International Journal of Advanced Engineering and Technology Research
ISSN: 3106-8677 (Print), ISSN: 3106-8685 (Online) | Vol. 1, No. 2, 2026

Research on the Operational Reliability of the Crankshaft Rotation Device of the Compressor Unit

Xingbo Tang [1], Jun Tang [1], Yue Li [2], Yong Li [3], Jianli Wu [3], Ningdan Xiong [1]

[1] China National Petroleum Corporation Southwest Oil and Gas Field Branch, Chongqing Gas Mine, Chongqing 400707, China

[2] Chongqing Shale Gas Exploration and Development Co., Ltd., Chongqing 401121, China

[3] China National Petroleum Corporation Jiechai Power Equipment Co., Ltd. Chengdu Compressor Branch, Chengdu 610100, China

Abstract: This paper conducts a structural safety optimization study on the gear strength of the automatic cranking device for natural gas reciprocating compressor units, aiming to enhance their overall reliability. By replacing the gear material with 20Cr and optimizing the radius of the fillet at the tooth root, the contact stress on the tooth surface was reduced by 46.3%, and the bending stress at the tooth root decreased by 31%. Both of these indicators fell below the allowable stress threshold after optimization. The research results show that the reliability of the optimized cranking device has been significantly improved, providing an effective solution for the safe and efficient operation of natural gas compressor units.

Keywords: Natural gas reciprocating compressor unit, Crankshaft rotation device, Structural safety, Optimization design.

1. Introduction

In recent years, the total consumption of natural gas in our country has been increasing continuously, with an external dependence rate as high as 42.3%. This has led to a continuous rise in the import volume of natural gas, and the vigorous development of natural gas is an important means to solve the energy security of our country [1]. The gas compressors in the booster stations, as the core equipment for boosting and transporting natural gas, require a turning operation before startup and after shutdown. However, some compressors do not have mature turning devices, and the turning method is manual turning, which is labor-intensive and inefficient, and the operation accuracy is difficult to guarantee. Moreover, the manual turning lacks real-time monitoring means, making it impossible to promptly detect equipment abnormalities, and there are safety hazards. Even with the existing mature turning devices, the output gears still bear complex and variable loads, and the transmission performance is poor. In harsh working environments, faults such as tooth breakage, pitting, and wear are prone to occur, seriously affecting the efficiency and reliability of the preparatory work before the compressor unit starts. Therefore, it is necessary to carry out the structural design of the turning device for the compressor unit and the analysis of the transmission stability.

At present, the structural design and fault analysis of the turning devices for steam turbines and generators are mainly carried out, while there are few reports on the relevant research on the turning devices for natural gas reciprocating compressors [1-8]. This paper focuses on the structural design of the turning device for natural gas compressors and provides an automated turning device. It uses the method of combining theoretical verification calculation and finite element analysis to optimize the design of the turning gear, and conducts a transmission performance analysis around the turning device to provide guarantees for the safe, efficient, and stable operation of the compressor unit.

2. Structural Design of the Crankshaft Rotation Device for the Compressor Unit

The cranking device of the compressor unit is composed of a cranking mechanism, a planetary gear reduction device and an air motor. The schematic diagram of the meshing between the air cranking device and the flywheel of the compressor is shown in Figure 1.

The cranking device needs to actively mesh and drive the flywheel of the compressor before the reciprocating compressor is started, thereby driving the crankshaft to rotate, ensuring that all moving parts are adequately lubricated and static friction is eliminated. Therefore, the cranking gear is the most vulnerable part of the cranking device. The structural safety analysis of the cranking gear will be carried out in the following text.

