

Dynamic simulation analysis of aircraft engine planetary gear system

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Abstract. The Geared Turbofan (GTF) engine is used for aircraft propulsion with fan drive, characterized by its complex structure and operation in challenging environmental conditions. During its operation, variations in gear meshing and loads on the transmission shaft exacerbate vibration within the transmission system, leading to a decline in engine performance. This paper established three-dimensional solid models of the X-type aviation engine planetary gear system, input shaft, and output shaft, and employed ADAMS for multibody dynamics analysis, achieving simulation of engine operational vibrations. Modal and harmonic response analyses were conducted using ANSYS to observe and analyze critical parameters such as vibration displacement, vibration acceleration, gear meshing forces, modal resonance, and harmonic resonance, evaluating the impact of vibration amplitude, frequency, and their effects on system performance and longevity, providing guidance for vibration control and optimization of aviation engine systems.

Keywords: Planetary gear system; ADMAS; ANSYS; Dynamic simulation.

1. Introduction

The Geared Turbofan Engine (GTF) serves as a new generation of high-bypass turbofan engines, representing a new configuration in the development of next-generation aircraft engine technology. Its structural feature involves the addition of a planetary gear reducer between the fan rotor and the turbine, enabling the fan and turbine to operate within their respective optimal speed ranges through the variable-speed connection of this gear system[1]. The inclusion of the planetary gear reducer reduces the speed of the fan, further diminishing the tip speed and stress of the fan blades, thereby allowing for an increased bypass ratio of the engine. While meeting the limits of the fan blade tip speed, it also ensures the high-speed operation of the low-pressure turbine. This configuration significantly reduces the number of compressor and turbine stages, reducing the engine's weight, enhancing its efficiency, and optimizing the engine structure. The planetary gear system in the GTF component operates in high-speed and high-load conditions, resulting in a large volume, heavy weight, and complex structure. Moreover, the gears and bearings of the support gears work under harsh conditions, making it a component prone to multiple failures and poor reliability. Therefore, conducting an analysis and research on the planetary gear transmission mechanism is of great significance for promoting the development of reduction transmission technology in civil gear fan engines.

By conducting three-dimensional solid modeling of the gear transmission system of the X-type aviation engine, including the planetary gear system, transmission shaft system, and overall assembly, and performing multibody dynamics analysis using ADAMS and modal and harmonic response analysis using ANSYS, the vibration characteristics of the gear transmission system are analyzed

2. Creating a Three-Dimensional Model of The Planetary Gear System

2.1 Model creation of the planetary gear system

The GTF engine is characterized by its complex geometric shape, multi-level assembly, and high dimensional and precision requirements. Due to the various factors that affect the dynamic

characteristics of the system, the following modeling guidelines were established prior to modeling[2]:

- (1) The spline connection between the sun gear and the input shaft cannot be simplified.
- (2) The impact of bearings on the rigid body motion of the system can be simplified during modeling.
- (3) Reasonably allocate and set the material properties of each component as required.
- (4) The function of the planet carrier is for positioning and does not participate in rigid body motion analysis.
- (5) Simplify the fillets and rounded corners of the various components of the planetary gear system.
- (6) After creating the part models, perform overall assembly and adjustment to ensure the uniformity of gear center distances and axial lines between components.

The multi-body dynamics simulation analysis is based on the three-dimensional solid model. The main components of the three-dimensional solid model of the GTF engine gearbox include the input shaft, sun gear, five planet gears, two annular gears, output shaft, and planet carrier.

2.2 Gear System Parameters and Model Creation

Table 1. Relevant Parameters of Gearbox

| | Sun gear | Planetary gear | Ring gear | Keyway |
|----------------------|----------|----------------|-----------|--------|
| Number of teeth | 48 | 37 | 122 | 41 |
| Module | 3.5 | 3.5 | 3.5 | 3 |
| Helix angle (°) | 20 | 20 | 20 | |
| Pressure angle (°) | 22.5 | 22.5 | 22.5 | 30 |
| Addendum coefficient | 0.25 | 0.25 | 0.25 | |

The reduction system in the GTF engine utilizes a helical epicyclic gear system. In the three-dimensional model of the planetary gear system, the input shaft is connected to the central sun gear through a spline connection. The planet carrier serves to position the planet gears, while the ring gear functions as the output end and is connected to the output shaft through bolts. All gears in the system are helical planetary gears, which provide smoother power output, reduce energy loss and friction, improve transmission efficiency, evenly distribute loads, and enhance the reliability and durability of the entire transmission system under high-speed transmission conditions. The rated reduction ratio of this set of planetary gear system is approximately 2.54. The basic characteristic parameters of each gear are summarized in Table 1.

