Thermal-structural Coupling Researches on Transmission Shafting of the Intermediate Gearbox of a Certain Helicopter

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Abstract. To study the lubrication performance and structural strength of transmission shafting of the intermediate gearbox of a certain helicopter, applying theories of tribology and heat transfer, thermal-structural coupling simulations of the entire transmission shafting, which mainly consists of a spiral bevel gear pair, two bearings, are performed based on the software platform of ANSYS Workbench. Both steady-state temperature field and stress field are obtained, and influences of rotational speed, torque on the thermal-structural coupling effect are discussed.

1. Introduction

Spiral bevel gear pair and bearings which are the main components of the intermediate gearbox of a certain helicopter, require good working and lubrication conditions. Under the conditions of high speed and heavy load, both the meshing of the bevel gear pair and the relative motion of different parts of the bearings cause friction, which lead the temperature of the transmission system to rise. Excessive temperature will cause severe thermal stress and thermal deformation on gears and bearings, which would easily lead to pitting and gluing of gears and bearings, endangering the stability of the entire transmission system. Therefore, in order to facilitate the analysis and design of the intermediate gearbox and its lubrication system and improve the working performance, it is very important to study the thermo-mechanical coupling effect of the spiral bevel gear transmission shaft system of the intermediate gearbox in the helicopter.

As early as 1995, Handschuh and Kicher [1] investigated the thermal behavior of spiral bevel gears by a finite element method, and verified the accuracy of the method through experiments. Since then, many scholars have conducted researches on the temperature distribution or stress of the geared transmission. Among them, Li and Tian [2] studied the temperature characteristics of a geared transmission, and obtained the influence of friction heat generation on gear bearing capacity and transmission sensitivity. Argyris et al. [3] applied a finite element method to conduct stress analysis; Mehrabi et al. [4] established a finite element model for thermal analysis of spiral bevel gears in helicopter’s main gearboxes, and studied the effects of initial temperature, heat flux and lubricant on system temperature; Sun et al. [5] established geometric models to analyze the heating of bearings under oil air lubrication and spray lubrication conditions respectively, and obtained the influence of different lubrication conditions on the bearing temperature field.

The thermal-stress coupling analysis is also the research focus. Linjamaa et al. [6] developed a multi-physical model to analyze the elastic deformation and thermal deformation of hybrid journal bearings, then evaluated the performance of the bearings. Zhang et al. [7] presented a thermo-elastic-hydrodynamic coupling model to study the thermal characteristics of the spiral bevel gear and obtain the temperature rise of the gear surface. Mei et al. [8] studied the thermal characteristics
of the gearbox, and obtained the relationship between deformation, structural load and thermal load. Liu et al. [9] established a thermo-elastic coupled finite element model to study the thermal deformation of the spiral bevel gear at different rotational speeds; Lu et al. [10] studied the thermal characteristics of the main shaft, and obtained the thermal deformation law under different rotational speeds and lubrication conditions.

In this work, the spiral bevel gear drive shaft system of the intermediate gearbox of a helicopter is taken as the research object, and following investigations are performed: firstly, the thermal-structural coupling simulation of the drive shaft system is carried out based on the finite element analysis method, then the steady temperature field and stress field distribution are obtained. Secondly, the influence of speed, torque and lubricating oil input temperature on the thermo-coupling effect is studied. Finally, the simulation results of steady-state temperature field is verified by experiments.

2. Determination of Boundary Conditions for Thermal Analysis

The heat generated by the transmission shaft of the intermediate gearbox mainly comes from load loss and non-load loss. The load loss mainly includes the friction power loss of the spiral bevel gear and the bearing. The transfer of heat in the transmission shaft system and its surroundings is mainly through three modes: heat conduction, heat convection and heat radiation [11]. Due to the heat transfer effect of heat radiation is the smallest, this paper ignores.

2.1. Transmission Shaft Heat Generation Analysis

2.1.1. Friction power loss of spiral bevel gear

Sliding friction is the main source of heat generated by spiral bevel gear transmission. According to the literature, the average heat flux density Q generated by the sliding friction of the tooth surface is

\[ Q = f_0 \sigma_0 v_s \]  

(1)

Among this, \( f_0 \) is tooth surface sliding friction coefficient; \( \sigma_0 \) is the maximum contact stress of meshing zone; \( v_s \) is the relative sliding speed of the driving and driven wheels at the meshing point.

