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Analysis of Planetary Gear Transmission Characteristics Based on ANSYS

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Authors' contributions

This work was carried out in collaboration between both authors. Both authors read and approved the final manuscript.

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Original Research Article

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ABSTRACT

Planetary gear transmission is widely used in roadhead, crane and other transmission equipment, its reliability and stability are related to the service life and work efficiency of equipment, but the current research on planetary gear transmission device is not comprehensive, lack of accurate analysis of gear meshing process load. Therefore, in this paper, ANSYS software as a platform, through the construction of dynamics module, to solve the gear teeth surface, gear teeth root and gear teeth tip stress analysis and deformation and displacement, the finite element analysis. The research results provide necessary technical support for the design and production of planetary gear transmission.

Aims: Through finite element analysis of planetary gear transmission, the dynamic parameters of planetary gear transmission are solved, and the reliability and stability of planetary gear transmission equipment are improved.

Study Design: By constructing the dynamics module of planetary gear, the stress analysis, deformation and displacement of gear tooth surface, gear tooth root and gear tooth top are solved, and the finite element analysis is carried out.

Place and Duration of Study: North China University of Water Resources and Electric Power, from March to July 2022.

Methodology: The finite element analysis was carried out on the model of planetary gear transmission mechanism, and the model was established and simplified by using SolidWorks software. The modal analysis and dynamic analysis were carried out by using ANSYS software, and the meshes were divided and constraints were added. Finally, the parameters such as stress, strain and displacement of the meshing position were solved.

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Results: The result of model meshes is 357575 units and 204871 nodes. The algorithm has 219 iterations and 15 convergence points. The experimental results show that there is a meshing position with the worst contact fatigue strength in the contact area between the outer ring gear and the planetary gear tooth surface. At this position, the contact stress value of the tooth surface is 22626Pa.

Conclusion: In this paper, the planetary gear device is modeled by SolidWorks software, and the finite element analysis is carried out by ANSYS software to obtain the stress and strain parameters in the transmission process of the device, as well as the worst meshing point of the gear, the most vulnerable point of fatigue and other dangerous points. It lays a certain technical foundation for the optimization and improvement of gear and the design and production of planetary gear transmission device.

Keywords: Stress and strain; planetary gear; finite element analysis; ANSYS software.

1. INTRODUCTION

In recent years, the manufacturing and research and development of planetary gear transmission equipment adopts modern manufacturing technology and research and development theory, constantly absorbing new structures and new materials at home and abroad, and the technical content and performance level of products have been greatly improved [1]. However, as an important component of roadheader and deceleration equipment, the research on load and stress analysis of planetary gear transmission mechanism plays an important role in improving its performance [2]. Abnormal vibration or unequal load will seriously affect its work efficiency, and even cause fatigue damage of system equipment, and shorten the service life [3]. With the rapid development of science and technology, finite element analysis technology is gradually mature. Researchers can use finite element software to carry out efficient strength check and modal analysis of planetary gear transmission mechanism, and effectively monitor the efficiency of equipment [4]. However, there is a lack of modal and dynamic analysis for the transmission structure with high working intensity and large load impact.

Germany and Japan have long held the top spots in the field of planetary gear transmission technology. Products from the planetary reducer series are likewise technologically advanced in other industrialized nations like France and Switzerland. The development of several documents as a result of numerous international academics' research on planetary gear load distribution has encouraged the company to focus on developing gear reducer products with large bearing capacities and high precision [5]. At the same time, research on the working performance of gears has been encouraged, the light computation of gears has been gradually improved, and the content of noise reduction, vibration control, and optimal design has been moved to current technology [6].

Through the active cooperation between enterprises and universities, it has made a great breakthrough in the optimization design, structure analysis and dynamics analysis of planetary transmission, which has made great progress in the development and application of various kinds of planetary transmission equipment in our country [7]. In addition, the improvement and development of finite element analysis technology provide effective support for the stress and strain analysis of planetary gear transmission [8].

Based on the above situation, this paper takes optimizing and improving the transmission efficiency of planetary gear as the goal, conducts finite element analysis on the planetary gear transmission mechanism model, establishes and simplifies the model with SolidWorks software, conducts modal analysis and dynamic analysis with ANSYS software, and conducts grid division and constraint addition respectively. The parameters such as stress, strain and displacement of meshing position were solved, and the analysis of these parameters provided necessary technical support for the optimization of planetary gear transmission, design and production.

