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Retraction

Retracted: Modal Analysis of Aeronautic Spiral Bevel Gear in the Temperature Field

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This article has been retracted by Hindawi following an investigation undertaken by the publisher [1]. This investigation has uncovered evidence of one or more of the following indicators of systematic manipulation of the publication process:

(1) Discrepancies in scope
(2) Discrepancies in the description of the research reported
(3) Discrepancies between the availability of data and the research described
(4) Inappropriate citations
(5) Incoherent, meaningless and/or irrelevant content included in the article
(6) Manipulated or compromised peer review

The presence of these indicators undermines our confidence in the integrity of the article's content and we cannot, therefore, vouch for its reliability. Please note that this notice is intended solely to alert readers that the content of this article is unreliable. We have not investigated whether authors were aware of or involved in the systematic manipulation of the publication process.

Wiley and Hindawi regrets that the usual quality checks did not identify these issues before publication and have since put additional measures in place to safeguard research integrity.

We wish to credit our own Research Integrity and Research Publishing teams and anonymous and named external researchers and research integrity experts for contributing to this investigation.

The corresponding author, as the representative of all authors, has been given the opportunity to register their agreement or disagreement to this retraction. We have kept a record of any response received.

References

[1] W. Yu, S. Liu, W. Xu, and D. Wang, "Modal Analysis of Aeronautic Spiral Bevel Gear in the Temperature Field," *Journal of Sensors*, vol. 2022, Article ID 1707808, 13 pages, 2022.

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Research Article

Modal Analysis of Aeronautic Spiral Bevel Gear in the Temperature Field

Wentao Yu,[1,2] **Shijun Liu**,[1] **Wenbo Xu**,[1] **and Dongfei Wang**[1]

[1]*Zhengzhou Research Institute of Mechanical Engineering Co. Ltd., Zhengzhou 450052, China*
[2]*School of Mechanical and Electronic Engineering, Zhongyuan University of Technology, Zhengzhou 450007, China*

Correspondence should be addressed to Wentao Yu; ywt@zut.edu.cn

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The temperature of the bevel gear tooth surface will increase greatly because of its special structure. Modal is the characteristic of reaction structure. In order to design and manufacture higher quality aerial bevel gears, it is necessary to study the modal of aerial bevel gears under the action of the temperature field. Firstly, a temperature field calculation model is established according to heat transfer theory. Then, based on the traditional linear modal theory, the modal analysis model under the influence of temperature was established considering the influence of rotational speed and temperature field on the stiffness. Based on the model, modal analysis of the driven gear of a certain aviation bevel gear is carried out. It can be seen from the analysis results that the speed has a great influence on the gear stiffness, and the positive and negative effects are related to the direction of the centrifugal force. Secondly, the inhomogeneity of temperature field distribution also affects the natural frequency and then the gear stiffness. When the speed field and temperature field work together, the pitch shape changes the most, so it is necessary to consider temperature as an influencing factor in the design of aerial bevel gears.

1. Introduction

Bevel gears are widely used in the helicopter transmission system and aeroengine for main deceleration and change of transmission direction. Compared with other industries, bevel gears should have high bearing capacity, small volume mass, high reliability, and smooth operation at high speed. Bevel gears are the indispensable key parts of helicopter transmission and engine, and their dynamic performance will directly affect the working condition of a helicopter. The aviation bevel gear has the characteristics of high overload, and due to the restrictions of the main reducer or aviation engine, its normal work under the condition of lubrication situation is relatively poor, often in the lack of oil lubrication state, so in the work produced by friction and heat flux, larger heat transfer to the environment and can be collected and cause tooth surface temperature increasing significantly. Then, there will be gluing and other gear failures. Structural modal is often used to reflect the physical characteristics of the structure, and the modal analysis of gear is through the analysis of the inherent characteristics of gear (frequency, damping, and modal). Studying the structural characteristics of the gear process purpose is to identify the modal parameters of the gear pair, for gear dynamic vibration characteristic analysis, and provides the reference for the optimization of structural dynamic characteristics. Therefore, it is necessary to study the modal changes of aerial bevel gears under the action of the temperature field.

At present, the modal analysis of gear mainly includes gear and gearbox, among which the modal analysis of gear parts is mainly involute gear such as spur gear or helical gear, and the modal analysis of bevel gear is less literature. Ben and Lin [1] performed a dynamic modal study of the vehicle's rear axle gears and claimed that high-speed gears have a "centrifugal rigidization effect" owing to centrifugal motion. Based on the resonance theory, Duo [2] constructed a modal and performed resonant analysis on spiral bevel gears in high-speed and heavy-duty conditions. Renyan [3] does a modal analysis of a spiral bevel gear, obtaining the inherent frequency and vibration type of the gears and verifying the resonance of the high-speed gear system. Haiguo and Bin [4] used APDL to create a finite element model of the whole structure of the spiral bevel gear. The model estimated the inherent characteristics of 5 gears with varying abdominal plate thicknesses, and the low-order vibration type is summarized on this basis. In order to avoid damage to gear pump parts caused by resonance, Guo and Haiming [5] used ANSYS to analyze its modal analysis and made statistics and analysis of the first 10 order modal of the used parts. Kaihong et al. [6] applied ABAQUS to analyze the modal forms and natural frequencies of elliptical gears according to the elastic theory and verified the analysis results by comparison in experiments. Li [7] used the ANSYS asymmetric gear to conduct modal analysis to verify the correctness of his derivation of the transient stiffness of asymmetric gear.

In terms of temperature field research of bevel gears, Handschuh and Kicher [8] first proposed a three-dimensional modeling method to analyze the thermal behavior of spiral bevel gears. Yunwen et al. [9] proposed a mathematical model for calculating the body temperature of spiral bevel gears based on heat transfer theory. Zhang et al. [10] studied the tooth surface temperature rise of spiral bevel gears based on the elastohydrodynamic theory and improved the thermal analysis method on this basis. Zongzheng Wang et al. [11] studied the temperature field of different contact traces based on elastohydrodynamic theory. Based on heat transfer and tribology theories, Fei et al. [12] analyzed the transient temperature field of the spiral bevel gear in the helicopter transmission system under the condition of lack of oil and obtained the calculation method of the transient temperature field under this condition. Shi et al. [13] used the finite element method to establish the temperature field model of the gear body and analyzed the steady temperature distribution law of gear teeth after obtaining the steady temperature field of locomotive traction gear. Li and Pang [14] carried out various load tests on the gear system with split teeth through finite element simulation and found the variation rule of the temperature field in the tooth domain around the split teeth under different load conditions. Li and Tian [15] studied the three-dimensional analysis and temperature sensitivity analysis of the unsteady temperature field of gear transmission, in which the frictional heat causes the rise of surface temperature and reduces its bearing capacity and antiglue capacity. Xue and Xu [16] use the finite element simulation method to calculate and analyze the influence of different oiling and large spiral bevel gear steering on the temperature field distribution of the gearbox for the cross monorail train.