The average heat flux density at the meshing point into the main and driven wheels \( q_1, q_2 \) is

\[
\begin{align*}
q_1 &= \frac{b \beta_0 \omega_1}{2 \pi v_1} \\
q_2 &= \frac{b(1 - \beta_0) \omega_2}{2 \pi v_2}
\end{align*}
\]  

(2)

Among this, \( b \) is the half width of contact at the meshing point; \( \beta_0 \) is the heat partition coefficient; \( \omega_1, \omega_2 \) is the angular velocity of the driving and driven wheels; \( v_1, v_2 \) is the tangential velocity at the meshing point of the driving and driven wheels.

2.1.2. Friction power loss of the bearing

This paper uses the Palmgren calculation model to calculate the frictional heat of the bearing. The heat generated by the friction of the bearing \( H \) is

\[ H = \frac{\pi n M}{30} \]  

(3)

Among this, \( n \) is the bearing speed; \( M \) is the bearing friction torque, \( M = M_l + M_v, M_l \) is friction torque associated with bearing load; \( M_v \) is torque associated with the properties of the lubricating oil.

2.2. Analysis of Convective Heat Transfer of Transmission Shaft

Convective heat transfer at the end face of a spiral bevel gear can be simplified into a convective heat transfer model of a rotating cone. The calculation formula \( h_s \) of the end face heat transfer coefficient in this state is
Among this, \( N_u \) is Nusslet number; \( \lambda_v \) is the heat transfer coefficient and kinematic viscosity of the fluid on the end face; \( \alpha_b \) is half cone angle of spiral bevel gear.

According to the literature [12], the convective heat transfer coefficient of spiral bevel gear meshing surface can be expressed as:

\[
\frac{h_u}{\lambda} = N_u \lambda \sqrt{\frac{\omega}{v_f \sin \alpha_b}}
\]  
(4)

Among this, \( N_u \) is Nusslet number; \( \lambda_v \) is the heat transfer coefficient and kinematic viscosity of the fluid on the end face; \( \alpha_b \) is half cone angle of spiral bevel gear.

According to the literature [12], the convective heat transfer coefficient of spiral bevel gear meshing surface can be expressed as:

\[
h_b = 1.418 \left( \frac{v_f c_f \rho_f}{\lambda z} \right)^{\frac{1}{3}} \sqrt{\frac{\lambda c_f \rho \mu}{3\tau}}
\]  
(5)

Among this, \( c_f \rho_f \) is specific heat and density of lubricating oil on the mating surface; \( z \) is the number of gear teeth.

The convective heat transfer coefficient of the root surface, the tooth top surface and the non-meshing tooth surface can be approximated within the range of 1/3 to 1/2 of the end surface convective heat transfer coefficient.

2.3. Convective Heat Transfer of Bearing

This paper uses Harris calculation model to calculate the convective heat transfer of tapered roller bearings. The convection heat transfer coefficient \( h_a \) of bearing is

\[
h_a = 0.332 \frac{k}{X} \rho_f^\frac{1}{3} R_e^\frac{1}{2}
\]  
(6)

Among this: \( k \), \( \rho_f \) is the thermal conductivity and Prandtl number of lubricating oil; \( R_e \) is Reynolds number.

3. Simulation Analysis

3.1. Establishment of Finite Element Model of Transmission Shafting

The transmission shafting of the intermediate gearbox of a certain helicopter includes a input spiral bevel gear shaft, an output spiral bevel gear and a pair of reverse mounted tapered roller bearings of the same type, as shown in Fig. 1. The basic parameters of the spiral bevel gears and the tapered roller bearings are shown in Table 1 and 2. The physical parameters of the drive shaft system are as follows:

The rotating speed of driving wheel is 4455 r/min, the torque is 200 N·m, and the lubricating oil’s input temperature and ambient temperature are both 50 °C. According to the analysis and calculation of the force of the tapered roller bearings, the bearing 1 is “relaxed”, the radial force is 20342 N, the axial force is 5983 N; the bearing 2 is “compressed” and the radial force is 20342 N, the axial force is 8429 N.