2. EXPERIMENTAL DETAILS AND METHODOLOGY

2.1 Solid Modeling and Assembly of Planetary Gear Transmission

The complexity of planetary gear transmission mechanism is high, and each gear in assembly also has the requirement of tooth number module, so the model is often simplified in general characteristic analysis [9]. The planetary gear transmission shall include a sun wheel (namely the central wheel) and at least two planetary wheels [10]. The number of teeth of the sun wheel and planetary wheel shall meet a certain number of relations to ensure the correct meshing between the two wheels and the normal rotation of the gear system [11]. In this paper, the SolidWorks modeling software is used to establish the model of each gear part. In order to ensure the correct meshing of the gear system, the parts are designed and assembled according to the integral multiple of the center Angle of the rotation Angle of the sun wheel equal to the tooth distance. The planetary gear model is shown in Fig. 1.

2.2 Construct the Transient Dynamics Module

(1) Open ANSYS software to select Transient Stuctural unit and import the planetary gear model. Select New Design Modeler Geometry to enter the Geometry processing interface, where the geometric units are uniformly named for subsequent operations. Set the material of the Model in the Model unit as structural steel. Then name the parts separately, as shown in Fig. 2.

(2) Set the connection pair and add five rotary pairs to the model, which are respectively added to the outer ring surface of the gear outer ring and the axial hole surface of three planetary wheels and one sun wheel. The rotary pairs added to the sun wheel and the planetary wheel are shown in Fig. 3.

The meshing contact of the planetary gear system is mainly the contact between the planetary wheel and the outer gear ring and between the planetary wheel and the sun wheel [12], as shown in the figure. There are six groups of contacts in this model. For example, the contact Settings of outer gear ring and planetary wheel should first hide the gear outside the target part, select a tooth surface of planetary gear 1, create a naming selection, and name the target part, and select the geometric item to apply the size. Similarly, create a naming selection for each of the five parts. Fig. 4 shows creating the resulting diagram.

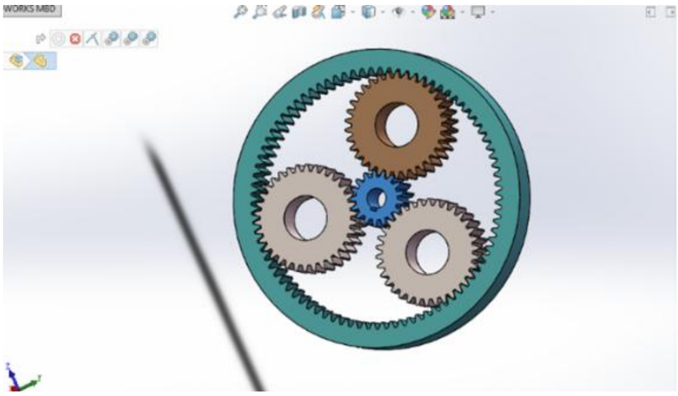


Fig. 1. Build model diagrams in SolidWorks

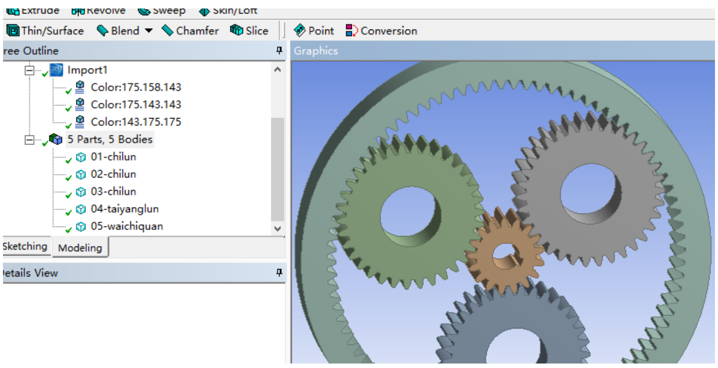
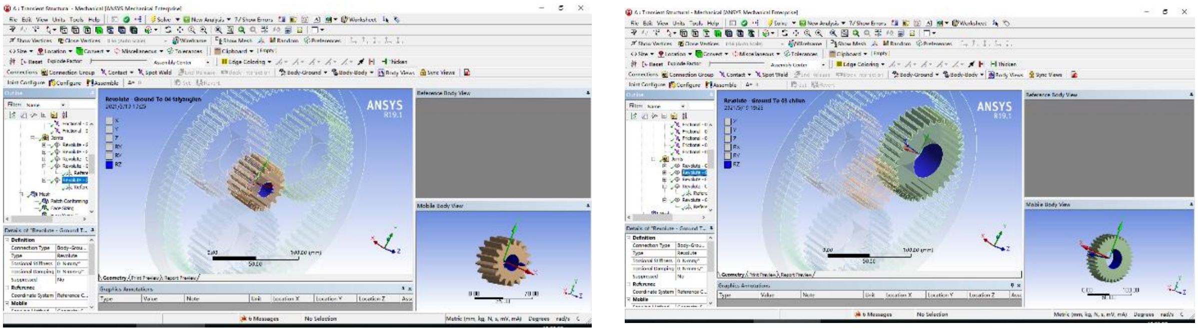