The creation and assembly of the planetary gear system, as well as the components for the input and output shafts, were accomplished using UG software, as shown in Fig. 1.

During the modeling process, it is recommended to simplify the end caps and the bolts. This simplification helps reduce the number of contacts and constraints that need to be added, effectively preventing over-constraints during load and contact setup. Furthermore, it also helps to reduce computation time. Fig. 2 illustrates this approach.

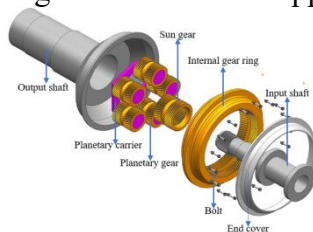


Fig. 1 Exploded view

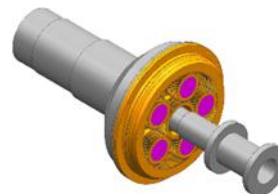


Fig. 2 Simplified assembly model

Fig. 2 Three-dimensional solid diagram of gearbox

After the assembly is completed, to ensure that the model can be properly constrained and have appropriate boundary conditions when imported into simulation software, it is necessary to adjust the gears that are experiencing interference. This is shown in Fig. 3.



(a) Gear interference before adjustment (b) Gear interference after adjustment

Fig. 3 Interference Adjustment Diagram between Gears

3. Kinematic Simulation Based on ADAMS

ADAMS is utilized to analyze the dynamic behavior of the gear transmission system. With ADAMS, dynamic simulation of the information processing system of the GTF engine can be performed to simulate gear contact, friction, backlash, and other factors. This allows for the study of system motion characteristics, torque transmission, speed variation, and other important parameters to evaluate the system's performance.

3.1 Establishment of Multi-body Dynamics Model for Gear Transmission System

The 3D solid model of the gear transmission system created in UG is converted to Parasolid format and then imported into the ADAMS software. The simulation parameters are set as follows:

The assembled components were modified to reflect their actual material properties. The materials and parameters of the various parts of the GTF engine gear transmission system are presented in Table 2.

Table 2. Material Properties and Parameters of GTF Engine Gear Transmission System Components

| | Material | Density (Kg/m ³) | Youngs modulus (Gpa) | Poisson's ratio |
|----------------|-----------|------------------------------|----------------------|-----------------|
| Input shaft | 40CrNiMoA | 7830 | 209 | 0.30 |
| Output shaft | 40CrNiMoA | 7830 | 209 | 0.30 |
| Sun gear | M50NiL | 7859 | 203 | 0.28 |
| Planetary gear | M50NiL | 7859 | 203 | 0.28 |
| Ring gear | M50NiL | 7859 | 203 | 0.28 |

In ADAMS, there are two methods available for calculating gear contact forces, namely the regression method and the Impact contact function method. The regression method is known for its lower calculation accuracy, hence the preference for using the Impact function to calculate contact forces. When employing the Impact function for contact force calculation, it is essential to consider that during the meshing process, the meshing position and the number of teeth involved result in varying meshing stiffness, and the friction coefficient also varies with different rotational speeds. Therefore, it is imperative to set the contact parameters based on the actual model.

In ADAMS, the constraint relationship between gears is typically described by mutual constraints of contact collision force (normal) and friction force (tangential). Stiffness, measured in N/mm, is an important parameter for describing the contact surface[3]. The stiffness value K for the mutual contact collision of gears based on the Impact function is related to the contact surface shape and the materials of the meshing pair. Equation (1) represents the method for calculating the stiffness of a pair of meshing gears, (2) calculates R at the meshing pair, and (3) calculates E* at the meshing pair. Other contact parameters need to be set and adjusted according to the actual operating conditions. The equations are as follows:

$$K = \frac{4}{3} \sqrt{RE^*}, \quad (1)$$

$$R = \frac{R_1 R_2}{R_1 + R_2}, \quad (2)$$

$$E^* = \frac{E_1 E_2}{E_2 (1 - \mu_1^2) + E_1 (1 - \mu_2^2)}, \quad (3)$$

where R_1 and R_2 represent the equivalent radii at the gear meshing contact, approximated by the dedendum radius; E_1 and E_2 denote the elastic moduli of the gears; μ_1 and μ_2 are the Poisson's ratios of the gears[4]. Based on the actual motion of the model and the constraint characteristics of ADAMS, the multi-rigid body virtual prototype model of the compound planetary gear transmission system is obtained as shown in Fig. 4.