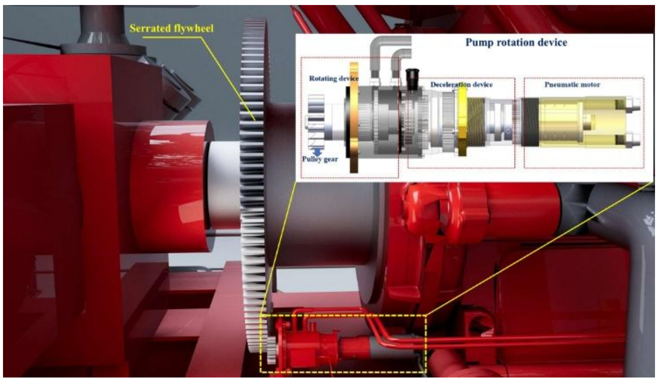

Figure 1. Overall structure of the pneumatic turning device

3. Finite Element Analysis of the Disc Gear of The Pneumatic Turning Device

3.1. Establishment of the Finite Element Model of The Crankshaft Gear

Based on the 3D software, a meshing model of the crankshaft gear and the compressor unit was established. Considering the efficiency of the computer and the accuracy of the simulation, the meshing model needs to be simplified. ① Some minor features, such as keyways and some rounded corners, are ignored; ② The details of some local structures are simplified; ③ The important structures that have a significant impact on the crankshaft gear are retained.

To ensure the accuracy of the simulation analysis, the settings should be made according to the actual materials of the crankshaft gear and the flywheel of the compressor unit. The materials of the crankshaft gear and the flywheel of the compressor unit are structural steel and 40Cr respectively. The material properties are shown in Table 1.

Table 1. Material properties of the crankshaft meshing pair

Gear name	Young's modulus (GPa)	Material density (kg/m^3)	Poisson's ratio
Pump gear	200	7800	0.3
Compressor flywheel	209	7890	0.295

The purpose of meshing is to discretize the model into several physical units and establish certain relationships among these units in order to obtain the precise solution of the model. The quality of the mesh directly affects the calculation accuracy of the simulation results [2]. Therefore, in order to balance the mesh quality and computational efficiency, considering the irregular shape of the gear structure, the crankshaft gear and the compressor flywheel are overall meshed with tetrahedral meshes. The overall mesh element size is 20mm, and the mesh is densified near the contact surfaces. The mesh element size is 2mm, and the mesh element size is 0.2mm on the contact surfaces of the gears. The finite element meshing model is shown in Figure 2.

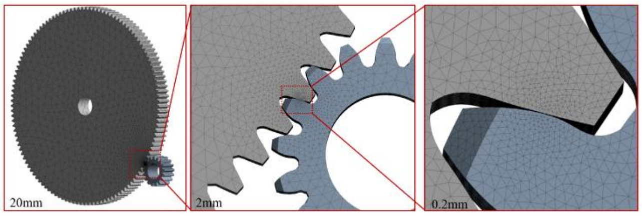

Figure 2. Grid partitioning model

Set the contact mode between the crankshaft gear and the compressor flywheel to be frictional contact, with the friction coefficient set to 0.15. Establish a rotational pair relative to the ground between the crankshaft gear and the compressor flywheel, and apply a torque of 207 N·m to the crankshaft gear in the X direction. Define the rotation of the compressor flywheel around the X axis by 10°. The finite element model of the meshing between the crankshaft gear and the compressor flywheel is shown in Figure 3.

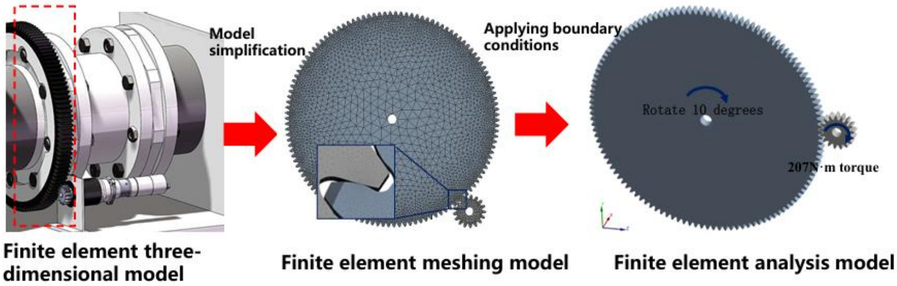

Figure 3. Finite element analysis model

3.2. The Influence Law of Contact Stress on The Gear Tooth Surface of The Turning Gear

The contact stress response law of the crankshaft gear in the designed pneumatic crankshaft device was analyzed to reveal the distribution characteristics and time-varying laws of the contact stress, providing a theoretical basis for the optimization design of the crankshaft gear. The contact stress on the tooth surface of the crankshaft gear is shown in Figure 4. When the crankshaft gear is made of traditional structural steel material, when the gear teeth mesh with the flywheel, the contact stress reaches its peak of 659.39 MPa at 0.24 seconds, but the contact stress exceeds the allowable contact fatigue stress of 603.2 MPa, resulting in low structural strength and poor safety.