Fig. 2. Name the parts in SolidWorks



(a) Add a turning pair to the center gear **(b) Add a turning pair to the planetary gear**

Fig. 3. Add a turning pair to the gear in SolidWorks

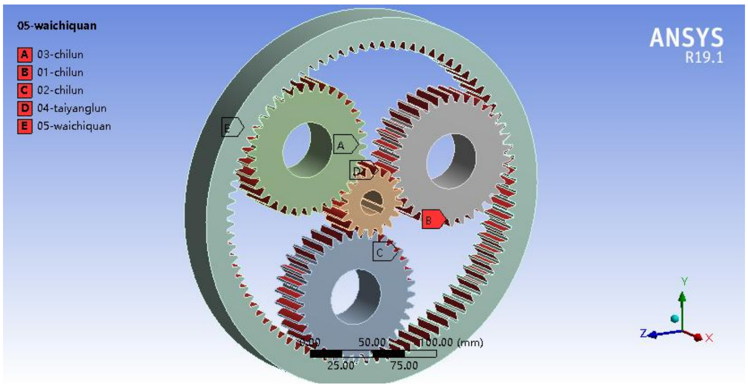


Fig. 4. Create a naming selection in SolidWorks

The scope limiting method in contact is set to Named Selection to select contact geometry and target geometry using the method described above. The gear meshing connection type is friction, the friction coefficient is generally 0.15, the normal stiffness factor is 1 by default, the updated stiffness should be set to each iteration, the time part control is automatic bisection, and the error offset in the geometric modification is 0.03 mm. Fig. 5 shows the contact surfaces of the planetary wheel, the sun wheel and the outer gear ring.

(3) Model meshing.

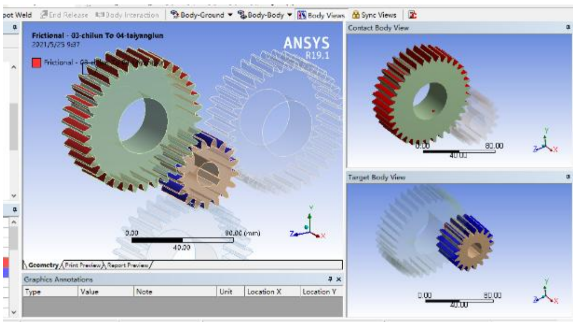
Meshing is one of the most critical steps in the analysis task. A reasonable meshing can not only speed up the calculation of the model, but also improve the accuracy of the data [13]. The Mesh Tool of ANSYS software provides a very fast operation way. At the same time, attention should be paid to the combination of the physical characteristics of the model, select the appropriate element shape, size and the position of the intermediate node, and Mesh the structure of the planetary gear system [14]. The quality of meshing determines the economy and accuracy of the analysis results, and the effective meshing results are the stress and displacement meshes generated by the finite element model as the carrier, which can be continuously changed.

In the analysis of this paper, the geometry structure is selected as the whole geometry, and the tetrahedral partition method and patch conformal algorithm are used to create five size adjustments. The range limiting method is selected as named selection, and the default unit size is set to 2 mm. The result is that the model grid is divided into 357,575 units and 204,871 nodes. And the meshed model is shown in Fig. 6.

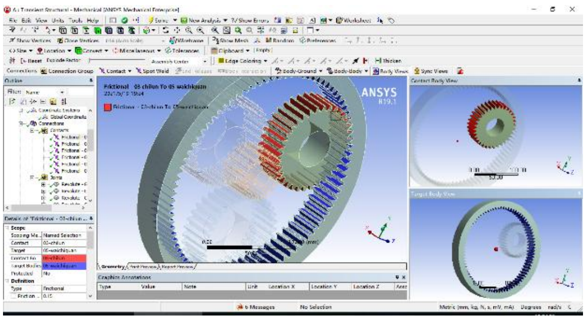
In this paper, we mainly study the force on the tooth surface of the gear, so the longitudinal division of the tooth is mainly carried out in the mesh division, and the outer surface is not processed in addition.

For further exploration in the face of the gear meshing, check the unit quality figure are shown in Fig. 7.