Currently, Yuning et al. [17] employ the Monte Carlo approach to investigate the effect of material parameter changes produced by temperature field changes on the intrinsic frequency, despite the fact that the temperature field has less influence on the modal state. In this regard, the investigation of geometrically more complicated curved spiral bevel gears is rarely described.

In this paper, the temperature field distribution model is firstly established according to the energy equation. Based on the linear modal theory, combining the temperature field distribution model with the stress, the modal equation of the aviation bevel gear under the action of the temperature field is established. Then, on this basis, the numerical calculation method is used to solve the natural characteristics of the driven gear of an aviation bevel gear. According to the numerical calculation results, the variation of the mode of the aviation bevel gear under the single or joint action of the speed field and temperature field is obtained, and its influence is analyzed.

1.1. Temperature Field Distribution Model. Since the change of temperature will produce additional stress stiffness, the temperature field distribution of the gear tooth should be determined first before the modal analysis. The working temperature of gear is mainly composed of two parts: the body temperature of gear and the instantaneous temperature (or flash temperature) of the meshing tooth surface. According to Reference [18], body temperature is the main influencing factor of gear thermal deformation [18]. Under the condition of constant speed and load, although the instantaneous temperature of the meshing tooth surface changes in the process of gear rotation, the influence range is small and the duration is short. Therefore, it is generally assumed that the temperature of each point on the gear does not change with time; that is, the temperature field of the gear tooth body is treated as a stable temperature field. At present, the finite difference method and finite element method are commonly used to study the temperature field of the gear body. For the tooth surface model with a complex shape, the stability of the finite difference method is poor [19]. Therefore, the finite element method is used to analyze the temperature field in this paper.

1.2. Temperature Field Calculation Model. According to the law of energy conservation, the heat transfer differential equation of the three-dimensional transient temperature field is obtained:

$$\frac{\partial T}{\partial t} = \frac{\kappa_g}{\rho_g C_g}\left(\frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2} + \frac{\partial^2 T}{\partial x^2} + \frac{q_0}{\kappa_g}\right), \tag{1}$$

where T is the body temperature of gear teeth, κ_g is the thermal conductivity of the material, ρ_g is the material density, C_g is the specific heat capacity of material, and q_0 is the intensity of the heat source in the material.

First, it is assumed that there is no internal heat source in the steady-state temperature field, that is, $q_0 = 0$. Secondly, in the steady-state condition, the temperature field no longer changes with time T, $\partial T/\partial t = 0$. Therefore, the heat transfer differential equation of the three-dimensional steady temperature field is

$$\frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2} + \frac{\partial^2 T}{\partial x^2} = 0. \tag{2}$$

1.3. The Boundary Condition of the Steady-State Temperature Field Thermal Analysis Boundary. To solve Equation (10), additional boundary conditions and initial conditions are required. There are three types of common boundary conditions [18]: The first type is the temperature or temperature function of the known boundary. The second boundary condition is the heat flow on the boundary of the known object. The third type of boundary condition is the known temperature and convective heat transfer coefficient of the medium in contact with the solid.

The temperature field boundary condition of spiral bevel gear is the following:

(1) *Meshing Tooth Surface*. There is a heat source generated by meshing friction on the meshing tooth surface. Therefore, it is the third type of boundary condition with a surface heat source:

$$-\kappa_g \frac{\partial T}{\partial n} = \alpha_1 (T - T_0) - Q \tag{3}$$

(2) *Top, Root, and Nonengaged Surface*. For the third type of boundary conditions:

$$-\kappa_g \frac{\partial T}{\partial n} = \alpha_2 (T - T_0) \tag{4}$$

(3) *Gear Tooth Face*. For the third type of boundary conditions:

$$-\kappa_g \frac{\partial T}{\partial n} = \alpha_3 (T - T_0) \tag{5}$$

In Equations (3)–(5), n is the external normal direction of the meshing surface, T_0 is the working temperature of the medium, Q is the average frictional heat flow, α_1 is the convective heat transfer coefficient of the meshing surface, α_2 is the convective heat transfer coefficient of the tooth tip, tooth root, and nonmeshing surface, and α_3 is the convective heat transfer coefficient of gear teeth. The temperature of the other surfaces varies little with the normal direction and can be regarded as $\partial T / \partial n = 0$.

1.4. The Amount of Heat Flow That Engages the Flank. There are three main types of friction between meshing tooth surfaces: sliding friction, rolling friction, and internal friction caused by metal elastic-plastic deformation. At high speed, due to geometric reasons, the tooth surface of spiral bevel gears has a relatively large sliding speed. Therefore, rolling friction and internal friction are ignored, and only sliding friction is considered. The influence factors of friction heat produced by sliding friction are related to the normal pressure and friction coefficient besides the relative sliding velocity. The frictional heat flow per unit time and per unit area due to the relative sliding of the tooth surface is [20]

$$q_c = \eta p_{nc} \mu v_s, \tag{6}$$

where q_c is the heat flow of meshing tooth surface; η is the thermal conversion coefficient; p_{nc} is the normal contact pressure, generally Hertz stress; μ is the coefficient of friction; and v_s is the relative sliding speed.

Frictional heat is assigned to the two flanks of the active gear and the passive gear, so that the heat distribution coefficient of the active gear, then β is the frictional heat flow of the active gear and the passive gear q_{1c}, which is q_{2c} [20]:

$$\begin{cases} q_{1c} = \beta q_c, \\ q_{2c} = (1 - \beta) q_c. \end{cases} \tag{7}$$

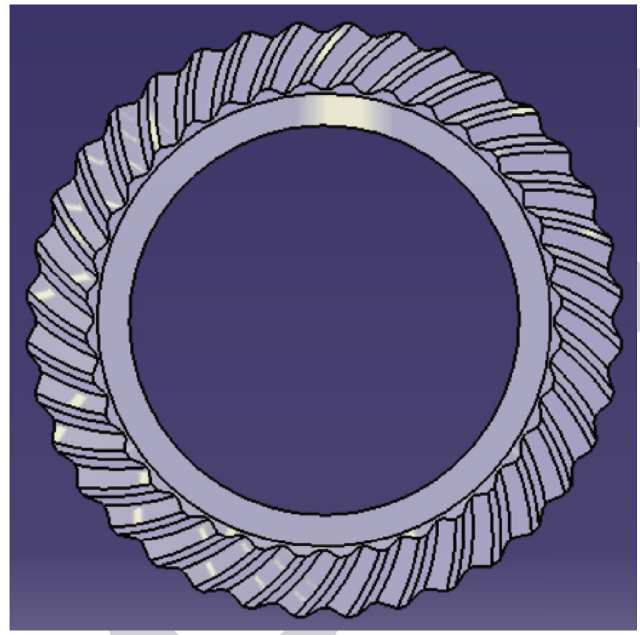


Figure 1: 3D model of spiral bevel gear.