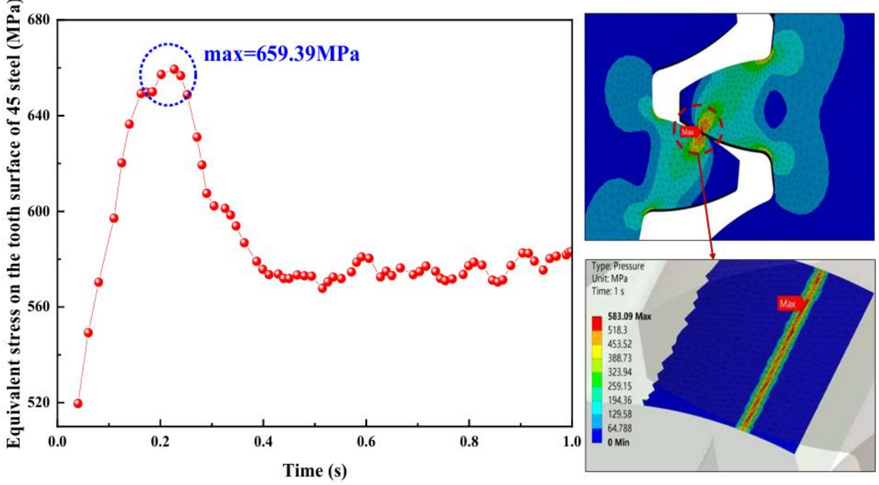

Figure 4. Contact stress on the gear tooth surface of the turning gear

The reason for this is that in the initial stage of gear meshing, the contact area on the tooth surfaces is very small, approaching a point contact. According to Hertz contact theory, during point contact, the contact stress is concentrated in a very small area, resulting in extremely high local stress. As the load increases, the contact area gradually expands from a point contact to a surface contact, and the contact stress distribution tends to be small-scale periodic fluctuations.

The 20Cr material with a higher elastic modulus was selected for comparison with the original structure steel of the handwheel gear. The properties of the 20Cr material are shown in Table 2.

Table 2. Properties of 20Cr material

Material Name	Contact Fatigue Limit (MPa)	Bending Fatigue Limit (MPa)	Elastic Modulus (GPa)	Material Density (kg/m^3)	Poisson's Ratio
20Cr (carburized and quenched)	650	400	208	7800	0.3

Simulations were conducted under the same boundary conditions. The contact stress on the tooth surface of the crankshaft gear after changing the material to 20Cr is shown in Figure 5.

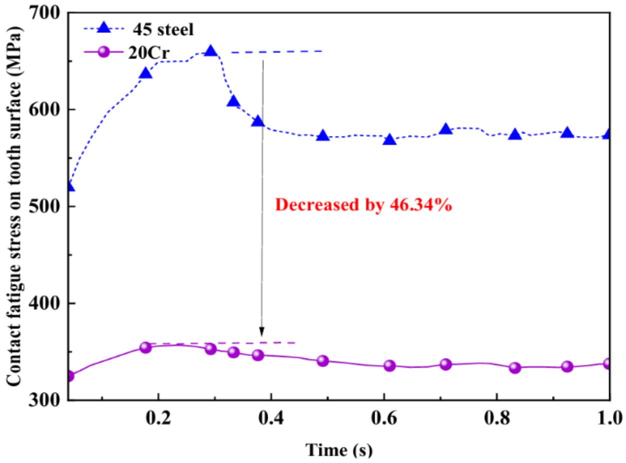

Figure 5. The variation law of contact stress between different materials of the crankshaft gears

Analysis shows that the equivalent stress of the tooth surface contact of the 20Cr crankshaft gear is 353.85 MPa, which is 46.34% lower than that of the structural steel and is less than the fatigue stress of 603.2 MPa of the tooth surface contact. It meets the strength requirements and significantly improves the safety.