By comparing the above results, it can be seen that the closer the quality coefficient of Mesh Metrics element is to 1, the closer the Mesh element shape is to tetrahedron, with regular shape, uniform size and high partition quality. It can also be seen from the figure that the number of elements with element quality coefficient close to 1 is relatively small and concentrated on the tooth surface of each gear, which also indicates that the result of meshing is close to the expected situation and the meshing situation is quite ideal. Fig. 8 shows the number and location of units corresponding to different element quality coefficients under different function methods.



(a) Contact between planetary gear and center gear



(b) Contact between planetary gear and outer ring gear

Fig. 5. Contact between gears

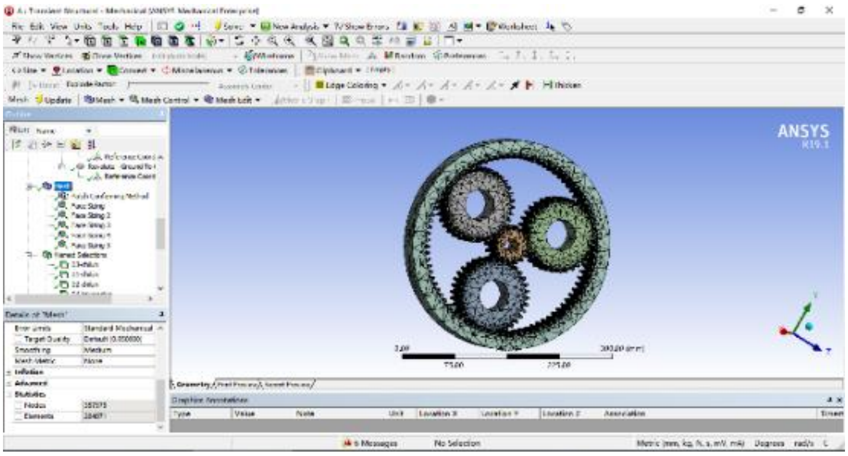
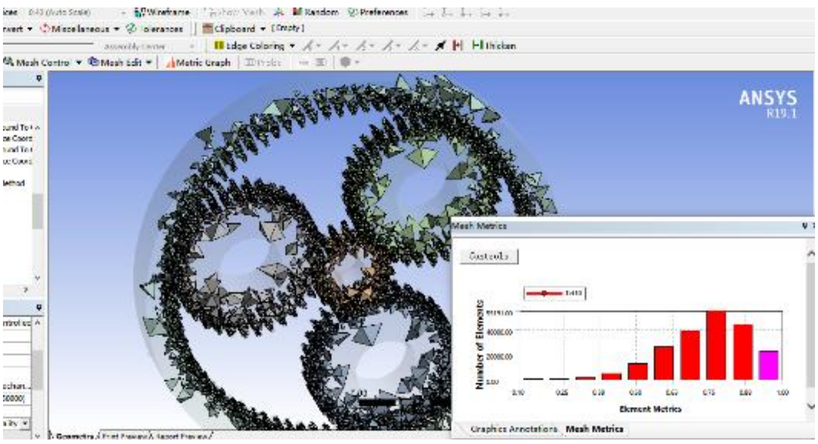
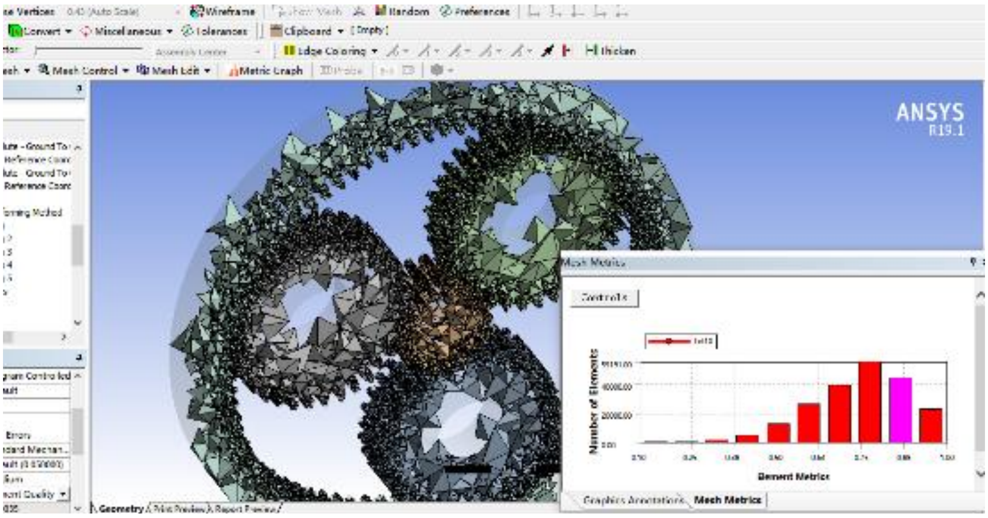