Fig. 4 Multi rigid body virtual prototype model of planetary gearbox

3.2 Drive and Load Setting

In ADAMS, the drive setting is primarily achieved by adding drive functions. The input shaft drives the system rotation, and during simulation, the rotational speed is applied to the input shaft. In accordance with actual operating conditions, the rotational speed is set to 48000°/s (8000r/m), and the output load is set to 50413.92N·m. These settings are primarily aimed at studying the gear transmission characteristics of the GTF under actual operating conditions.

The output curves of the solution results are closely related to the solver used. To meet the requirements of the solution results, the GSTIFF-SI2 solver is employed, and the simulation time is set to 0.5s with 10000 solution steps. Upon verification, the simulation results are found to satisfy the analysis requirements[5].

3.3 Theoretical calculation of component angular velocities in gear reducers

Based on the structural parameters of the planetary gear system, with a sun gear tooth count of 48, a planet gear tooth count of 37, and a ring gear tooth count of 122, the internal meshing transmission ratio of the system is approximately calculated using the transmission ratio formula to be:

$$i_1 = \frac{Z_3}{Z_2} = \frac{122}{37} = 3.30, \quad (4)$$

The external meshing transmission ratio is approximate:

$$i_2 = \frac{Z_1}{Z_2} = \frac{48}{37} = 1.30, \quad (5)$$

After the meshing of the gears, the transmission ratio of the system approximates:

$$i = \frac{Z_3}{Z_1} = \frac{122}{48} = 2.54, \quad (6)$$

where Z_1 , Z_2 , Z_3 represent respectively the number of teeth on the sun gear, planet gear, and internal ring gear. After calculation, the theoretical speeds of the gear system components are as shown in Table 3.

Table 3. Comparison of Simulation Results with Theoretical Analysis

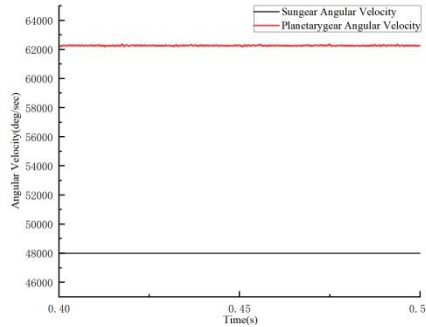
| | Rotational speed (rpm) | Angular velocity (°/s) |
|----------------|------------------------|------------------------|
| Sun gear | 8000 | 48000 |
| Planetary gear | 10378.36 | 62270.27 |
| Ring gear | 3149.61 | 18885.23 |
| Input shaft | 8000 | 48000 |
| Output shaft | 3146.61 | 62270.27 |

3.4 The simulated angular velocity results in ADAMS

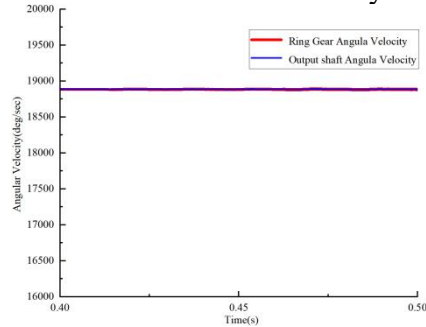
Using the ADAMS post-processing software to extract the speed curves of the input and output shafts for the sun gear, planet gear, and ring gear, as shown in Fig. 5.

Comparing the extracted average simulation curve parameters for each mechanism with the theoretical calculated values, as shown in Table 4.

By observing the simulated angular velocity curves of the gear transmission system, the angular velocity of the sun gear shows no fluctuation or error compared to the input shaft. This is because the driving speed is applied to the input shaft, and the input shaft is fixedly connected to the sun gear. The instantaneous driving of the active wheel causes brief fluctuations in the planet gear and ring gear during the startup process, which quickly recover and maintain stable speed. After the speeds of all components reach a stable state, the angular velocity curves of the planet gear and internal gear exhibit slight periodic fluctuations, mainly due to vibration impact generated by gear contact. The adoption of helical gears ensures smoother meshing during the transmission process, ensuring that the comprehensive meshing stiffness of the gear teeth remains approximately continuous during changes. Under normal operating conditions, the stable angular velocities of the individual components in the planetary gear system, as well as the overall transmission ratio, align with the theoretical calculation values.