Table 1: Convective heat transfer coefficient of each part of gear teeth [8].

The surface position	Convection heat transfer coefficient (W/(m^2*°C))
Meshing nonworking face	4450
Tooth top and alveolar surface	50
The small end face	50
Large end face	160
The other side	160
Meshing nonworking face	0

Table 2: Thermal analysis of different mesh densities.

Grid density	Maximum temperature (°C)	The number of grids
2	281.7	187322
3	281.06	65454
4	281.22	39570
5	281.54	24760

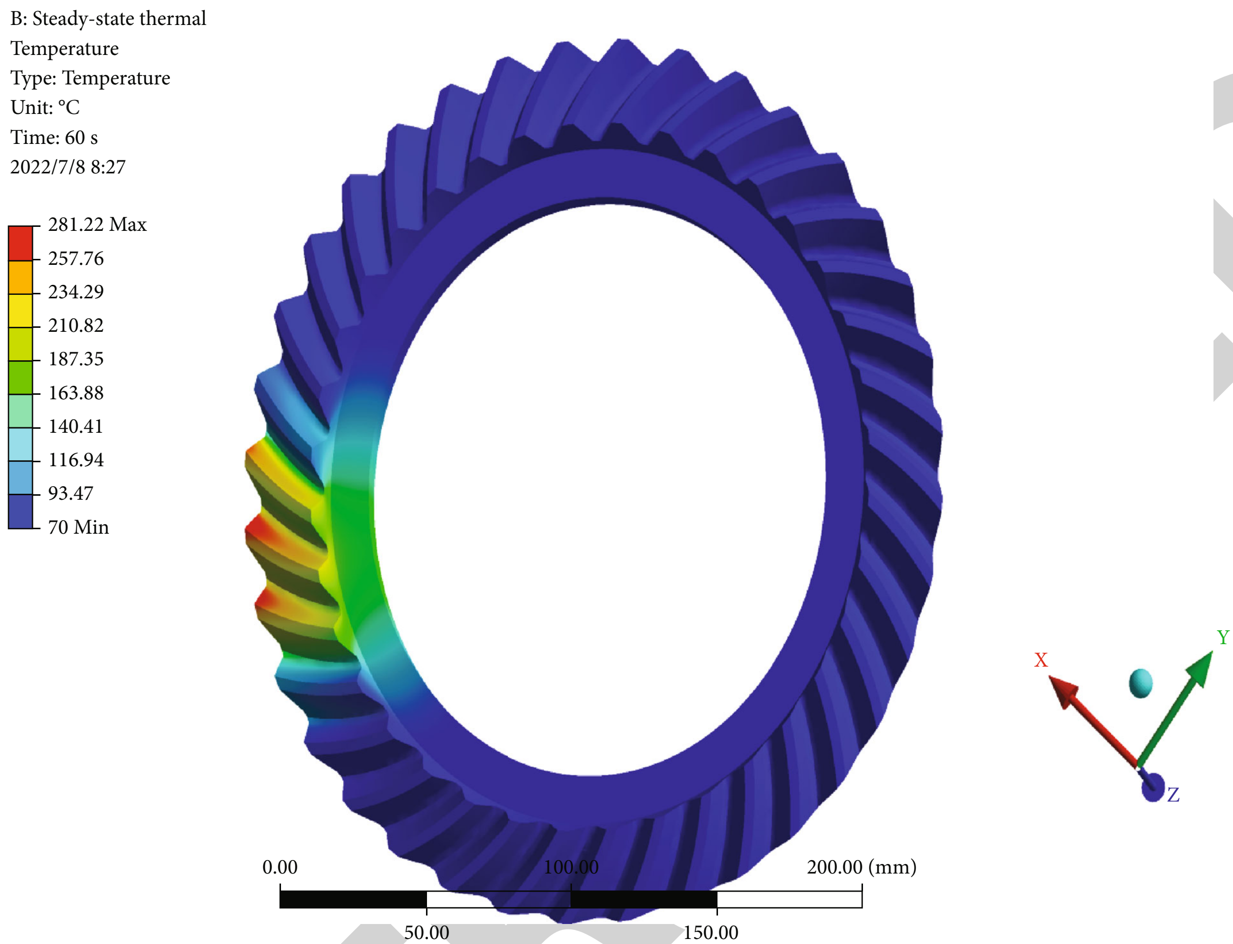


Figure 2: Temperature field distribution calculated by steady-state thermal analysis.

Because the material of the active gear and the passive gear is the same, in order to solve this paper differently from the numerical, $\beta = 0.5$.

During the whole transmission process, the heat flow at the meshing point will keep circulating input with its rotation [21]. Therefore, in order to facilitate the calculation and take the average value, the driving gear and driven gear rotate for 1 cycle, and the average friction heat flow is

$$\begin{cases} q_1 = \dfrac{b}{v_{t1}} \dfrac{n_1}{60} q_{1c}, \\ q_2 = \dfrac{b}{v_{t2}} \dfrac{n_2}{60} q_{2c}, \end{cases} \tag{8}$$

where b is the contact width; v_{t1}, v_{t2} are the tangential speed of the pinion and wheel gear; and $n_1 n_2$ are the rotary speed of the pinion and wheel gear.

1.5. Calculate the Convection Heat Exchange Coefficient. According to Section 1.2, the accuracy of the convective heat transfer coefficient directly affects the calculation accuracy of the temperature field. Because of the complex geometry of spiral bevel gears and the different cooling and lubrication methods, the heat transfer coefficients of each part are quite different. For the convenience of numerical calculation, they are divided into three categories: heat transfer coefficient of end face, heat transfer coefficient of meshing surface, and convective heat transfer coefficient of tooth tip, tooth root, and nonmeshing surface.

