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Research on Thermal Deformation of Solid Lubricated Gears for Space Deployable Mechanisms

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Abstract: Extreme temperature fluctuations in the space environment can cause thermal deformation of transmission components in aerospace equipment, thereby affecting the normal operation of transmission mechanisms. Focusing on gears coated with solid lubricating films, this paper conducts theoretical research and analysis related to thermal deformation. Based on the theory of thermal expansion of materials, the quantitative relationships among linear, areal, and volumetric expansion coefficients are clarified, and an equivalent conversion model between volumetric expansion and linear expansion is established. Combined with thermoelasticity, the thermoelastic equations of gears and solid lubricating coatings under temperature changes are derived, and their thermal deformation characteristics are analyzed. On this basis, the influence mechanism of temperature changes on the normal backlash of gear pairs is emphatically studied. By analyzing the thermally induced changes of parameters such as base circle radius and pressure angle, the final calculation expression of normal backlash is derived. The research results provide a theoretical basis for evaluating the meshing safety of solid-lubricated gear pairs under extreme thermal environments in space and avoiding interference risks, and have important engineering significance for ensuring the stable and reliable operation of transmission systems in aerospace deployable/retractable mechanisms.

Keywords: Solid-lubricated Gear, Thermal Deformation, Normal Backlash, Space Environment

1. Introduction

The space environment, serving as the unique operational medium for aerospace equipment, is characterized notably by the absence of atmospheric buffering and regulation of temperature[1]. Under the alternating influence of space environmental factors such as direct solar radiation and Earth shadow occlusion, the surface temperature of spacecraft undergoes severe periodic fluctuations, creating an extreme temperature differential environment. Such drastic temperature variations pose serious challenges to the structural stability, functional reliability, and operational lifespan of aerospace equipment, with particularly pronounced impacts on core moving components such as transmission systems[2, 3].Gear transmission, with its advantages such as high transmission efficiency, strong load-bearing capacity, and precise transmission ratio, is widely used in critical transmission scenarios such as aerospace deployment mechanisms, attitude control systems, and propulsion systems. It serves as a core component in ensuring the successful implementation of aerospace missions. In the extreme temperature-difference environment of space, gear materials undergo significant thermal deformation due to thermal expansion and contraction[4]. This deformation not only alters key geometric parameters of the gear, such as tooth profile shape, base circle radius, and tooth thickness, but also directly affects the meshing clearance and transmission accuracy of gear pairs. When temperature changes, the thermal deformation of components such as the gear body, shaft system, and housing may lead to a reduction or even elimination of the meshing backlash, causing issues such as interference, jamming, and accelerated wear in the gear pair. Therefore, conducting in-depth research on the thermal deformation mechanisms of solid lubricant gears under extreme temperature environments and revealing the influence patterns of temperature variations on gear geometric parameters and meshing characteristics have become a key technical challenge in ensuring the reliable operation of aerospace transmission systems[5].

Research on gear thermal deformation began in the mid-20th century. Scholars from both domestic and international backgrounds have conducted extensive work on the thermal deformation mechanisms, calculation methods, and influencing factors under conventional operating conditions[7, 8]. At the theoretical level, early studies established thermal deformation calculation models for key geometric parameters of gears based on material thermal expansion theory and elasticity mechanics. Subsequently, the application of finite element methods advanced the development of thermal-structural coupling analysis for gears, enabling numerical simulation of temperature distribution and thermal deformation[9, 10]. On the experimental front, dedicated testing platforms have been built to measure gear tooth profile deformation and meshing clearance variations under different temperature and load conditions, providing data support for the validation of theoretical models[11]. However, existing research has predominantly focused on conventional temperature ranges and uncoated gears, lacking adaptability to the extreme temperature-difference environment in space. Moreover, it often fails to adequately consider the thermophysical property differences between solid lubricant coatings and substrates, making it difficult to accurately describe thermal deformation behavior under complex coupled effects[12, 13].