(a) Simulation Results of Sun Gear and Planetary Gear Angular Velocity



(b) Simulation Results of Output Shaft and Ring Gear Angular Velocity

Fig. 5 Simulation Results of Gear Speed of Each Gear System

Table 4. Comparison of Simulation Results with Theoretical Analysis

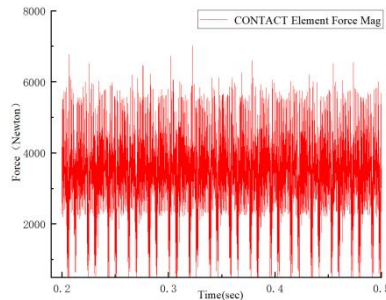
| | Theoretical Value | Simulated Value | Error |
|--------------------|-------------------|-----------------|---------|
| Input shaft | 48000 | 48000 | 0 |
| Sun gear | 48000 | 48000 | 0 |
| Planetary gear | 62270.27 | 62270.35 | 0.0005% |
| Ring gear | 18885.23 | 18885.32 | 0.001% |
| Output shaft | 18885.23 | 18885.32 | 0.001% |
| Transmission Ratio | 2.54 | 2.54 | 0 |

3.5 ADAMS contact force simulation results

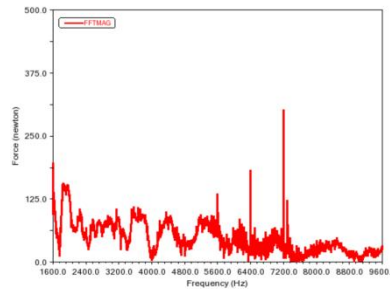
During the meshing process of gears, the meshing force curve between gears can effectively reflect the stability of gear transmission, the magnitude of meshing force, and the meshing frequency of the gears. In the GTF gear transmission mechanism, the number of teeth on the planet gear is denoted as Z_2 . According to gear transmission dynamics, the meshing frequency between the

planet gear and the internal gear can be calculated as: $f = Z_2 \cdot n_0 / 60 = 6400 \text{Hz}$, where Z_2 represents the number of teeth on the planet gear, and n_0 represents the theoretical speed. Due to the constant speed applied to the driving gear, there is an instantaneous change in the driving force of the planet gear during the initial meshing, which is reflected as a significant peak in the meshing force. For ease of observation, we selected the period of 0.2-0.5 seconds during which the gears operated smoothly to observe the meshing force, as shown in Fig. 6(a). It can be observed that there is a significant periodic variation in the peak values of the meshing force, mainly due to the periodic changes in the meshing stiffness of the gears during the meshing process, leading to impact vibrations during the meshing of the gear teeth, thereby reflecting the dynamic characteristics of the gear transmission process.

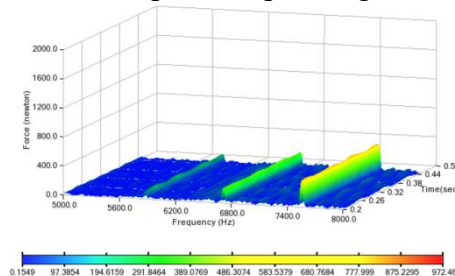
After performing FFT transformation on Fig. 6(a), the vibration spectrum of the gear meshing curve can be obtained, as shown in Fig. 6(b), along with its three-dimensional frequency spectrum, as shown in Fig. 6(c). Through the gear meshing curve vibration spectrum, the gear resonance frequencies are mainly distributed around 5344Hz and its harmonics. The simulation results align with the theoretically calculated meshing frequency. Fig. 6 illustrates the changing pattern of the meshing spectrum over time during the simulation, indicating that under normal operating conditions, the meshing spectrum of the planetary gear system remains stable during operation and does not shift over time.



(a) Time-domain variation curve of the contact force between the sun gear and the ring gear



(b) Two-dimensional spectrum plot of gear mesh vibration



(c) Three-dimensional spectrum plot of gear mesh vibration

Fig. 6 Gear Meshing Vibration Curves

4. Modal and Harmonic Response Analysis Based on Ansys

Performing modal and harmonic response analysis on the planetary gear system using ANSYS can provide crucial information about the system's vibration characteristics, stability, and reliability. This analysis can contribute to optimizing designs, reducing vibration noise, improving operational

smoothness of the system, addressing vibration issues, and enhancing the performance and reliability of the planetary gear system[6].

4.1 Model pre-processing Settings

Convert the 3D model of the planetary gear system to the Step format, and then import it into the WORKBENCH software. Subsequently, modify the material properties of the system's components according to the actual material attributes. The materials and their parameters for each part of the system are listed in Table 2.

Due to the complexity of the model structure, tetrahedral meshing was chosen for its high adaptability. The overall mesh quality exceeded 0.75, enabling subsequent modal and harmonic response analysis. The meshing results are illustrated in Fig. 7.