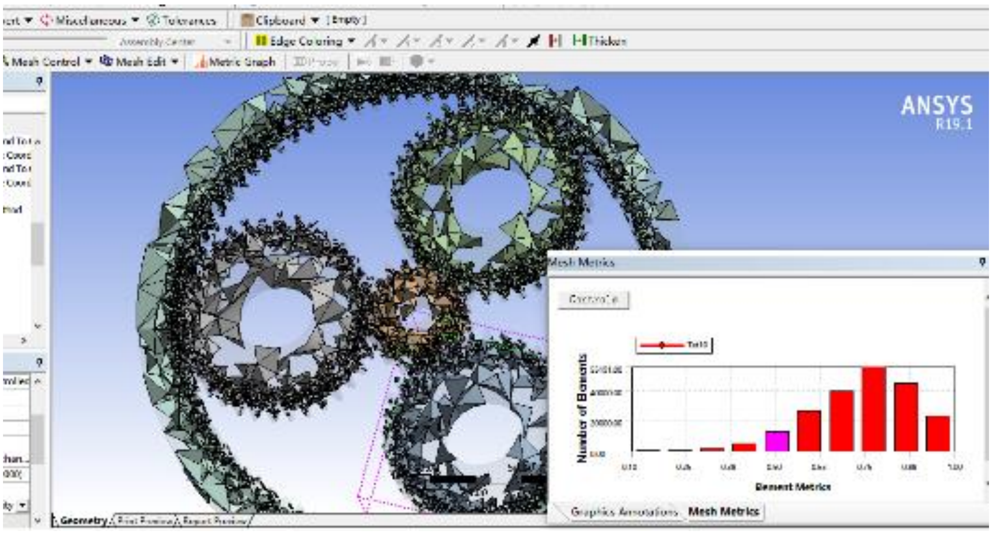
Fig. 6. Meshing diagram of gear model



(a) Unit quality diagram with quality coefficient equal to 1

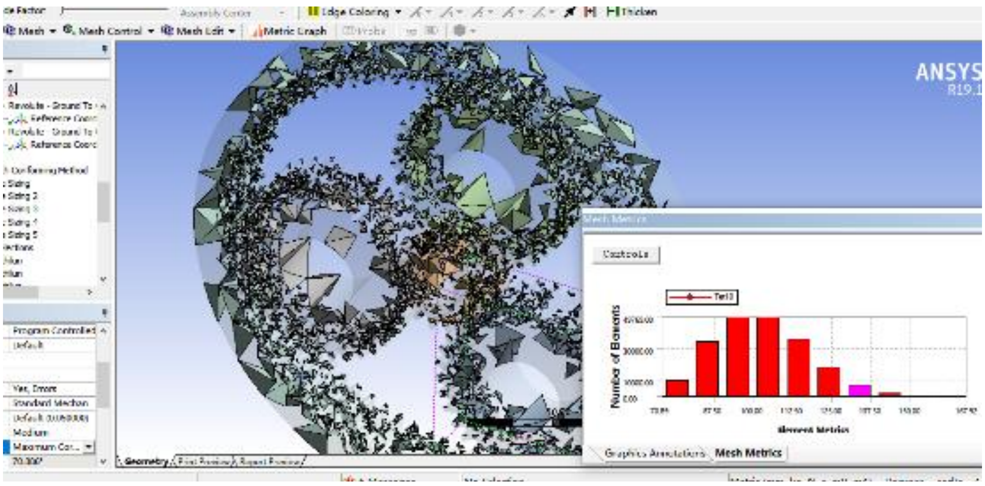


(b) Unit quality diagram with quality coefficient equal to 0.88

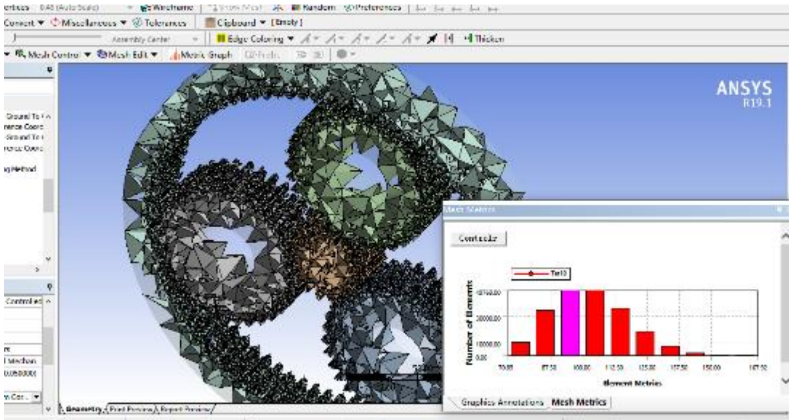


(c) Unit quality diagram with quality coefficient equal to 0.5

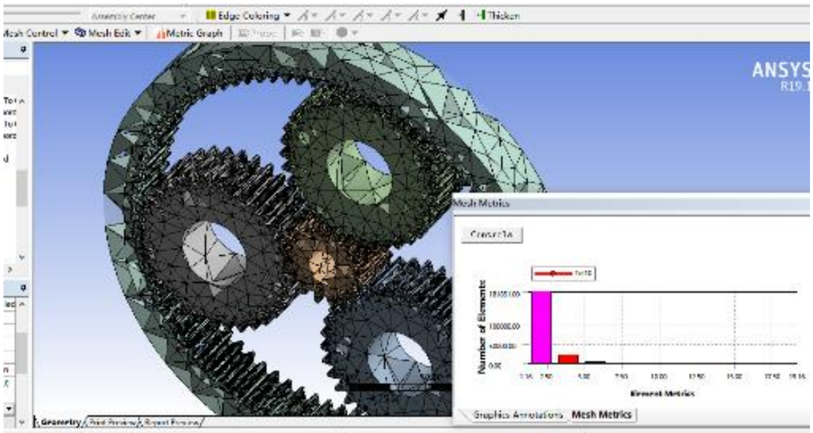
Fig. 7. The unit quality diagram



(a) Quality diagram at Maximum Corner Angle equal to 137.5



(b) Quality diagram at Maximum Corner Angle equal to 100



(c) Quality diagram at Aspect Ratio equal to 2

Fig. 8. The quality diagram

3. RESULTS AND DISCUSSION

3.1 Solving Model

(1) Setting constraints on the model is also a very important operation step in the finite element analysis [15]. The constraints set for the planetary gear train in this analysis are: the gear train is rigidly coupled. That is, the sun wheel, the planet wheel and the outer ring gear are pairwise divided into the inner rigid region and the outer rigid region [16]. In the second part, load should be applied to the model, and the main power should be applied to the sun wheel. The rotation center of the gear generates speed, and the torque is transferred to the planet wheel through the center wheel, and the planet wheel will transfer the torque to the outer gear ring, that is, the inner and outer rigid zone is used as the carrier for transmitting speed or torque, and the whole system has the same degree of freedom [17]. Fig. 9 shows the situation of sun wheel with rotating auxiliary load.

(2) The setting of step control in the analysis setting: the initial substep is 25, the minimum substep is 20, and the maximum substep is 250.

(3) Check the convergence curve in the solution information. The purple line represents the convergence line of the force, and the light blue line represents the standard line of the force. When the substep converges, the green dotted line represents the convergence of this step. As shown in Fig. 10, the algorithm takes 219 iterations and 15 convergence points appear.

3.2 Deformation and Overall Displacement

In this paper, the rotational speed of the center wheel of the planetary transmission device is set counterclockwise. According to the characteristics of meshing transmission, the planetary wheel rotates clockwise. Fig. 11 shows the displacement of each transmission component under this transmission characteristic.