In summary, while existing research has achieved certain progress in gear thermal deformation theory and the application of solid lubricant coatings, significant shortcomings remain in studies on solid lubricant gears for extreme space thermal environments. This is particularly evident in the absence of a thermal deformation theoretical model that can effectively characterize the thermophysical property differences between coatings and substrates.

2. Theory of Thermal Deformation of Solid Lubricated Gears

2.1. The Theory of Thermal Expansion of Materials

In mechanical design, changes in ambient temperature can cause variations in the dimensions of components. Typically, we use the following empirical formula for calculation:

$$\Delta L = L \cdot \alpha_m \cdot \Delta t \tag{1}$$

Where:

L ——represents the length of the component at the initial temperature (usually 25°C), with the unit of (mm);

α_m ——represents the average linear expansion coefficient of the component within a certain temperature range, with the unit of (1E-6/°C);

ΔL ——represents the length change of the component during the initial stage of temperature change, with the unit of (mm);

Δt ——represents the temperature variation, with the unit of (°C).

Typically, thermal expansion leads to linear expansion, areal expansion, and volumetric expansion of components, which correspond to the linear expansion coefficient, areal expansion coefficient, and volumetric expansion coefficient respectively, and there exists a certain quantitative relationship among the three.

The average linear expansion coefficient in the above formula can be obtained through conversion as follows:

$$a_m = \frac{\Delta L}{L \Delta t} \tag{2}$$

Wherein, let the length of the component be *L* at the initial temperature t0; when the temperature is t1,the length of the component becomes L1. Therefore, at this point $\Delta L = L_1 - L$, $\Delta t = t_1 - t_0$.

Substituting it into the original formula and performing differentiation on the formula, the instantaneous thermal expansion coefficient can be obtained, which is expressed as:

$$\alpha_s = \frac{1}{L_i} \lim_{t_1 \to t_0} \frac{L_1 - L}{t_1 - t_0} = \frac{(dL/dt)}{L_i} \tag{3}$$

Wherein Li is the length of the component at the instantaneous temperature.

For the average areal expansion coefficient Sm, let the initial length and width, as well as the length and width after temperature change, be A0,B0,A1 and B1 respectively. Then, the area S1 of the material after temperature change is expressed as:

$$\begin{aligned} S_1 = A_1 \cdot B_1 = A_0 \cdot B_0 \cdot [1 + \alpha_m (t_1 - t_0)]^2 \\ = S_0 [1 + 2\alpha_m (t_1 - t_0) + \alpha_m{}^2 (t_1 - t_0)^2] \end{aligned} \tag{4}$$

It can be derived with the aid of the formula for the areal expansion coefficient as follows:

$$S_m = \frac{s_1 - s_0}{s_0 (t_1 - t_0)} = 2\alpha_m (t_1 - t_0) + \alpha_m{}^2 (t_1 - t_0)^2 \tag{5}$$

Since the term αm in the formula is much smaller than 1, their mathematical relationship can be approximately obtained Sm=2αm.Similarly, the relationship between the linear expansion coefficient and the volumetric expansion coefficient can be derived as Vm=3αm.

By sorting out and deriving the above formulas, the mathematical correspondence between the expansion amounts in different directions can be clarified, thereby establishing an equivalent conversion model between the linear expansion and volumetric expansion of materials. Based on this mathematical relationship, in the subsequent thermal deformation analysis, the volumetric expansion coefficient of the material can be replaced by the average linear expansion coefficient to effectively predict the variation trend of the material volume with temperature. Due to the definite proportional relationship between the volumetric expansion coefficient and the linear expansion coefficient, this replacement method can significantly simplify the difficulty of theoretical calculations while ensuring calculation accuracy, and improve the operability and engineering application value of the analysis process.