Friction contact is set between the sun wheel and the planet wheel, the planet wheel and the inner gear ring, and the sun wheel spline and the input shaft spline, and the friction coefficient is 0.15; Then set the friction contact between each planetary wheel and the planetary frame, and the coefficient is still 0.15; In order to facilitate the calculation, reduce the number of grid units, and avoid the phenomenon of over-constraint affecting the computing speed, the bolts and end caps used to connect the two ring gear rings and the output shaft are suppressed, and the binding constraint is used to replace the bolt connection effect.

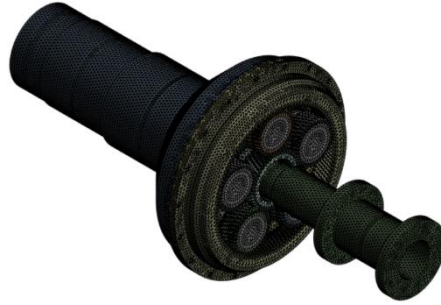


Fig. 7 Meshing of the finite element model

4.2 Modal analysis theory

Modal analysis is the most fundamental dynamic analysis. Through modal analysis, we can understand the vibration characteristics of the planetary gear transmission system at different frequencies, including natural frequencies and their corresponding mode shapes[7].

For a general structure, the dynamic equation can be expressed as:

$$[M]\{\ddot{x}\} + [C]\{\dot{x}\} + [K]x = Ft, \quad (7)$$

where $[C]$ is the mass matrix, $[C]$ is the damping matrix, and $[K]$ is the stiffness matrix. For modal analysis, $\{F\}=0$ and $[C]=0$.

Therefore, the motion equation for modal analysis can be expressed as:

$$[M]\{\ddot{x}\} + [K]\{x\} = \{0\} \quad (8)$$

According to vibration dynamics, the free vibration of the structure is simple harmonic motion, and its displacement x can be represented by (9):

$$x = X \cos(\omega t - \varphi) \quad (9)$$

Substituting (9) into (10) yields:

$$[M] + \omega^2 [K]\{x\} = \{0\} \quad (10)$$

By solving (9), the natural angular frequency ω_i can be obtained, and the natural frequency is given by $f=\omega_i/2\pi$. Each ω_i corresponds to a vector $\{x\}_i$ representing the mode shape corresponding to the natural frequency $f=\omega_i/2\pi$ [8].

4.3 Modal analysis results

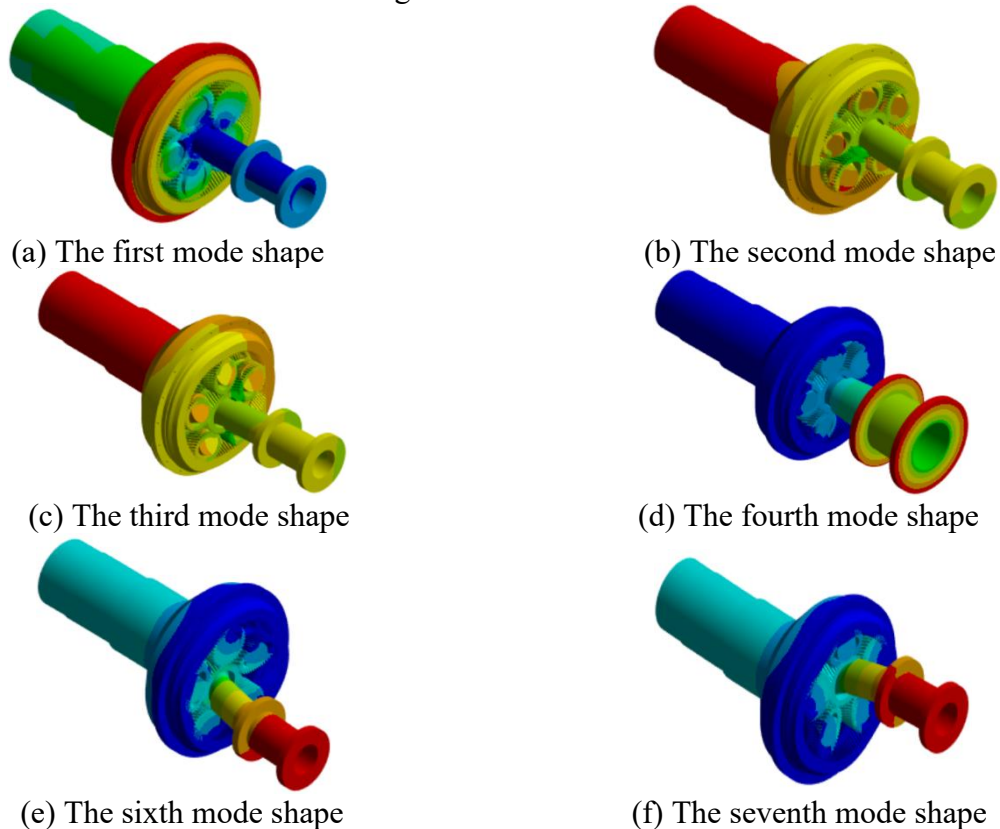
The rationality of the boundary condition constraints in the finite element model directly affects the accuracy of the modal analysis solution. A comprehensive analysis of the actual installation method of the planetary gear transmission system in the GTF engine reveals that the gear system's