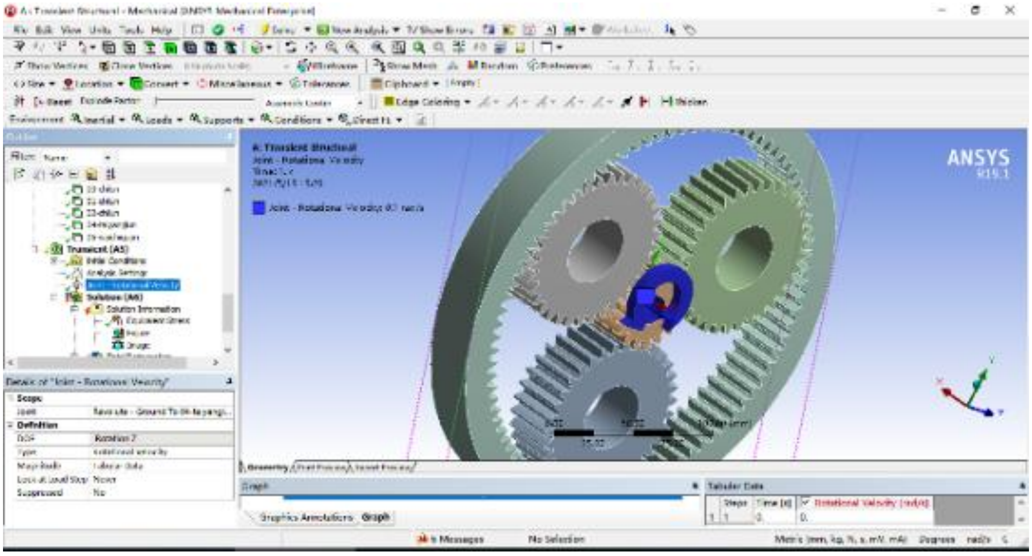


Fig. 9. Add torque to the center gear

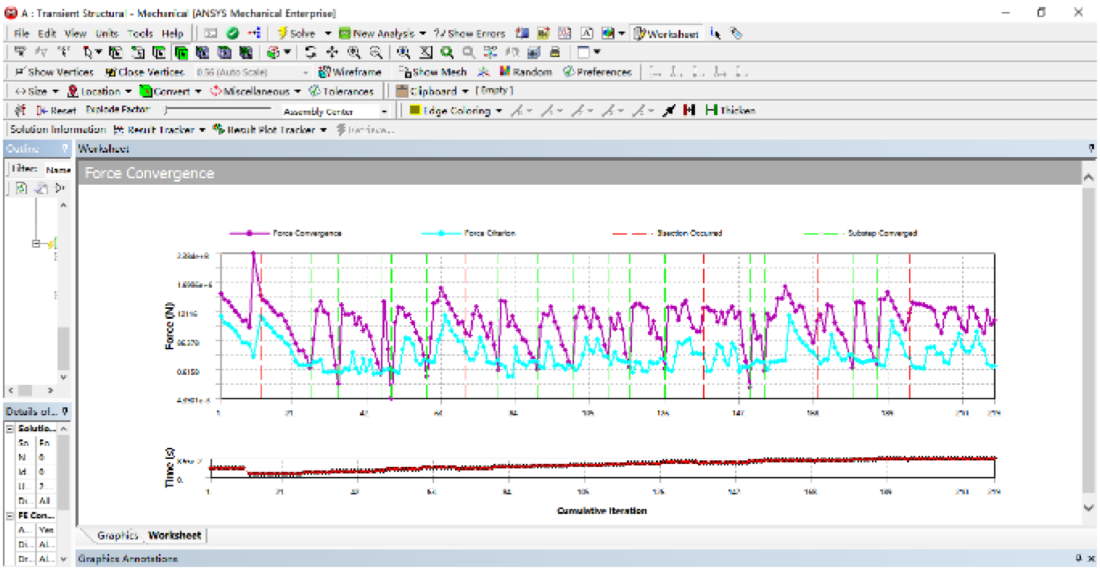


Fig. 10. The convergence curve

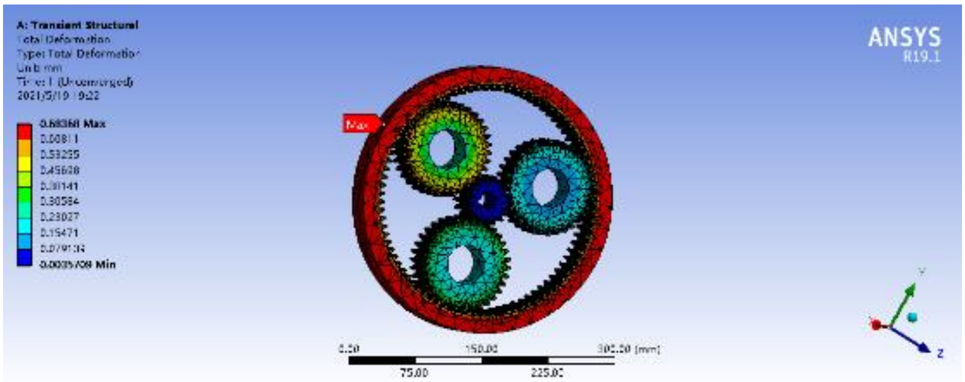


Fig. 11. The model deformation cloud image

3.3 Dynamic Tooth Surface Contact Stress

The materials of the planetary gear mechanism in this paper are the same, so there will be a meshing position with the worst contact fatigue strength in the contact area between the outer gear ring and the planetary gear tooth surface, as shown in Fig. 12, where the contact stress value of the tooth surface is 22626Pa. The contact of gear meshing is not a simple linear contact. In planetary gear transmission system, the actual contact area between gears is a uniform contact zone with stable stress fluctuation. The analysis results are consistent with the actual situation.