Compared with spur gears, the geometric shape of spiral bevel gears is more complex. Therefore, the method in Reference [21] can be adopted to simplify the gear into a rotating conical disc, and the end face is a rotating conical surface. After simplification, the convective heat transfer coefficient α_3 of the spiral bevel gear face is calculated as [21]

$$\alpha_3 = N_u \kappa_g \left(\frac{n_i \pi}{30 v \sin(\delta_a/2)} \right)^{0.5}, \tag{9}$$

where κ_g is the thermal conductivity, $n_i (i = 1, 2)$ is the pinion or wheel gear speed, v is dynamic viscosity of lubricating oil, δ_a is the spiral bevel gear surface cone angle, and N_u is the Nuschelt number, which is related to the surface fluid flow state.

When Reynolds number $\mathrm{Re} < 2 \sim 2.5 \times 10^4$, the fluid near the tooth surface is in a laminar flow state, and the Nusselt number is independent of R_e [18]:

$$N_u = 2\left(\frac{1.02P_r(m+2)}{60}\right)^{1/3}. \quad (10)$$

When $\mathrm{Re} > 2.5 \times 10^4$, the surface fluid is turbulent and the Nuschelt number is

$$N_u = 0.0197(m+2.6)^{0.2}\,\mathrm{Re}^{0.8}\mathrm{Pr}^{0.6}. \quad (11)$$

In Equations (10) and (11), m is the gear module and Pr is the Prandtl number.

The convective heat transfer coefficient α_1 of the meshing surface will change with the position of the meshing point of the tooth surface, the rotational speed, and the working environment, so it is relatively difficult to calculate the exact value, which can be estimated by the following formula [22]:

$$\alpha_1 = \sqrt{\frac{n}{120}} \cdot \sqrt{\lambda\rho_f c} \cdot \left(\frac{\nu_f H_c}{\gamma r_c}\right)^{1/4} \cdot q_{\mathrm{tot}}, \quad (12)$$

where n is rotational speed, λ is the thermal conduction coefficient, ρ_f is the density of lubricating oil or air, c is the thermal capacity, and q_{tot} is the standardized total cooling quantity; see Reference [22]. The other surface heat transfer coefficient α_2 can be approximated by $1/2 \sim 1/3$ of the surface heat transfer coefficient α_3 according to experience. The convective heat transfer coefficient of the central hole and the positioning surface is very small, which can be approximated to 0.

2. Aeronautical Spiral Bevel Gear Modal Equation with Temperature

According to the temperature distribution model established above and considering that the aerial bevel gear mainly works at high speed, based on the traditional linear modal equation, this paper introduces the effect of velocity field and temperature field on its stiffness and establishes the modal equation of the aerial bevel gear under the action of the temperature field.

2.1. Linear Modal Analysis Theory. According to the elastic mechanic finite element method, the motion differential equation of the spiral bevel gear can be obtained:

$$M\ddot{X} + C\dot{X} + KX = F, \quad (13)$$

where M, C, and K are the mass matrix, damping matrix, and stiffness matrix of the gear, respectively; $\ddot{X}$, $\dot{X}$, and X are vibration acceleration, velocity, and displacement vectors of the gear, respectively; and F is the external excitation force vector.

The modal analysis does not consider the external excitation force, that is, $F = 0$. Since the damping matrix is not easy to be diagonalized and has little influence on the natural frequency, the damping term is ignored for the convenience of numerical calculation. The undamped differential equation of motion is obtained:

$$M\ddot{X} + KX = 0. \quad (14)$$

In order to facilitate decoupling, coordinate transformation is needed to transform it into a modal coordinate system. Set $X = X_i \sin \omega_i t$ be the solution of Equation (14), substitute it into Equation (13), and get

$$\left(K - \omega_i^2 M\right)X_i = 0. \quad (15)$$

Because X_i is a non-0 vector, if formula (15) is guaranteed to have a solution, the eigenvalue equation is

$$\left|K - \omega_i^2 M\right| = 0, \quad (16)$$

where ω_i^2 is the eigenvalue and the natural frequency of order i and the corresponding X_i is the eigenmatrix and also the mode vector.

2.2. Modal Equation for Aeronautic Spiral Bevel Gear. The vibration of spiral bevel gears during rotation will change the direction of centrifugal force and lead to the instability of the gear structure. This phenomenon of stiffness matrix changes due to rotation is called rotational softening. As mentioned in Reference [1], the phenomenon of rotational softening is not obvious at low speed, but it has a great

Table 3: The first 20 order natural frequency values without heat load and without rotational speed.

The order	Natural frequency (Hz)
1	4515.4
2	4578.8
3	4581.6
4	4889.3
5	4894.9
6	5487.3
7	5495.5
8	6377.4
9	6377.5
10	6451.7
11	6462.4
12	7795.7
13	7808.7
14	9479.6
15	9494.5
16	11438
17	11454
18	12520
19	12521
20	13608

Table 4: The first 20 order natural frequency values at each speed without thermal load.

The order	The natural frequency (Hz) $n = 100$ rpm	The natural frequency (Hz) $n = 500$ rpm	The natural frequency (Hz) $n = 1000$ rpm	The natural frequency (Hz) $n = 2500$ rpm	The natural frequency (Hz) $n = 5000$ rpm
1	4515.4	4515.4	4515.4	4515.4	4511.1
2	4578.8	4573.3	4566.3	4545.6	4515.4
3	4581.6	4587.2	4594.1	4615.1	4650.3
4	4889.3	4878.2	4864.3	4823	4755
5	4894.9	4906	4920	4962.1	5033.1
6	5487.3	5471	5450.6	5389.8	5290.1
7	5495.5	5512	5532.6	5594.9	5700.4
8	6377.4	6377.3	6377.3	6325	6195.7
9	6377.5	6377.5	6377.6	6377.2	6376.9
10	6451.7	6430.4	6403.9	6377.7	6378
11	6462.4	6483.8	6510.6	6591.8	6729.4
12	7795.7	7770	7737.8	7642.3	7485.8
13	7808.7	7834.6	7867.1	7965.4	8131.9
14	9479.6	9449.8	9412.8	9302.5	9121.5
15	9494.5	9524.4	9561.9	9675.3	9867.2
16	11438	11405	11363	11240	11038
17	11454	11488	11529	11656	11869
18	12520	12520	12520	12520	12520
19	12521	12521	12521	12521	12521
20	13599	13563	13518	13384	13164