carrier only serves to position the planet gears and does not participate in the dynamic analysis. Therefore, it is constrained with fixed constraints. In this modal analysis, the focus is on studying the radial vibration characteristics of the planetary gear transmission system. Therefore, the axial displacement of the main components is restricted to zero.

This approach to setting boundary conditions for the modal analysis considers the specific behavior and interaction of the components within the planetary gear system, which is crucial for obtaining accurate and meaningful modal analysis results.

In the modal analysis, Block Lanczos method in ANSYS is used to extract the natural frequency and vibration mode of GTF gear transmission system. In ANSYS, Block Lanczos method is usually used to extract the modal analysis results of GTF gear system to obtain its natural frequency and vibration mode[9]. In dynamic analysis, it is generally the low-order modes that have a greater impact on the vibration system, and 5-10 orders are usually sufficient for analysis. Due to the particularity and complexity of the system, the first 12 modes are mainly extracted for analysis. The specific analysis results of the first 12 natural frequencies and modes are shown in Fig. 8 and Table 5.

According to the analysis of the first 12 modal shapes and natural frequencies of planetary gear transmission system presented in Table 5, after limiting the Z-axis free degree of motion of the system, the minimum frequency is 268.53 Hz, and the main modal shape is the deformation of the epicyclic circle, which is one of the main modal shapes of wheel-type structures. This reflects the flexibility of the planetary gear system. Subsequently, as the natural frequencies increase, higher-order epicyclic circle deformation and eccentricity deformation appear one after another. When the input and output shafts bend, the vibration deformation of the planetary gear system is obviously smaller, indicating that the natural frequency of the planetary gear system is higher than that of the input and output shafts. This is mainly related to the geometry and material stiffness of the planetary gear system. In addition, the bending deformation occurs near the end of the shaft, with compression, twist, and other deformations. When designing similar shaft systems, it is advisable to increase the stiffness and strength near the end of the shaft.