2.2. Analysis on Thermal Environment Adaptability of Solid-Lubricated Gears

When studying thermal deformation, it is first necessary to analyze thermal stress and establish the relationship between thermal stress and thermal deformation based on thermoelastic mechanics. Meanwhile, for solid-lubricated gears coated with molybdenum disulfide, attention should be paid to the differences between the solid coating and the gear material when considering their thermal deformation. For two components with uniform heterogeneity, when temperature changes occur, both will undergo tensile strain expansion or contraction simultaneously without generating shear strain. Thus:

$$\begin{cases} \delta_{x1} + \delta_{x2} = \delta_{y1} + \delta_{y2} = \delta_{z1} + \delta_{z2} = \alpha_m \Delta t \\ \varepsilon_{x1y1} + \varepsilon_{x2y2} = \varepsilon_{y1z1} + \varepsilon_{y2z2} = \varepsilon_{z1x1} + \varepsilon_{z2x2} = 0 \end{cases} \tag{6}$$

Where:

δ_{x1}, δ_{x2}, δ_{y1}, δ_{y2}, δ_{z1}, δ_{z2} ——represent the tensile strains of the two materials in the x, y, and z directions respectively;

ε_{x1y1}、ε_{x2y2}、ε_{y1z1}、ε_{y2z2}、ε_{z1x1}、ε_{z2x2} —— represent the shear strains of the two materials on the three planes respectively.

Substituting the above equations into the elastomechanics equations, the thermoelastic mechanics equations for the tensile strains in each direction can be obtained, namely:

$$\begin{cases} \delta_{x1} + \delta_{x2} = \dfrac{1}{E}\left[\sigma_x - (\upsilon_1 + \upsilon_2)(\sigma_y + \sigma_z)\right] + \alpha_m \Delta t \\ \delta_{y1} + \delta_{y2} = \dfrac{1}{E}\left[\sigma_y - (\upsilon_1 + \upsilon_2)(\sigma_x + \sigma_z)\right] + \alpha_m \Delta t \\ \delta_{z1} + \delta_{z2} = \dfrac{1}{E}\left[\sigma_z - (\upsilon_1 + \upsilon_2)(\sigma_y + \sigma_x)\right] + \alpha_m \Delta t \end{cases} \tag{7}$$

Where:

E—— represents the elastic modulus;

υ—— represents the Poisson's ratio;

σ—— represent the magnitudes of forces acting in each direction respectively.

2.3. Temperature Field in the Gear Base Circle

When analyzing the thermal deformation theory of gears, it is assumed that the gear is in an ideal state, the meshing contact is ideal contact, and there is no uneven load in the tooth surface direction. Meanwhile, it is assumed that the temperature variation along the axial direction is zero, and the temperature distribution of the gear is uniform under stable operating conditions. Thus, the temperature can be considered to vary only with the radial radius r. Under these conditions, the heat conduction differential equation can be simplified to the following form:

$$\frac{d}{dr}\left(r\frac{dt}{dr}\right)=0 \tag{8}$$

Integrating this formula yields:

$$t(r)=C_1\ln r+C_2 \tag{9}$$

Substituting r=ra, t=ta; r=rb, t=tb, the relevant parameters into the formula, the temperature distribution in the base circle of the gear without considering the solid coating can be obtained as follows:

$$t(r)=t_a+(t_b-t_a)\frac{\ln r-\ln r_a}{\ln r_b-\ln r_a} \tag{10}$$

Wherein, t represents the temperature at different time periods, and ra and rb represent the gear inner diameter and the gear base circle radius respectively.

By combining this formula with the aforementioned thermoelastic mechanics equations, the thermal strain and thermal deformation caused by the thermal stress on the gear can be calculated.

3. Formulaic Study on the Effect of Temperature Variation on Gear

3.1. Theoretical Calculation of Thermal Deformation for Disc-Type Gears

For disc-type gears, the base cylinder of the gear can be regarded as being subjected only to plane stress. Meanwhile, the ambient temperature is uniformly distributed. Therefore, when considering the radial deformation β of the gear, it can be regarded as being related only to the gear radius r, and the tangential displacement Y in all directions is zero. Substituting into the previous equation yields a new expression as follows:

$$\begin{cases}\delta_r=\dfrac{\partial\beta}{\partial r}=\dfrac{1}{E}\left[\sigma_r-\upsilon\sigma_\theta\right]+\alpha_m\Delta t\\[2ex]\delta_\theta=\dfrac{\beta}{r}=\dfrac{1}{E}\left[\sigma_\theta-\upsilon\sigma_r\right]+\alpha_m\Delta t\end{cases} \tag{11}$$

Where:

σ_r, σ_θ, σ_z ——represent the radial stress and circumferential stress of the gear respectively.