3.3. The Influence Law of Root Bending Stress of The Crankshaft Gear Teeth

The distribution characteristics of bending fatigue stress at the tooth root and its relationship with the tension side and compression side are the key factors for evaluating the fatigue strength of the gear for turning. The difference in stress states between the tension side and the compression side will lead to different behaviors of fatigue crack initiation and propagation in the tooth root area, thereby affecting the overall fatigue life of the gear. Therefore, studying the relationship between the bending fatigue stress at the tooth root and the tension side and compression side is of great significance for optimizing the design of the gear. The distribution law of the bending fatigue stress at the tooth root of the gear for turning is shown in Figure 6.

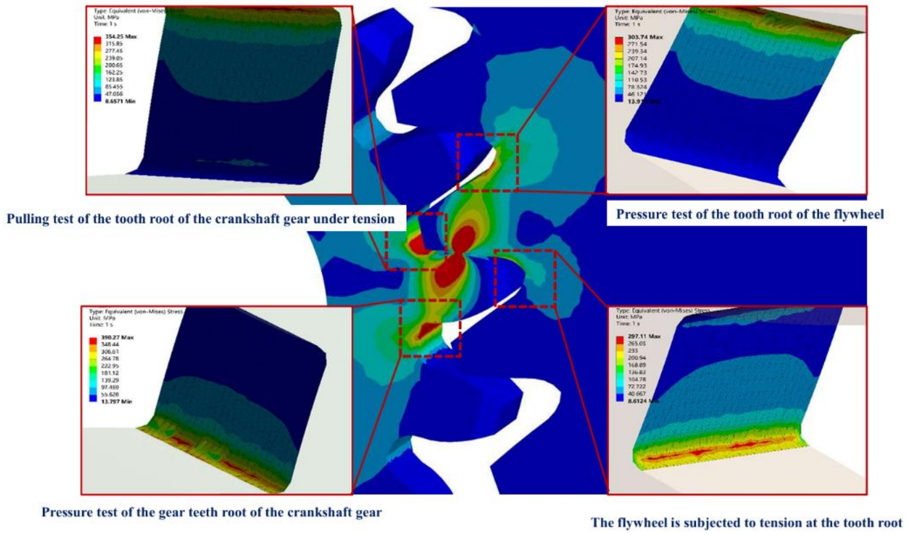

Figure 6. The stress distribution pattern of the root bending of the crankshaft gears and the flywheel teeth

When the compressor unit is undergoing the turning operation, the turning gear serves as a key component for transmitting the output torque of the pneumatic motor to the compressor flywheel. The root part of the gear will be subjected to bending stress, resulting in significant stress concentration at the transition rounded corner of the gear root. The time-varying law of the bending stress at the root of the turning gear and the compressor flywheel is extracted, as shown in Figure 7.

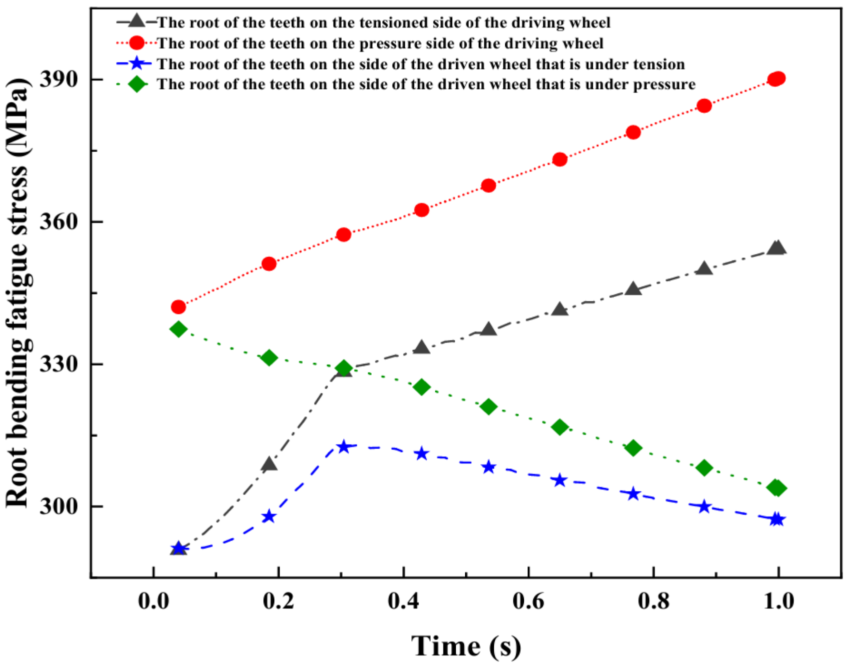

Figure 7. The time-varying law of bending stress at the tooth root and other effects