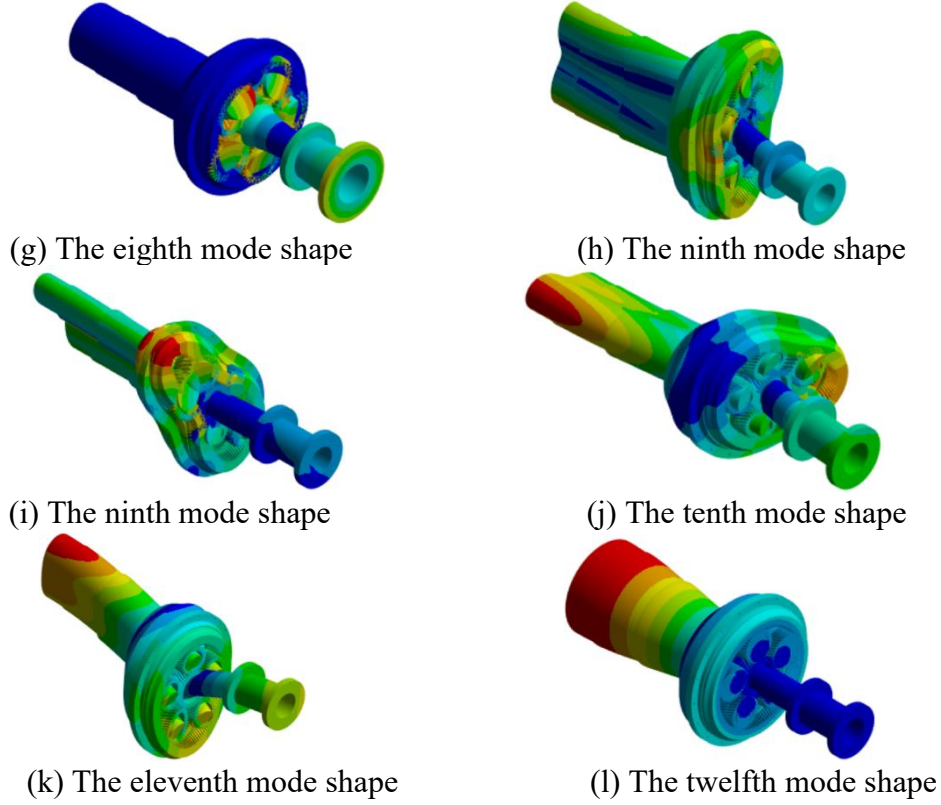


Fig. 8 Results of First 12 Order Modal Analysis of Fan Rotor System

Table 5. Natura Frequencies and Modes Descriptions of First12 Order Modes

| Mode order | Natural frequency (Hz) | Mode shape description |
|------------|------------------------|--|
| 1 | 268.53 | First order circumferential deformation |
| 2 | 346.24 | First order bending along the Z-axis |
| 3 | 359.04 | First order bending along the Y-axis |
| 4 | 486. | Second order circumferential deformation |
| 5 | 690.17 | Second order bending along the Z-axis |
| 6 | 697.41 | Second order bending along the Y-axis |
| 7 | 1079. | Third order circumferential deformation |
| 8 | 1108.3 | First order radial deformation |
| 9 | 1113.3 | First order radial deformation |
| 10 | 1118.1 | Third order bending along the Z-axis |
| 11 | 1121.6 | Third order bending along the Y-axis |
| 12 | 1181.9 | Fourth order circumferential deformation |

4.4 Harmonic Response Analysis Theory

Harmonic response analysis is based on modal analysis. Through harmonic response analysis, the steady-state response laws of components and mechanisms under sinusoidal (harmonic) loads can be determined, reflecting the structural components' motion characteristics under the action of different frequency harmonic loads. During operation, the GTF engine is subjected to exciting forces from gear meshing, transmitted through the input shaft from the sun gear, and from other components due to rotor dynamic balancing. When the exciting forces and the frequency of the planetary gear system are close or identical, resonance can occur. This can cause a sudden increase in vibration and noise in the gear system, affecting the service life of critical parts and components, and impacting the smooth operation of the aircraft. In order to avoid resonance, it is necessary to perform harmonic response analysis on the planetary gear system, providing a basis for optimizing the performance of the GTF engine.

The differential equation representing the forced vibration of a structure under harmonic load can be expressed as:

$$[M]\{\ddot{x}\} + [C]\{\dot{x}\} + [K]\{x\} = [F(t \sin(\theta t))], \quad (11)$$

where $\{F\}$ is the vector of the amplitude of the harmonic load and θ is the frequency of the exciting force.

$$x = \{A\} \sin(\theta t - \varphi), \quad (12)$$

where $\{A\}$ is the vector of displacement amplitude, and φ is the phase angle between displacement response and the excitation load. Substituting (12) into (11) yields the relationship between displacement and frequency, as shown in (13):

$$\{A\} = \frac{\{F\} \sin(\theta t)}{-\{M\} \theta^2 \sin(\theta t + \varphi) + [C] \theta \cos(\theta t + \varphi) + [K] \sin(\theta t + \varphi)} \quad (13)$$

After setting the frequency range and interval in the finite element software, the peak frequencies of the curve can be obtained by solving the results.

4.5 The results of the harmonic response analysis

The modal analysis results indicate that the natural frequency range of the planetary gear system is 265 to 1125 Hz. Therefore, the frequency range defined for the harmonic response analysis is 265 to 1125 Hz, with an analysis interval of 4 Hz. A dynamic balancing load is applied to the entire system at the center of gravity, and the boundary constraints are set consistent with the modal analysis. After solving, the frequency spectrum of the components is extracted.

The planetary gear system is a critical structural component of the GTF aircraft engine, and its structural performance can affect the engine's performance, thereby impacting aircraft operation and stability.

Before applying the dynamic balancing load, it is necessary to evaluate the balance state of the rotor. Currently, there are two methods for evaluating the balance state: evaluating the permissible residual unbalance and assessing the unbalanced vibration state of the entire machine. This article uses the permissible residual unbalance evaluation method as the criterion.

According to dynamic balancing standards, there are three indicators to evaluate the quality of dynamic balancing, and their relationships are as follows[10]:

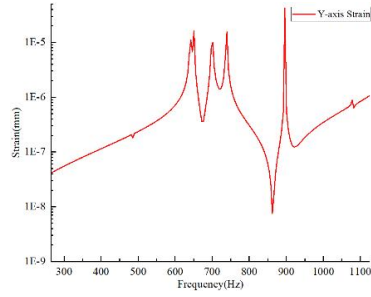
$$G = \frac{e_{per} \times \omega}{1000} = \frac{U_{per} \times \omega}{1000m}, \quad (14)$$

where G (mm/s) is the balance quality grade, U_{per} (g·mm) is the allowable residual unbalance, and e_{per} (g·mm/s) is the allowable residual unbalance per unit speed. According to the steady (rigid) rotor balance quality grade guidelines, the planetary gear transmission system of the GTF engine should adopt the gear rotor balance quality grade, hence $G=6.3$ mm/s. After calculation, the system's $U_{per}=2994.5$ g·mm.