By rearranging the formula and expressing the result in terms of displacement, the corresponding stress is given by:

$$\begin{cases}\sigma_r=\dfrac{E}{(1-\nu^2)}\left[\dfrac{d\beta}{dr}+\nu\dfrac{\beta}{r}-(1+\nu)\alpha_m\Delta t\right]\\[2ex]\delta_\theta=\dfrac{E}{(1-\nu^2)}\left[\dfrac{\beta}{r}+\nu\dfrac{d\beta}{dr}-(1+\nu)\alpha_m\Delta t\right]\end{cases} \tag{12}$$

The stress equilibrium equation in the cylindrical coordinate system is as follows:

$$\begin{cases}\dfrac{\partial\sigma_r}{\partial r}+\dfrac{\partial\tau_r}{\partial z}+\dfrac{\sigma_r-\sigma_\theta}{r}=0\\[2ex]\dfrac{\partial\tau_r}{\partial r}+\dfrac{\partial\sigma_z}{\partial z}+\dfrac{\tau_r}{r}=0\end{cases} \tag{13}$$

Where τ_r is the shear stress. Substituting the above equation into the stress equilibrium equation yields:

$$\frac{d^2\beta}{dr^2}+\frac{1}{r}\frac{d\beta}{dr}-\frac{\beta}{r}=(1+\nu)\alpha_m\frac{dt}{dr} \tag{14}$$

By integrating, the thermal deformation displacement β can be obtained as follows:

$$\beta=\frac{(1+\nu)\alpha_m}{r}\int_{ra}^{r}tr\ dr+\frac{k_1}{2}r+\frac{k_2}{r} \tag{15}$$

Substituting the aforementioned base circle temperature field formula into the radial displacement β and setting r=rb, the thermal deformation formula of the gear base circle can be obtained as follows:

$$\beta_b=r_b\alpha_m t_a+\frac{a_m r_b^3(t_b-t_a)}{r_b^2-r_a^2}-\frac{a_m r_b(t_b-t_a)}{2(\ln r_b-\ln r_a)} \tag{16}$$

3.2. Expression for the Normal Backlash of Gear Pairs After Thermal Deformation

Temperature variation has a significant influence on gear backlash, and its mechanism of action is mainly reflected in the thermal expansion and contraction of materials. When a gear transmission system is in operation, if the temperature rises, key components such as the gear body, gearbox, and shaft will undergo dimensional changes due to thermal expansion. This expansion will increase the gear center distance and change the base circle tooth thickness of the gear itself, thereby directly affecting the non-working tooth surface clearance during meshing. If the initial backlash is improperly designed, temperature rise may lead to a series of issues such as insufficient tooth surface lubrication, intensified tooth wear, and increased vibration and noise, which seriously affect the transmission accuracy and service life. Therefore, when designing gears for high-precision applications or working conditions with large temperature differences, it is essential to take the expected operating temperature range and the thermal expansion coefficient of materials as key parameters, and conduct necessary temperature compensation corrections on the theoretical backlash to ensure the stable and reliable operation of the transmission system under all working conditions. For this reason, this chapter will focus on deriving the calculation expression for the normal backlash increment of gear pairs considering the deformation of coated solid lubricants[14].