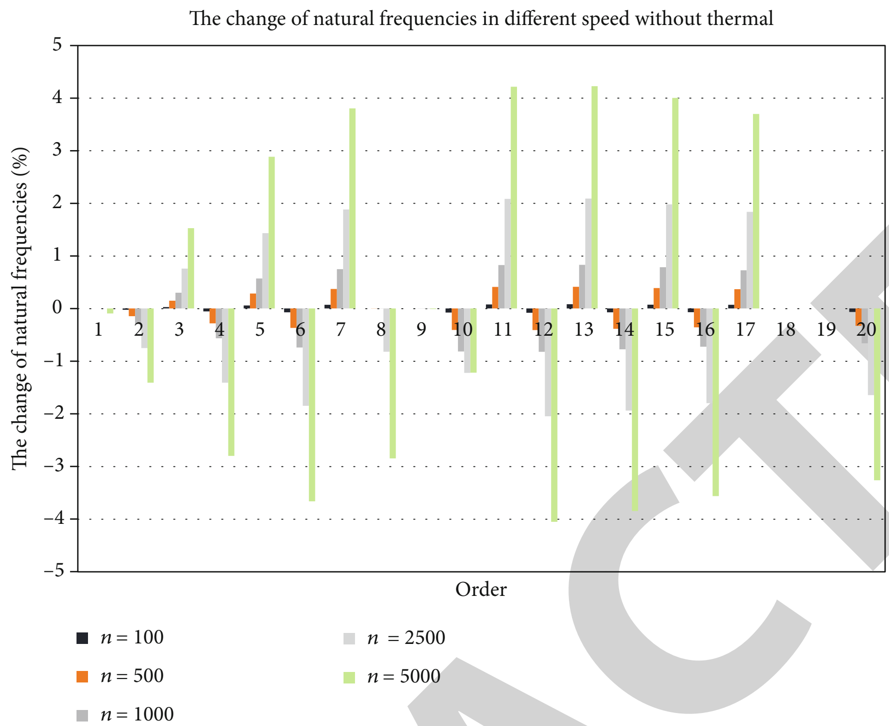


Figure 3: The change of natural frequency at each rotational speed without thermal load.

Table 5: The first 20 order natural frequency values with thermal load and no rotational speed (heat flow = 1000 W).

The order	Natural frequency (Hz)
1	4585.6
2	4586.2
3	4892.8
4	4893.1
5	5485.5
6	5485.6
7	6377.6
8	6377.8
9	6444
10	6444.5
11	7783.4
12	7783.8
13	9463.6
14	9464.8
15	11420
16	11420
17	12523
18	12523
19	13580
20	13581

influence on stiffness at high speed. In modal analysis of aerial bevel gears, the stiffness matrix should include the condition of rotation softening. On the basis of the original stiffness, considering the influence of centrifugal force [23], set

$$K' = K + \Omega^2 M, \tag{17}$$

variables

$$\Omega^2 = \begin{bmatrix} -\left(\bar{\omega}_y^2 + \bar{\omega}_z^2\right) & \bar{\omega}_x\bar{\omega}_y & \bar{\omega}_x\bar{\omega}_z \\ \bar{\omega}_x\bar{\omega}_y & -\left(\bar{\omega}_x^2 + \bar{\omega}_z^2\right) & \bar{\omega}_y\bar{\omega}_z \\ \bar{\omega}_x\bar{\omega}_z & \bar{\omega}_y\bar{\omega}_z & -\left(\bar{\omega}_x^2 + \bar{\omega}_y^2\right) \end{bmatrix}, \tag{18}$$

where $\bar{\omega}_x\bar{\omega}_y\bar{\omega}_z$ is the angular velocity vector $\bar{\omega}$ along the x, y, and z axes.

Substituting K in Equation (16) for Equation (17), the modal equation considering the action of the velocity field is

$$\left|\left(K + \Omega^2 M\right) - \omega_i^2 M\right| = 0. \tag{19}$$

When the temperature field is considered, there will be thermal deformation in the gear tooth surface caused by temperature change. According to thermoelasticity, thermal deformation will produce the corresponding thermal stress field. The existence of the thermal stress field changes the stress state of gear and then produces stress

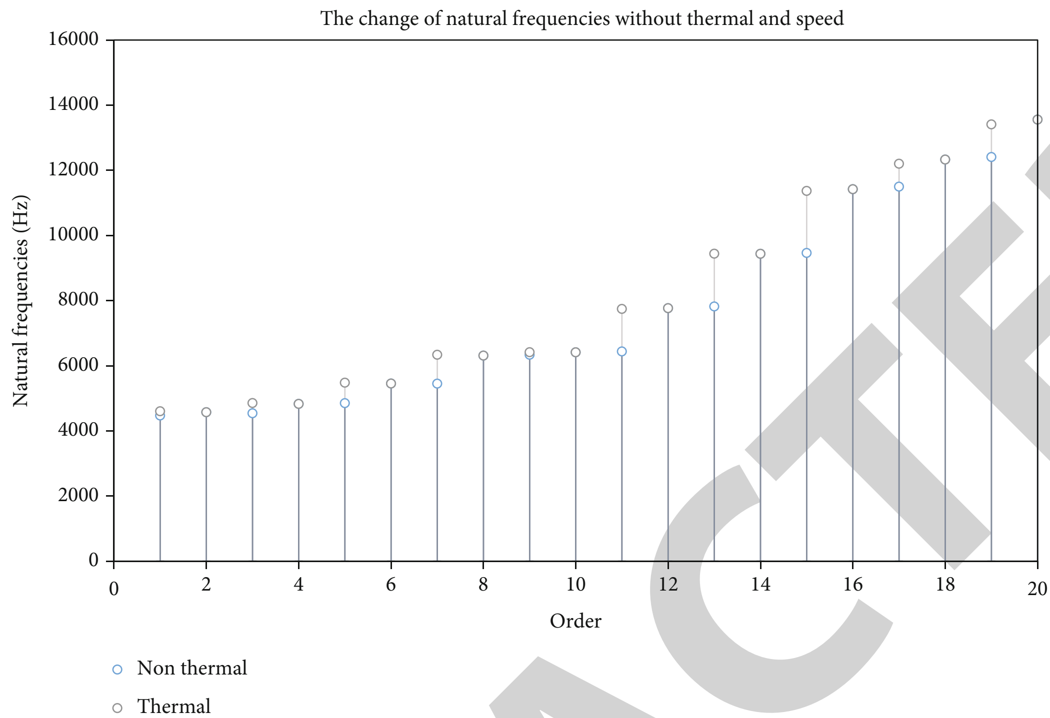


Figure 4: The first 20 order changes of natural frequency value with thermal load and without rotational speed (heat flow = 1000 W).

stiffness (or softening) effect. According to the stress-stiffness relationship in Reference [23], the thermal stress stiffness matrix of the element is assumed to be

$$k_s^e = \int G^T \sigma G d\Omega, \tag{20}$$

where G is the derivative matrix of shape stiffness and σ is the Cauchy stress matrix generated by unit heat.