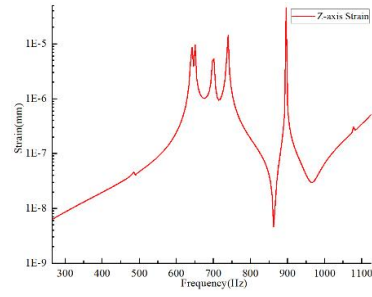
Afterwards, the results of the harmonic response analysis were extracted, and the stress and strain frequency spectrum diagrams of the planetary gear transmission system in the Y-axis and Z-axis radial directions were obtained, as shown in Fig. 9.

Analysis of Fig. 9 reveals that the stress and strain amplitudes predominantly occur near 650.9Hz, 739.8Hz, with the main peak values concentrated around 265Hz and 1125Hz. These frequencies closely align with the natural frequencies of the 2nd and 3rd modes identified in the modal analysis. This observation indicates that the modal and harmonic response analysis methods used in this study are reliable.

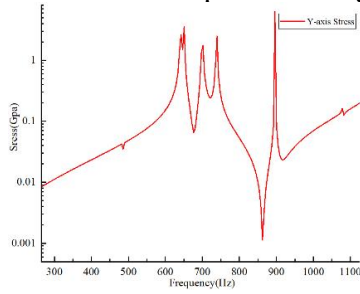
Further analysis shows that, whether in strain or stress frequency spectrum diagrams, the maximum amplitudes for the planetary gear transmission system occur in the Z-axis direction, with the maximum amplitude occurring at 650.9Hz. Therefore, when the system is in operation, the rotational speed should be avoided as much as possible around 650.9Hz[11].



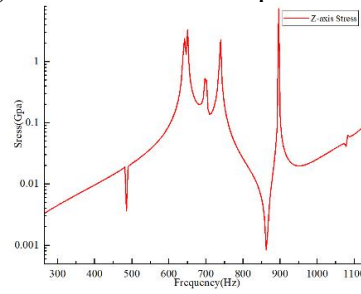
(a) Y direction strain spectrum diagram



(b) Z direction strain spectrum diagram



(c) Y direction stress spectrum diagram



(d) Z direction stress spectrum diagram

Fig. 9 Strain and stress spectra in Y and Z directions

5. Summary

This study established a three-dimensional model of the planetary gear system based on the X-type GTF engine, and conducted dynamic analysis using ADAMS, modal analysis and harmonic response analysis using ANSYS. The following conclusions were drawn from the post-processing and cloud map analysis:

ADAMS simulation indicated stable operation of the various components of the planetary gear system under normal operating conditions, with angular velocity and gear ratio matching the theoretical calculations, validating the accuracy of the simulation model. Analysis of the gear meshing spectra was conducted by examining the variation of contact forces in the gear mesh, revealing that the fluctuations were caused by periodic changes in gear mesh stiffness over time.

Modal analysis results showed that the finite element model of the planetary gear system has a minimum natural frequency of 353.97Hz. At higher frequencies, oscillations and torsional deformation were observed at the ends of the shafts and the gear ring. To enhance the natural frequency of the system, it is recommended to strengthen the output shaft, input shaft, and gear ring.

Harmonic response analysis indicated that within the range of 265 to 1125Hz and under the influence of dynamic balancing loads, resonance frequencies primarily occurred near the 2nd and 3rd natural frequencies of the planetary gear system. Significant stress and strain amplitudes were observed near these frequencies, highlighting the need to avoid operating the system around 650.9Hz and 739.8Hz, as well as under higher frequency loads, to prevent resonance during practical operation.

Due to the heavy-duty nature of the GTF engine, the planetary gear system requires higher stiffness and strength to fulfill operational requirements. The system's higher natural frequencies necessitate significant excitation for resonance to occur. Furthermore, various components of the system provide fixation and isolation. Therefore, avoiding sensitive frequencies during operation can prevent gear drive resonance. The research findings can provide a theoretical basis for the design of subsequent GTF engines of the same type, and offer guidance for further optimization of the planetary gear system.

Acknowledgements

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