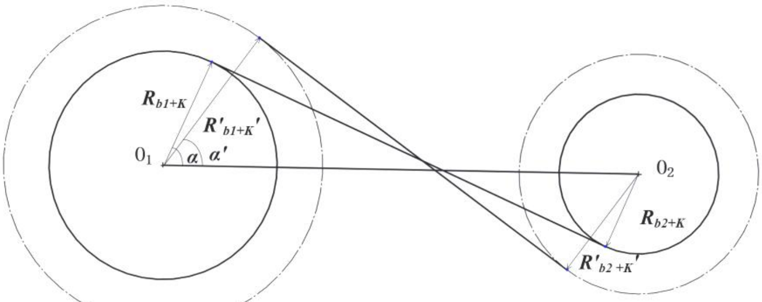

Figure 1: Schematic Diagram of Thermal Deformation of the Base Circle

Wherein, O1 and O2 represent the rotation centers of the two meshing gears; Rb1, Rb2 R'b1, and R'b2 denote the base circle radii of the gears before and after temperature change; K and K' indicate the thicknesses before and after solid coating; and α and α' represent the meshing angles of the gear pair before and after thermal deformation.

By means of the derived expression for the thermal deformation of the gear base circle mentioned above, the changed base circle radii R'b1 and R'b2 can be expressed accordingly.

$$\begin{cases} R'_{b1} = R_{b1} + \beta_{b1} + K' \\ R'_{b2} = R_{b2} + \beta_{b2} + K' \end{cases} \tag{17}$$

Where:

β1, β2—— thermal deformation of the gear base circles.

Under the condition that the gear center distance remains unchanged due to heat, the reference circle pressure angle α can be expressed as follows:

$$\frac{R_{b1}' + 2K' + R_{b2}'}{\cos\alpha'} = \frac{R_{b1} + R_{b2} + 2K}{\cos\alpha} \tag{18}$$

Therefore, the new meshing angle after temperature variation is as follows:

$$\alpha' = \arccos\left(\frac{R_{b1} + \beta_{B1} + 2K' + R_{b2} + \beta_{b2}}{R_{b1} + R_{b2} + 2K}\right) \tag{19}$$

By means of the formula between the base circle and the reference circle, the base circle pitch Yb1 and the base circle tooth thickness Nb after thermal deformation can be calculated as follows:

$$Y_{b1} = \frac{2\pi(R_{b1}')}{Z} \tag{20}$$

$$\begin{cases} N_{b1} = (R_{b1}' + K')\left[\frac{S_1}{R_1} - (inv\alpha' - inv\alpha)\right. \\ N_{b2} = (R_{b2}' + K')\left[\frac{S_2}{R_2} - (inv\alpha' - inv\alpha)\right. \end{cases} \tag{21}$$

S1, S2— represent the reference circle tooth thicknesses respectively.

Finally, the final normal backlash caused by the gear backlash variation induced by temperature change is derived as follows:

$$F = Y_{b1} - N_{b1} - N_{b1} + 2(R_{b1}' + R_{b2}' + K')inv\alpha' \tag{22}$$

Through a systematic analysis of the thermal expansion behavior of solid-lubricated bevel gears under temperature variation, an analytical expression for the final normal backlash of gears after the adjustment of meshing geometric parameters caused by ambient temperature changes is obtained. The derived results provide a clear theoretical basis for subsequent evaluation of the meshing safety margin of gear pairs, judgment of potential interference risks, and conduct of thermal-structural coupling analysis in the space thermal environment.

4. Validation via Simulation Analysis

The analytical expression for the normal backlash considering the gear thermal deformation effect derived above from a theoretical perspective provides a key theoretical model for evaluating the on-orbit precision of space mechanisms. However, the accuracy and applicability of this formula still need to be verified through rigorous numerical experiments. Therefore, based on the aforementioned theoretical achievements, a corresponding parametric simulation model of the gear pair will be established to systematically conduct simulation analysis under the thermal-structural coupled field. By simulating the dynamic response of the mechanism under actual high and low temperature operating conditions, the variation of the normal backlash of the gear pair will be accurately quantified, thereby cross-validating the correctness of the theoretical derivation and providing reliable data support for subsequent on-orbit performance prediction.

4.1. Simulation Results

To quantitatively evaluate the effect of temperature load on transmission accuracy, the probe function of ANSYS was adopted to arrange displacement monitoring points at the key contact points of gear meshing. The set monitoring points cover typical positions including the start of meshing, the middle of contact and the end of disengagement, and the three-directional displacement time-series signals varying with temperature were recorded under a unified coordinate system. To ensure the reliability of the results, all monitoring points correspond to the nodes after mesh generation or are interpolated to the mesh nodes, and post-processing analysis was carried out under the premise that the boundary conditions and contact definitions remain consistent. Finally, the precise variation of the relative position of the tooth surface under different thermal deformation conditions can be obtained.