The thermal stress stiffness matrix k_s^e of each element is combined to obtain the overall thermal stress stiffness matrix K_s. Add the overall thermal stress stiffness matrix K_s to the right end of Equation (17), and obtain the modal equation of aerial bevel gear considering the action of the temperature field:

$$\left|\left(K + K_s + \Omega^2 M\right) - \omega_i^2 M\right| = 0. \tag{21}$$

The temperature field not only produces stress stiffness but also influences the properties of materials, which is not considered in this paper. Therefore, the material property is set to a constant value in the analysis.

3. Modal Analysis of Aeronautic Spiral Bevel Gear at a Steady Temperature Field

3.1. Build Finite Element Model. According to Reference [8], the parameters of an aeronautic spiral bevel gear are tooth number $z_1 = 12 z_2 = 36$, normal module $m_n = 4.9407$ mm, tooth width $b = 22.5$ mm, spiral angle $\beta_m = 35^\circ$, pitch $R_m = 81.1$ mm, normal pressure angle, $\alpha_n = 22.5^\circ$, axis angle, $\Sigma = 90^\circ$.

Firstly, according to the gear parameters, a 3D model of spiral bevel gear is created in UG, as shown in Figure 1. Then, it was imported into the finite element analysis software, and the material characteristics were added for modal analysis. In this paper, the influence of temperature on material properties is not considered, and the material property coefficients such as elastic modulus, density, and Poisson's ratio are all fixed. The gear material used is 16Cr3NiWmoVbE, with elastic modulus $E = 2.10 \times 10^5$ MPa, Poisson's ratio $\upsilon = 0.3$, and density 7.85×10^3 kg/m^3. According to Reference [8], the boundary conditions of steady-state thermal analysis were set. The heat flow was 1000 W, the working temperature was 70°C, and the convective heat transfer coefficient parameters at each part of the gear tooth are shown in Table 1. In general, the calculation accuracy increases with the decrease of mesh density, but the corresponding calculation cost also increases greatly. Therefore, before the modal analysis, the influences of different mesh densities on the calculation results under the same heat flow rate were firstly analyzed in the thermal analysis, and the results are shown in Table 2. As can be seen from Table 2, the influence of mesh density on thermal analysis results can be ignored basically. However, in consideration of the need to transform into thermal stiffness in the next step, the mesh density is selected as 4, and the unit type is SOLID187 to divide the mesh, and the mesh number is 39570. Due to the large number of elements, the overall stiffness matrix state is large, so it is not convenient to give in this paper; this paper only analyzes the finite element results.

Table 6: The first 20 order natural frequency values at each speed with thermal load (heat flow = 1000 W).

The order	The natural frequency (Hz) $n = 100$ rpm	The natural frequency (Hz) $n = 200$ rpm	The natural frequency (Hz) $n = 1000$ rpm	The natural frequency (Hz) $n = 2500$ rpm	The natural frequency (Hz) $n = 5000$ rpm
1	4522.9	4522.9	4522.9	4522.9	4516.8
2	4584.5	4578.9	4572	4551.2	4522.9
3	4587.3	4592.8	4599.8	4620.8	4655.9
4	4890.1	4879	4865.2	4823.9	4755.9
5	4895.7	4906.9	4920.8	4962.9	5034
6	5481.4	5465.1	5444.7	5383.9	5284.2
7	5489.6	5506.1	5526.7	5589	5694.5
8	6377.6	6377.6	6377.6	6312.2	6182.9
9	6377.8	6377.8	6377.8	6377.4	6377.2
10	6438.9	6417.6	6391.1	6378	6378.3
11	6449.6	6471	6497.8	6579	6716.6
12	7777.1	7751.4	7719.3	7623.7	7467.2
13	7790.1	7816	7848.5	7946.8	8113.4
14	9456.7	9427	9389.9	9279.6	9098.7
15	9471.7	9501.5	9539	9652.4	9844.4
16	11412	11379	11337	11214	11012
17	11429	11462	11504	11630	11843
18	12523	12523	12523	12523	12522
19	12523	12523	12523	12524	12524
20	13572	13536	13491	13357	13137

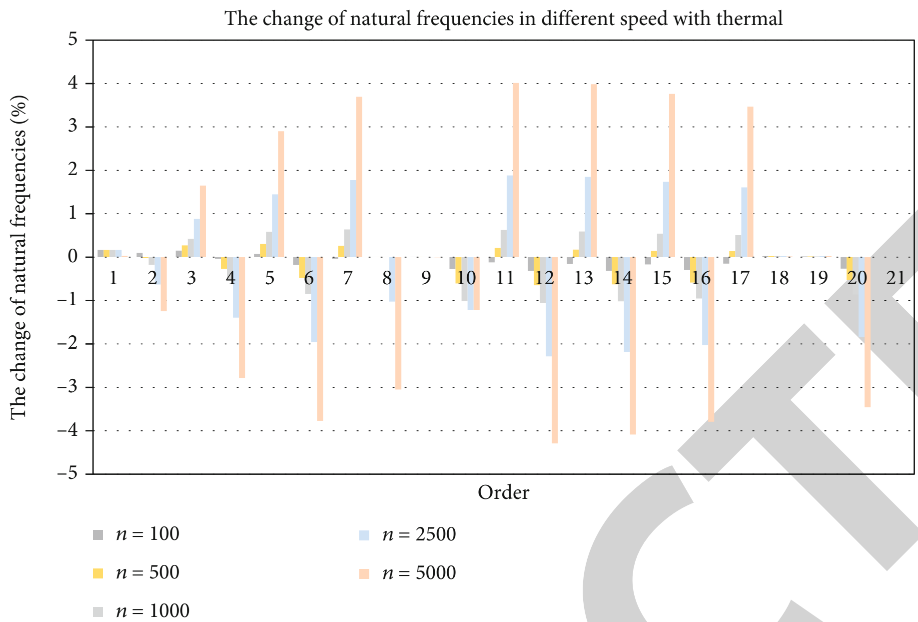


Figure 5: The change of natural frequency value under various rotational speed conditions with thermal load (heat flow = 1000 W).