3.4 Dynamics of Root Apex Bending Stress

The planetary gear transmission has good stability [18], but with the increase of working time, there are corresponding worst meshing regions on the central wheel, the planetary wheel

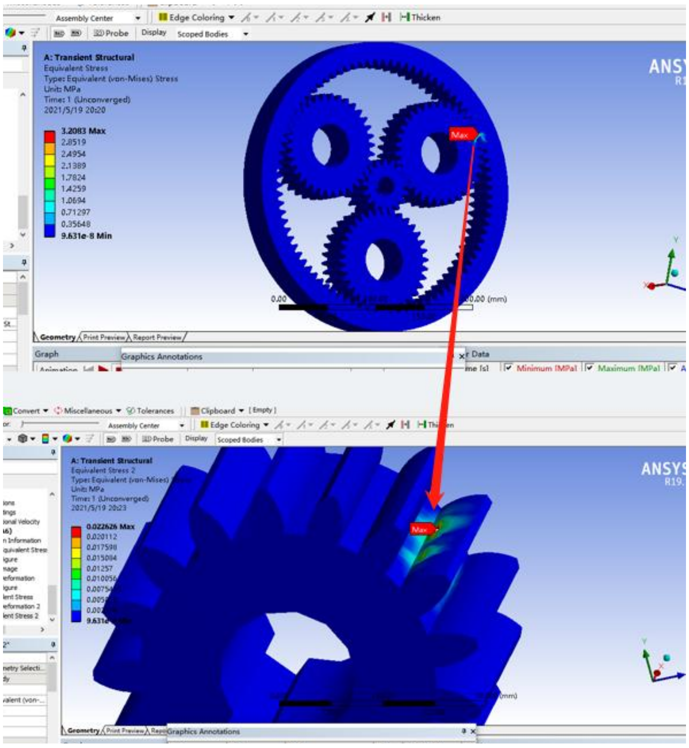


Fig. 12. The surface contact stress cloud diagram between gears

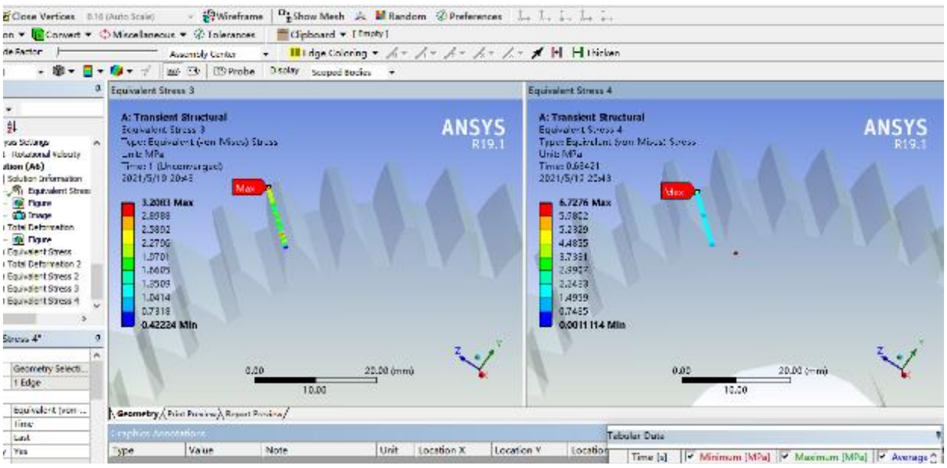


Fig. 13. The stress distribution diagram of gear root and gear top

and the outer gear ring [19]. After long-term work, the gear will produce fatigue cracks under the action of load, generally appeared earlier in the root of the gear tensile edge, so the fatigue fracture is easy to occur in the tensile edge [20]. Therefore, this paper takes the tensile stress of gear as the research object to analyze the planetary gear transmission mechanism. The analysis results are shown in Fig. 13, the stress distribution cloud diagram at the worst meshing position. The changes of stress and strain are basically the same, and the variation trend of strain with equivalent stress is normal.

4. CONCLUSION

In this paper, the planetary gear device is modeled by SolidWorks software, and the finite element analysis is carried out by ANSYS software to obtain the stress and strain parameters in the transmission process of the device, as well as the worst meshing point of the gear, the most vulnerable point of fatigue and other dangerous points. The result of model meshes is 357575 units and 204871 nodes. The algorithm has 219 iterations and 15 convergence points. The experimental results show that there is a meshing position with the worst contact fatigue strength in the contact area between the outer ring gear and the planetary gear tooth surface. At this position, the contact stress value of the tooth surface is 22626Pa. It lays a certain technical foundation for the optimization and improvement of gear and the design and production of planetary gear transmission device.

COMPETING INTERESTS

Authors have declared that no competing interests exist.

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