Figure 1. Three-dimensional model of a transmission shaft system
Table 1. Basic parameters of spiral bevel gear

|                     | Number of teeth $Z$ | Modulus $m$/(mm) | Tooth width $B$/(mm) | Helix angle $\beta$/(°) | Pressure angle $\alpha$/(°) | Axis angle $\Sigma$/(°) | Rotation    |
|---------------------|---------------------|------------------|----------------------|-------------------------|--------------------------|-------------------------|-------------|
| Driving wheel       | 39                  | 3.75             | 26                   | 35                      | 20                       | 128                     | Left-handed |
| Driven wheel        | 51                  | 3.75             | 26                   | 35                      | 20                       | 128                     | Right-handed|

Table 2. Basic parameters of tapered roller bearings

| Inside diameter $d$/(mm) | Outer diameter $D$/(mm) | Width $T$/(mm) | Number of rolling elements | Dynamic load rating $C$/(kN) | Static load rating $C_0$/(kN) |
|--------------------------|-------------------------|---------------|----------------------------|-----------------------------|-------------------------------|
| 60                       | 95                      | 17            | 12                         | 57.57                       | 76.12                         |

3.2. Finite Element Analysis of Steady-state Temperature Field

In the steady state thermal analysis module of ANSYS Workbench, divide all entities of the imported driveshaft 3D model for free meshing. In order to improve the accuracy of simulation calculation, the mesh of the gear meshing area and the bearing roller and the inner and outer raceway contact areas are refined. The total number of nodes divided is 1080122, and the number of units is 656359. The divided transmission shaft system mesh model is shown in Fig. 2.

Apply the thermal analysis boundary conditions to the corresponding surfaces of the drive shaft system model, and the steady state temperature field of the drive shaft system is obtained by simulation, as shown in Figure 3.

As can be seen from Fig.3:

1. In the entire spiral bevel gear transmission shaft system, the temperature of the outer ring raceway and the roller contact area of the bearing 2 is the highest, reaching 95.49 °C. The main reason is that under high-speed and heavy-load conditions, the bearing 2 is in a state of being "pressed". The frictional torque is large, resulting in a large frictional heat generation, while the small size results in a weak convective heat transfer capability.
(2) For the helical bevel gear pair, the gear meshing zone has the highest temperature, reaching 78.84 °C. The main reason is that the friction of the gear meshing region is the main source of heat, and the convective heat transfer coefficient of this region is small.

3.3. Thermal-structural Coupling Analysis

This paper uses the sequential coupling method to analyze the thermal-structural coupling of spiral bevel gear transmission shafting. The steady-state temperature field of the drive shaft system calculated in the thermal analysis module is used as a boundary condition and imported into the structural analysis module for simulation calculation.

The simulation results show the stress field of the drive shaft under the thermal-structural coupling effect, as shown in Fig. 4. In order to facilitate the comparative analysis, the stress field of the drive shaft system is calculated without considering the temperature, as shown in Fig. 5.

![Figure 4. Stress field under thermal-structural coupling.](image1)

![Figure 5. Stress field without considering temperature.](image2)

From Fig. 4 and 5, it can be seen that it is in accordance with the actual force of the transmission shaft of the intermediate gearbox under high speed and heavy load conditions, while the maximum equivalent stress of the drive shaft system also appears at the gear meshing position. After the application of the thermal load, the maximum stress at the gear meshing position increases by 10.68 MPa, an increase of 1.4%; the maximum stress of the bearing pair increases by 239.60 MPa, an increase of 52.5%. The main reason is that the thermal stress caused by temperature and the stress generated by the load overlap each other, so that the stress increases, the larger temperature rises, and the greater the temperature rise, the greater the thermal stress increase. Therefore, for bearings, the thermal stress cannot be ignored.

4. Analysis of Influencing Factors

Speed, torque are the main factors affecting the thermal-structural coupling effect of the spiral bevel gear drive shaft. This paper mainly analyzes and discusses the impact of these three factors.

4.1. Analysis of the Influence of Speed

In the fixed working condition where the driving wheel torque is 200 N·m and the lubricating oil input temperature is 50 °C, the gear shaft rotation speed is set to be 4455 r/min, 5455 r/min, 6455 r/min,
7455 r/min, 8455 r/min. The maximum temperature and maximum stress of the shafting system are shown in Figs. 6 and 7.