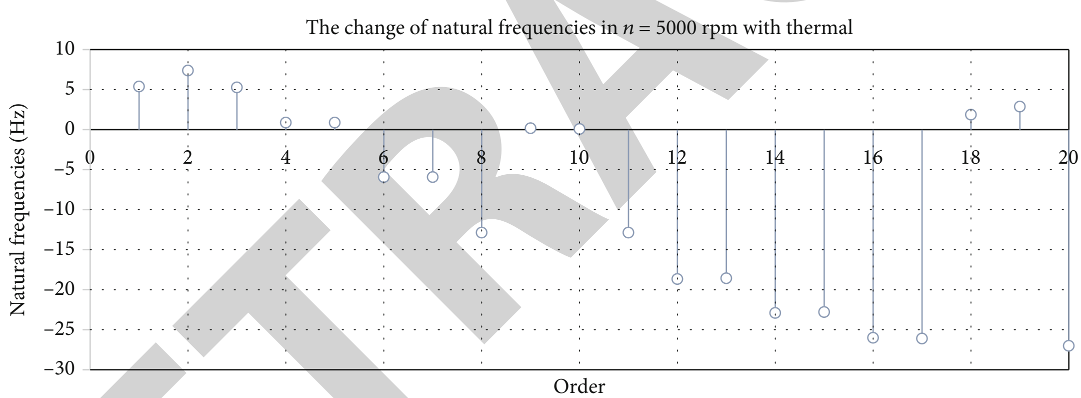


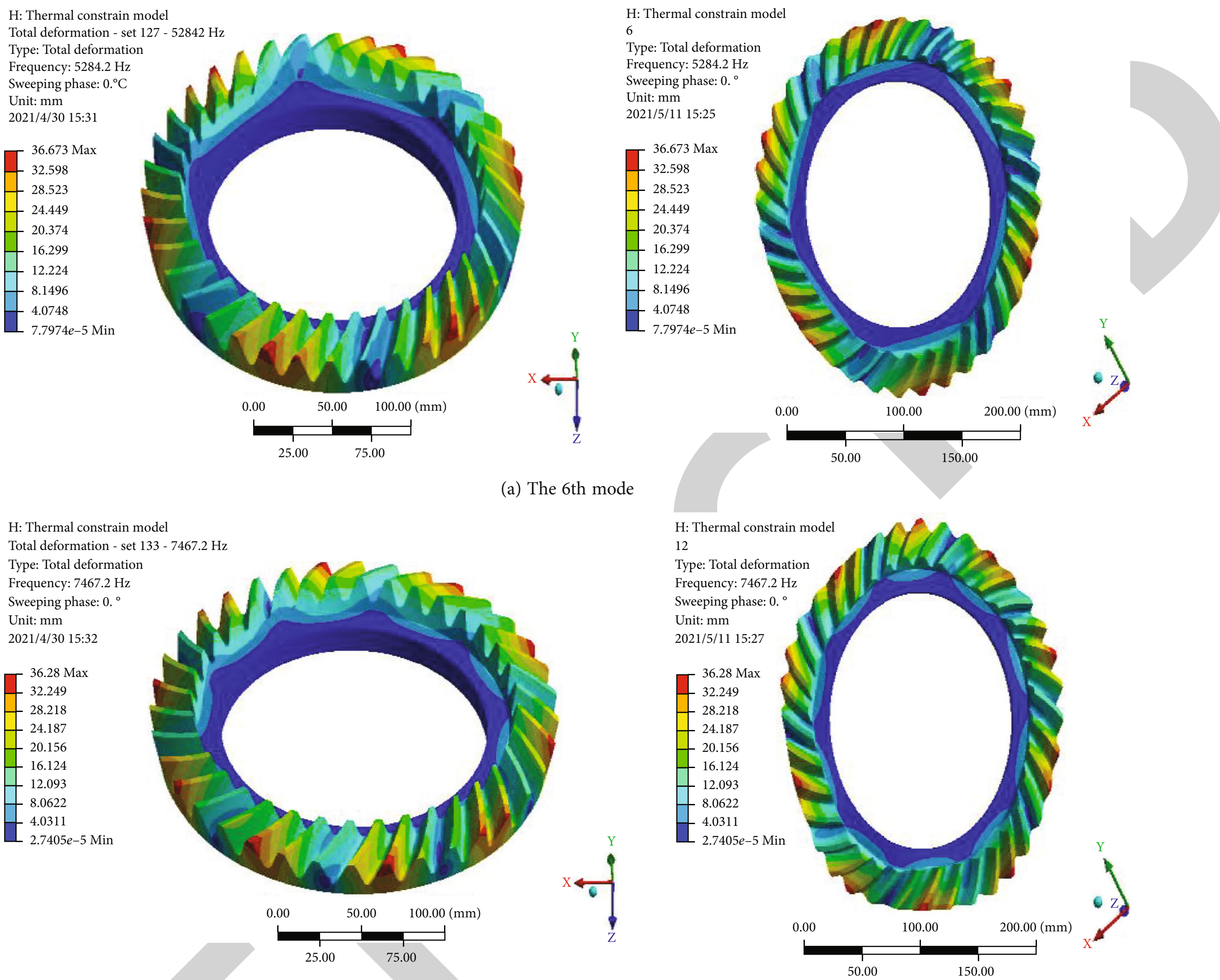
Figure 6: $n = 5000$ rpm change of natural frequency with and without thermal load (heat flow = 1000 W).

3.2. Finite Element Analysis and Results. The output of thermal analysis is thermal deformation, which needs to be transformed into thermal stress through the structural module. And because the nonlinear coupling degree between temperature and structure is not high, the sequential coupling method is used for thermosolid analysis in this paper. The modal analysis of aviation bevel gear under the steady-state temperature field is as follows: Firstly, the temperature field and thermal deformation are calculated by the steady-state thermal module. Then, the thermal stress is calculated according to the deformation of the steady-state structural module. Finally, the calculated thermal stress is introduced into the modal analysis.

Figure 2 shows the temperature field distribution calculated by the steady-state thermal analysis module. As the heat transfer speed is relatively slow, the solution time is set as 60 s to ensure that the calculated value is more consistent with the real working condition. Except for the three teeth engaged in meshing, the temperature of the remaining teeth is equivalent to the working temperature, and the highest temperature is at the top of the teeth engaged, because the relative slip velocity of the tooth tip is relatively large, which is consistent with the theoretical analysis. Therefore, it can be considered that the temperature distribution is reasonable.