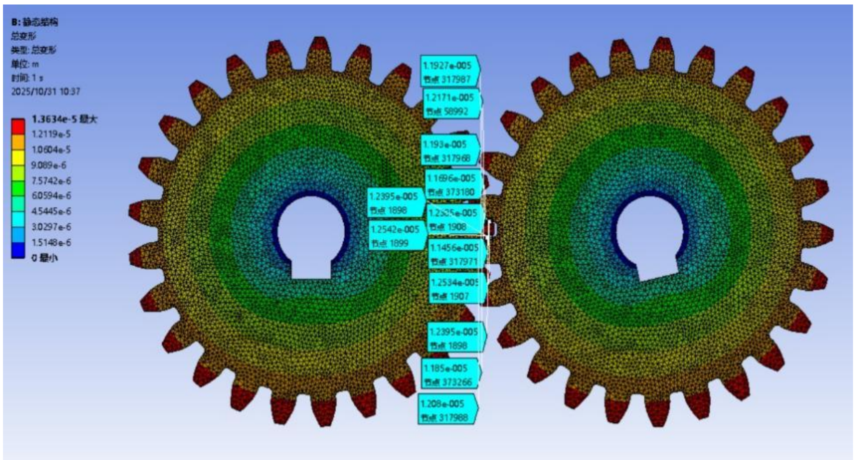

Figure 2: Deformation Diagram of Gear Pair at 100℃

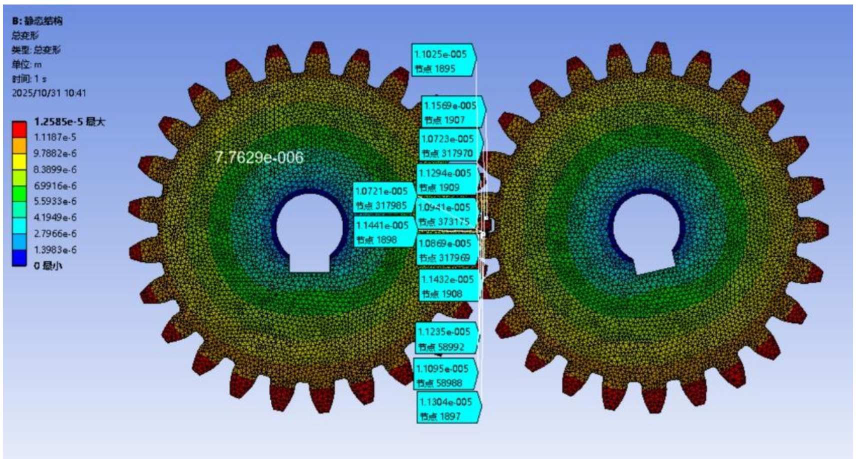

Figure 3: Deformation Diagram of Gear Pair at -75℃

4.2. Analysis of Simulation Experiment Results

It can be seen from the results of the simulation analysis that within the temperature range of -75℃ to +100℃, under the combined effect of thermal expansion and thermal contraction of the gear pair, the tooth surface clearance in the meshing area undergoes slight changes, but the overall meshing relationship remains good without any tooth surface interference. Meanwhile, the coordinates of the nodes and the deformation values in the X, Y and Z directions at these positions are extracted to calculate the clearance of the gear pair after thermal deformation, which is then compared with the formula derived earlier. Similarly, the simulation analysis of another gear pair is completed.