![Figure 6](image1.png) ![Figure 7](image2.png)

Figure 6. Maximum temperature at different speeds  
Figure 7. Maximum stress at different speeds.

It can be seen from the variation law of Fig. 6:

1. When the speed of the gear shaft is in the range of 4455 ~ 8455 r/min, the maximum temperature of the bearing pair is always higher than the gear pair.
2. The maximum temperature of the bearing pair and gear pair increases with the increase of the rotational speed, because the increase of the rotational speed causes the frictional heat generation and the convective heat transfer coefficient of the bearings and gears increase.
3. When the rotational speed increased by 89.8%, the maximum temperature of the bearing pair and the gear pair increased by 38.3% and 18.9% respectively, indicating that the influence of the rotational speed on the bearing temperature field is more significant than that on the gear temperature field.

It can be seen from the variation law of Fig. 7:

1. The maximum stress of the bearing pair increases significantly with the increase of the rotational speed. The main reason is that as the rotational speed increases, the maximum temperature of the bearing pair increases significantly, causing the thermal stress generation also to increase significantly. Besides, the maximum stress of the gear pair gradually increases with the increase of the rotational speed.
2. When the rotational speed increases by 89.8%, the maximum stress of the bearing pair and the gear pair increases by 28.6% and 0.6% respectively, indicating that the influence of the rotational speed on the bearing stress field is more significant than that on the gear stress field.

4.2. Analysis of the Influence of Torque

Under the fixed working conditions of the driving wheel speed of 4455 r/min and the lubricating oil input temperature of 50 °C, the transmission is calculated when the driving wheel torque is 150 N·m, 200 N·m, 250 N·m, 300 N·m, 350 N·m. The maximum temperature and maximum stress of the shafting system are shown in Figs. 8 and 9.
Figure 8. Maximum temperature at different torques.

Figure 9. Maximum stress at different torques.

It can be seen from the variation law of Fig. 8:

1. When the driving wheel torque is in the range of 150 – 350 N·m, the maximum temperature of the bearing pair is higher than that of the gear pair.
2. The maximum temperature of the bearing pair increases with the increase of the driving wheel torque. The main reason is that as the driving wheel torque increases, the bearing friction torque increases gradually, so that the friction heat generation increases, causing the maximum temperature of the bearing to increase. The maximum temperature of the gear pair also increases with the increase of the driving wheel torque. The main reason is that the increased driving wheel torque increases gear contact pressure, which in turn leads to an increase in the heat flow density and cause the maximum temperature of the gear to increase.

It can be seen from the variation law of Fig. 9:

1. When the torque is in the range of 150 to 175 N·m, the maximum stress of the bearing pair is larger than that of the gear pair. When the torque is in the range of 175 to 350 N·m, the maximum stress of the gear pair is larger than that of the bearing pair.
2. The maximum stress of the bearing pair gradually increases with the increase of the driving wheel torque. The main reason is that as the driving wheel torque increases, the maximum temperature of the bearing pair gradually increases, so that the generated thermal stress gradually increases. At the same time, the radial and axial forces of the bearing increases gradually, so that the load stress is also gradually increased. The maximum stress of the gear pair increases significantly with the increase of the driving wheel torque. The main reason is that as the driving wheel torque increases, the maximum temperature of the gear pair increases significantly, so that the generated thermal stress increases significantly. At the same time, the contact pressure of the gear pair is significantly increased, so that the load stress is also significantly increased.

5. Conclusion

1. Under the condition of high speed and heavy load, the temperature of the contact area between the outer ring raceway and the roller of the bearing 2 in the drive shaft system is the highest, thus the lubrication here needs to be strengthened.
2. Under the combined influence of load stress and thermal stress, the stress of the drive shaft system is mainly distributed at meshing zone of the spiral bevel gears, the bearing roller support and gear shaft journal. The stress at the gear meshing is the largest, and the deformation is most likely to occur here, thus structural optimization can be appropriately performed to improve stress concentration.
3. As the rotational speed, driving wheel torque increase, the maximum temperature and maximum stress of the bearing and gear will also increase. The influence of the rotational speed on the bearing temperature field and the stress field is more significant. The influence of driving wheel torque on the gear temperature field and stress field is more significant. The effect of temperature rise on bearing stress field is more significant than that on gear stress field.
6. References

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