The maximum equivalent stress at the root of the flywheel teeth occurs on the compressed side, with a value of 337.33 Mpa. The maximum equivalent stress at the root of the teeth on the tension side is 312.8 Mpa. The maximum equivalent stress at the root of the crankshaft gear occurs at the transition corner on the compressed side of the crankshaft gear, with a value of 390.27 Mpa. The maximum equivalent stress at the root of the tension side is 354.25 Mpa. The transition corner on the compressed side of the crankshaft gear is the most dangerous position, exceeding the allowable fatigue equivalent stress at the root by 301.6 MPa. Long-term operation will lead to the risk of bending fatigue fracture at the root of the teeth.

Based on the analysis in the previous text, the root of the teeth on the pressure side of the gear is the risk area for damage. The study systematically investigated the influence law of the root transition arc radius of the teeth on the bending strength of the gear. According to the ISO 6336-3:2019 standard, the root transition arc radius is (0.25~0.4) × the module. The module of the gear drive device is 8mm, and the recommended standard value of the root transition arc radius is 2~3.2mm. This paper selected the root transition arc radius of the gear root on the pressure side as 2, 2.6, and 3.2mm for comparative analysis. The influence law of the root transition arc radius on the bending equivalent stress of the root of the teeth on the pressure side of the gear drive gear is shown in Figure 8.

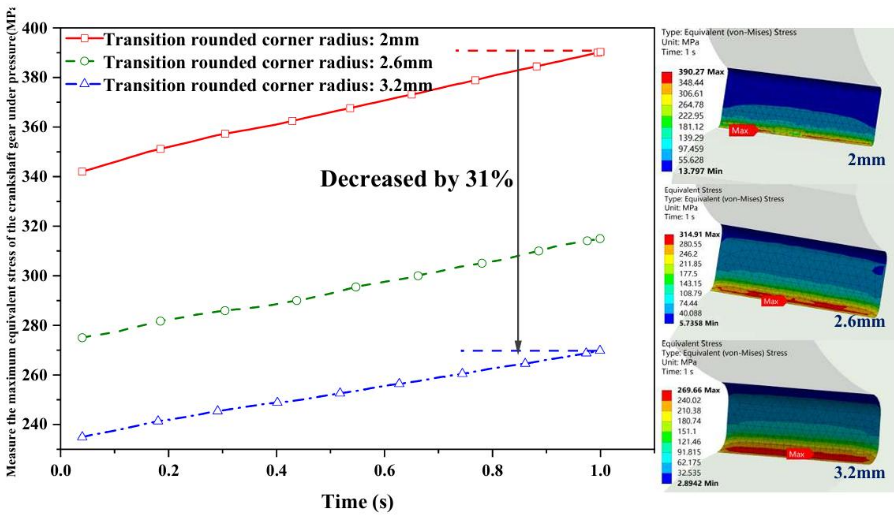

Figure 8. The influence law of transition rounded corner radius on the bending stress at the tooth root of the pressure side of the crankshaft gear

Analysis shows that within the reasonable design range of the root transition fillet of the teeth, as the radius of the root transition fillet increases, the bending fatigue stress at the tooth root decreases, while the bending fatigue strength increases. When the radius of the root transition fillet is 3.2mm, the maximum stress is 269 MPa, which is 31% lower than that when the radius is 2mm, and is less than the allowable bending fatigue stress of 294.5 MPa for the tooth root. Therefore, the optimization of the tooth root bending fatigue is relatively reasonable.

4. Summary

This study focused on the stress concentration problem of the crankshaft gears in the compressor unit. Through finite element analysis, the material and structure of the gears were optimized. After the improvement, the contact stress on the gear surface and the bending stress at the gear root were significantly reduced by 46.3% and 31% respectively, both of which were superior to the allowable stress threshold, effectively alleviating the risk of stress exceeding the limit. This solution effectively improved the maintenance efficiency and operational reliability of the gas compression disc, and has significant engineering value for ensuring the safety of energy equipment.

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