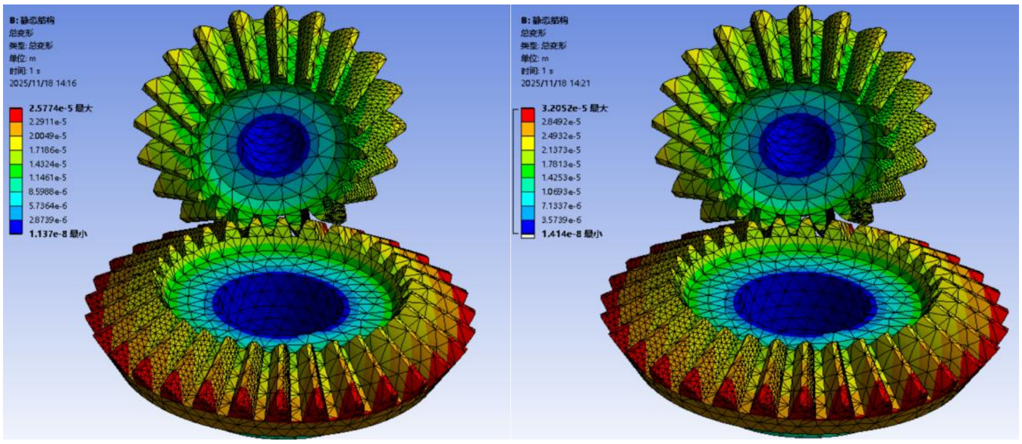

Figure 4: Simulation Result Diagrams of Gear Pair at 100℃ and -75℃

Finally, the normal backlash is calculated by extracting the coordinate parameters of the nodes, and the results are compared with those obtained from the formula calculation, as shown in

Table 1 Gear Pair Backlash

Gear Pair	Normal Backlash	Simulation Analysis at 100℃	Formula Calculation	Simulation Analysis at -75℃	Formula Calculation
1	0.0326	0.0310	0.0328	0.0348	0.0339
2	0.0445	0.0426	0.0439	0.0459	0.0469

Since the simulation analysis model adopted in this study is an idealized three-dimensional geometric model, the influences of factors such as actual machining errors, assembly clearances and tooth surface micro-damage were not fully considered in the modeling process. Therefore, there may be a certain deviation between the simulation results and the real gear meshing state. This paper mainly focuses on the discussion and verification of the rationality of the theoretical model and the reliability of the calculation formula. It can be seen from Table 1 that under different temperature conditions, there is no interference in the meshing state of the gear pair, indicating that the meshing geometric relationship of the system after thermal deformation is still within the safe range. Meanwhile, by comparing the backlash variation obtained from the simulation with the theoretical calculation results, it is found that the two sets of values are close and the variation trends are consistent, which verifies the correctness and engineering applicability of the derived thermal deformation model and normal backlash correction formula. Therefore, this theoretical model can accurately describe the thermal deformation characteristics of solid lubricated gears in the space environment, and provides a reliable theoretical basis for subsequent transmission efficiency analysis and structural optimization design.

5. Conclusion

This study investigates the thermal deformation of solid lubricant gears under extreme space thermal environments. By integrating theoretical modeling, formula derivation, and ANSYS simulation validation, the quantitative relationships among material linear, area, and volume expansion coefficients are clarified, and an equivalent conversion model is established. A theoretical system for gear thermal deformation that considers the coating-substrate coupling effect is constructed, along with a temperature field calculation model for the base circle. Finally, an accurate calculation expression for the normal backlash of gear pairs is derived. The quantitative comparison between simulation and theoretical calculations shows that for Gear Pair 1, at 100°C, the simulated backlash of 0.0310 mm has a relative error of 5.8% compared to the theoretical value of 0.0328 mm. At -75°C, the simulated backlash of 0.0348 mm has a relative error of 2.6% compared to the theoretical value of 0.0339 mm. For Gear Pair 2, the relative errors at 100°C and -75°C are 3.0% and 2.1%, respectively. Under all operating conditions, no meshing interference occurred in the gear pairs, and the numerical values from both methods were close, with consistent variation trends. These results validate the correctness and engineering applicability of the model. The research outcomes fill a theoretical gap in this field and provide key technical support for backlash optimization design, thermal environment adaptability evaluation, and interference risk avoidance in transmission systems of aerospace deployment mechanisms. This work is of significant importance for enhancing the transmission reliability and service life of aerospace equipment.

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