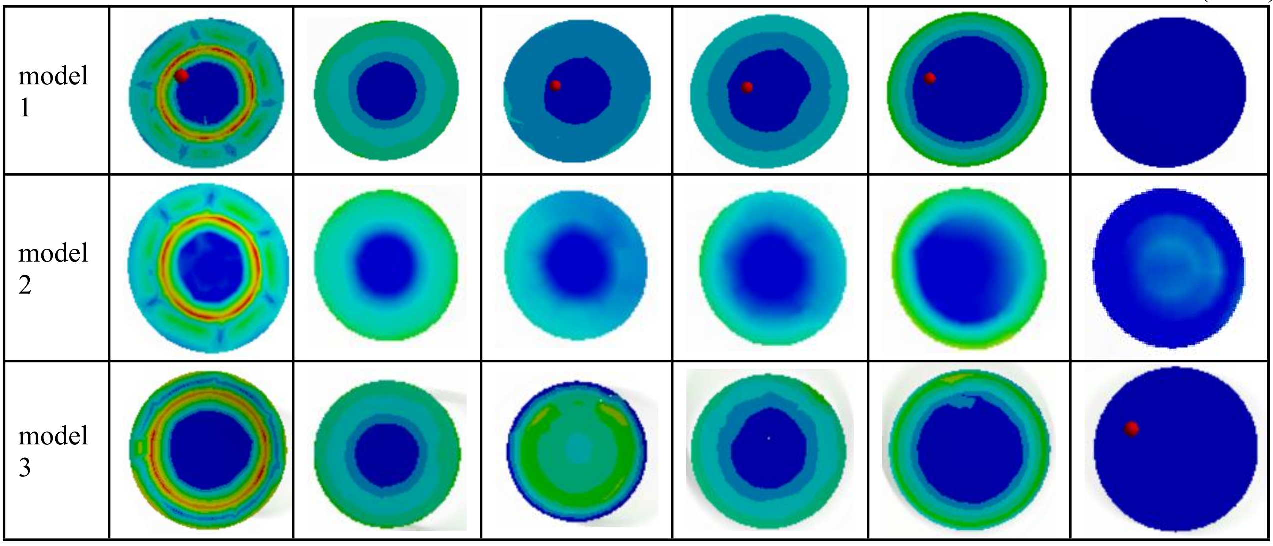
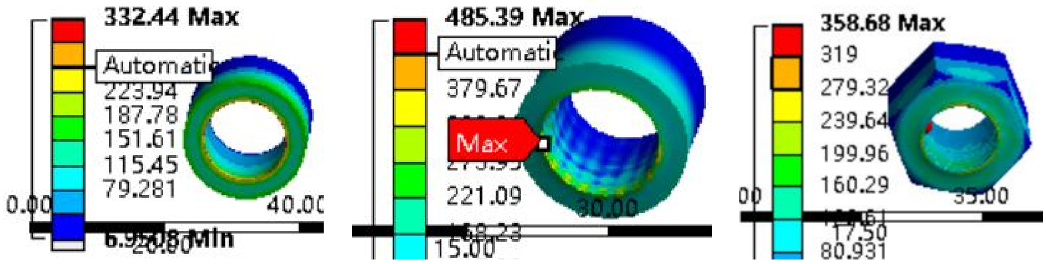


Fig.8 Von-Mises stress cloud map at cross sections I to VI.

Figure 8 shows that the three modeling methods have basically the same maximum stress position at the connection between the nut and bolt, with a maximum error of only 2.26%. There is a slight difference in stress magnitude at the washer position of section II, but the trend is the same. In other screw positions, but at the first thread position of section III and section IV, there is a significant difference in stress trend between model 3 and model 1, and model 2, mainly because model 3 effectively reflects the stress concentration at the root of the thread. In contrast, model 1 and model 2 cannot reflect the stress concentration due to the model. At the contact between the thread and nut on section V, the stress levels of model 2 and model 3 are higher than those of model 1, mainly because model 2 The relevant parameters of the input bolt when using non-separated contact reflect the situation of the bolt in the model calculation process.



model1 model2 model3

Fig.9 Von-Mises stress cloud map of nut.

The stress distribution cloud map of the nut in Figure 9 shows that the stress distribution of Model 1 and Model 3 is closer to the first connection position of the nut and bolt at the beginning of the rotation. As the number of threads increases, the stress gradually decreases. However, in Model 2, there is a larger axial force distributed along the circumference. From the analysis of von-Mises stress results, it is also found that Model 1 and Model 3 are closer and closer to the theoretical calculation of about 1.5. The maximum von-Mises stress level of Model 2 far exceeds the theoretical calculation value.

5. Conclusion

From the analysis results, it can be seen that all three methods are in good agreement with the theoretical values at the screw position, indicating that finite element simulation is feasible in the analysis of bolt pre-tightening force. Moreover, the stress concentration positions are mainly on the first and second threads of the nut support surface, the intersection between the bolt rod and the thread, and the connection between the nut and the bolt. This is consistent with the bolt failure statistics in reference [15].

From the bolt calculation results, it can be seen that the stress distribution states of the three methods are basically the same. In the ordinary simulation calculation process, all three methods are relatively effective. However, compared with Model 3 and Model 1, Model 2 has a higher stress result at the position where the bolt and nut start to rotate, while Model 3 is more effective in reflecting the stress situation at the root of the thread.

From the comparison of the nut's calculation results, it can be seen that at the position where the bolt and nut start to rotate, the calculation results of the nuts in Model 1 and Model 2 are more similar. The one closest to the theoretical calculation is about 318.4 MPa, with a maximum difference of about 12%. However, the maximum stress calculated using the Model 2 method is 485.39 MPa, far exceeding the theoretical calculation value.

The calculation results of Model 1 and Model 3 are closer. Therefore, the simple modeling method of Model 1 can be used in general situations, but for components that may fail due to fatigue at the root of the thread, it is necessary to use refined modeling in the fatigue life prediction.

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