In order to analyze the effects of temperature distribution and rotational speed on the mode of aerial bevel gears, it is necessary to calculate the constrained mode of aerial bevel gears without heat load and rotational speed. For reasons of space, only the first 20 modes are listed. Specific values are shown in Table 3.

Firstly, the influence of rotational speed on aviation bevel gear is considered separately. The speed constraint was added to the modal model without heat load and without speed. The speed conditions were 100, 500, 1000, 2500, and 5000 rpm, respectively, corresponding to the three working conditions of low speed, medium speed, and high speed, respectively. The specific values of the first 20 orders



(a) The 6th mode

(b) The 12th mode

FIGURE 7: The modes of 7th and 13th order under heat load conditions.

are shown in Table 4. By comparing the data in Tables 1 and 4, the trend of modal changes is shown in Figure 3. On the whole, the modal change can be ignored at low speed. At high speed, especially at $n = 5000$, the modal changes greatly. Among them, the change range of the 12th and 13th order is the largest, and the maximum change value can reach 4.22%. Besides, except for the 1st, 9th, 18th, and 19th order, the modal change of each order is above 1%. It can be seen that the stiffness change generated at high speed is not negligible. Therefore, the effect of high speed on the natural characteristics of bevel gears must be considered when designing them. In addition, it can also be seen from Figure 3 that the symbol of modal difference is changing; that is, the effect of high speed on different orders is different. When the angle between vibration direction and centrifugal force direction of a certain order of vibration mode is less than 90°, stiffness strengthening is displayed; otherwise, stiffness softening is displayed.

Secondly, the influence of the temperature field on modal is considered separately. According to the theory of thermoelasticity, the thermal deformation caused by temperature is calculated first; then, the thermal stress is calculated according to the thermal strain through the constitutive equation of the material, and finally, the calculated stress field is loaded onto the constrained mode without speed. The first 20 order natural frequency values are shown in Table 5. Compared with the natural frequency value under the condition of no heat load and no speed, the change of natural frequency value under the condition of no heat load and no speed was obtained, as shown in Figure 4. As can be seen from Figure 4, modal changes in the first 20 orders are mostly positive changes, among which the 11th order has the largest change range, but overall, the change rate is less than 0.0035%. Therefore, in the case of no rotational speed, the influence of the temperature field on the natural characteristics of aviation bevel gears can be ignored.

Finally, the air bevel gear modal under the action of the temperature field and velocity field is considered. The speed constraint condition was added to the previous model with hot load and no speed, and the speed conditions were 100, 500, 1000, 2500, and 5000 rpm, respectively, corresponding to the three conditions of low speed, medium speed, and high speed, respectively. The natural frequency values of the first 20 orders are calculated numerically, and the specific values are shown in Table 6. Compared with the natural frequency without thermal load and without rotational speed, the modal change trend considering temperature field and rotational speed was obtained, as shown in Figure 5. It can be seen that, similar to the change trend of only rotational speed without considering thermal load, the modal changes greatly at high rotational speed, but the value of modal change changes. Due to space problems, this paper only plots the change of the modal change value of $n = 5000$ (see Figure 6). As shown in Figure 6, the change of multimodal is negative, so it can be seen that the thermal stress generated by the temperature field is opposite to the stress generated by centrifugal force in most cases. To sum up, the rotational speed has a great influence on the variation trend of natural frequency value, while the temperature field has an impact on its value. Therefore, it is necessary to consider the influence of temperature and speed when designing aviation bevel gears.

Modal analysis not only considers the change of natural frequency but also cannot be ignored; that is, temperature and speed have a greater impact on which type of modal. For lack of space, only the case of 5000 r/min is discussed here. It can be seen from Figure 3 that the influence of rotational speed on natural frequency is greater than 4% in orders 11, 12, and 13 and between 3% and 4% in orders 6, 7, 14, 15, 16, 17, and 20. From the temperature field shown in Figure 4, it can be seen that orders 7, 11, and 13 have a great influence on the natural frequency. As shown in Figure 5, when the temperature field and rotational speed coexist, orders 11, 12, 13, and 14 have a greater impact, while orders 6, 7, 15, 16, 17, and 20 take second place. Therefore, the 6th and 12th orders under the action of thermal load were selected to represent low-order and high-order conditions, respectively, for analysis, and the results are shown in Figure 7. The two modes shown in Figure 7 are pitch mode modes. If the node-diameter vibration is involved, the corresponding shape of the vibration mode in the time domain is the beat shape, which is the bending deformation perpendicular to the disk surface, and the change of this type will affect the gear structure, making it prone to failure, fracture, and other failures. Therefore, from the perspective of vibration mode, it is necessary to consider the influence of temperature and speed on the air bevel gear.

4. Conclusion

Based on the linear modal theory, a modal analysis model based on temperature field action is established. According to the numerical analysis of the model for a certain aeronautical bevel gear, the natural frequencies of each order under the action of the temperature field were obtained. Based on the modal analysis, the following conclusions were obtained:

(1) The uneven distribution of the temperature field produces certain thermal stress, and the overall stiffness of gear will change under the action of thermal stress. The complex geometry of bevel gears will increase the inhomogeneity of temperature field distribution. Therefore, temperature has a greater impact on the stiffness of gears than spur gears

(2) Aviation bevel gears will be in high-speed condition, and the centrifugal force generated by high-speed rotation will produce a stiffness hardening effect, but the geometry of bevel gears will change its direction, and when reversed, the softening effect will be reflected

(3) By comparing the results of numerical calculation, it can be seen that the influence of rotational speed on modal is far greater than that of the temperature field on natural frequency when only a single factor is considered in modal analysis, especially in middle and high order modes. The results show that temperature has a certain effect on the natural frequency, and the positive and negative effects are variable

(4) According to the analysis of vibration modes of each step, the temperature field and speed field mainly affect the pitch shape, and the vibration mode has a certain influence on gear failure. Therefore, it is suggested that temperature and speed should be taken into account when designing bevel gears

Data Availability

The dataset can be accessed upon request.

Conflicts of Interest

The authors declare no conflict of interest.

Authors' Contributions

Wentao Yu and Shijun Liu conceived and designed the experiments; Wenbo Xu performed numerical calculation; Wentao Yu and Dongfei Wang performed the result analysis; Wentao Yu, Shijun Liu, Wenbo Xu, and Dongfei Wang wrote the paper. Dongfei Wang and Wentao Yu contributed equally to this work.

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