Simulation calculation of temperature field of gearbox in straddle monorail train

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Abstract. For a straddle-type monorail train gearbox, analyze the heating mechanism and heat transfer mode of each element in the gearbox, establish a heat transfer model of the gearbox, use empirical formula to calculate the heating power of each heating element, and use the flow field analysis to monitor it. Accurately calculate the convective heat transfer coefficient for the volume fraction of oil and gas in each cavity. Using the method of finite element simulation, the influence of different oil injection and large spiral bevel gear steering on the temperature field distribution of the gearbox is calculated and analyzed. The results show that when the large spiral bevel gear turns for a certain amount of time, the maximum temperature of each part of the gearbox at high oil level is higher than that at low oil level; When the amount of oil injection is constant, the temperature of each part in the reverse rotation of the large spiral bevel gear is higher than that in the forward rotation. This article provides the main reference for the prediction of the temperature field of the gearbox of the straddle monorail train and the improved design of the lubricating oil passage in the gearbox.

1. Introduction

Straddle-type monorail trains have been introduced and used by more and more cities with medium traffic demand due to their advantages of adapting to complex terrain, strong climbing ability, and low cost. Gearbox is an important part of monorail train transmission system. Excessive temperature will cause problems such as poor lubrication and cooling, bearing seizure, tooth surface gluing, and gearbox seal failure. Therefore, studying the temperature field of the gearbox is of great significance to the normal operation of the train and the improved design of the lubricating oil passage inside the gearbox.

At present, the main methods of gearbox thermal analysis are thermal network method\(^{[1-2]}\), experimental method\(^{[3]}\) and finite element method\(^{[4-5]}\). Although the thermal network method is simple, the calculation accuracy is not high; the experimental method has a long research period and consumes a lot of manpower and material resources; The finite element method can achieve higher accuracy in the calculation, better meet the needs of the project, and at the same time more intuitively observe the temperature cloud images of the parts of the gearbox.

The internal structure of the straddle-type monorail train gearbox is very complicated. As shown in Figure 1, the first stage adopts spiral bevel gear transmission, and the second stage adopts cylindrical helical gear transmission. The input gear shaft is cantilevered by a four-point angular contact ball bearing (number 1) and two cylindrical roller bearings (number 2, 3). The intermediate shaft is supported by a four-point angular contact ball bearing (number 6) and 2 cylindrical roller bearings (number. 4, 5), and an oil slinger (number. 9) is installed on the shaft. The output shaft also adopts
straddle support, supported by two tapered roller bearings (number 7, 8)\textsuperscript{[3]}

![Fig 1 Schematic diagram of gearbox transmission](image)

The complicated internal structure makes it very difficult to measure the temperature of the internal parts of the gearbox with experimental methods. However, there are few reports on the thermal analysis of the gearbox of straddle-type monorail trains using the finite element method.

In this paper, a straddle-type monorail train gearbox is used as the research object, combined with the volume fraction of oil and gas in each cavity obtained by flow field analysis, to accurately calculate the convective heat transfer coefficient of each component. The finite element method is used to calculate and analyze the temperature distribution law of the gearbox under different working conditions with the consideration of the external wind field, providing a theory for the prediction of the gearbox temperature field of the straddle monorail train and the improved design of the internal lubricating oil flow channel support. The specific simulation conditions are shown in Table 1. The oil level line of the gear box and the definition of the forward rotation of the large spiral bevel gear are shown in Figure 2.

| Working condition | Oil filling amount(oil level) | Input torque/Nm | Input speed /rpm | Big bevel gear steering |
|-------------------|-----------------------------|----------------|-----------------|------------------------|
| 1                 | High oil level              | 550            | 3035            | Forward               |
| 2                 | High oil level              | 550            | 3035            | Reverse                |
| 3                 | Low oil level               | 550            | 3035            | Forward               |
| 4                 | Low oil level               | 550            | 3035            | Reverse                |

![Fig 2 Gearbox oil volume and direction of forward rotation](image)

2. Gearbox heat production and heat transfer analysis
The main sources of heat generated during the working process of the gearbox are the frictional heat generated by gear meshing, the frictional heat generated by the bearing, the heat generated by oil stirring, and the heat generated by wind resistance.
2.1 Gearbox heat generation analysis

2.1.1 Heat produced by gear meshing. The first-stage gear is a spiral bevel gear, and its spatial meshing theory is relatively complicated, and it is difficult to directly calculate its frictional power loss. In this paper, the spiral bevel gear is equivalent to a cylindrical helical gear, and a power loss model equivalent to the original gear is established[6]. The equivalent gear pitch radius is $R\tan\delta_1$, $R\tan\delta_2$, the number of teeth are $z_1/\cos\delta_1$, $z_2/\cos\delta_2$, the angular velocity is $\omega_1\cos\delta_1$, $\omega_2\cos\delta_2$, the helix angle is $\beta$ cylindrical helical gear pair, where $R$ is the cone of the original spiral bevel gear Distance; $\delta_1$ and $\delta_2$, $z_1$ and $z_2$, $\omega_1$ and $\omega_2$ are the cone angle, number of teeth, and angular velocity of the two spiral bevel gears, respectively.

There are three common methods to study the heat generation of gear pairs: Townsend method, Anderson method and the calculation method involved in the standard ISO/TR 14179. In this paper, the Anderson and Loewenthal formula[5] is used to calculate the meshing heating power of the primary equivalent gear and the secondary gear.

The formula for calculating the average normal load of the gear is equation (1):

$$F_n = \frac{T}{r\cos\alpha\cos\beta}$$

(1)

In the formula, $T$ is the torque; $r$ is the gear pitch radius; $\alpha$ is the normal pressure angle; $\beta$ is the helix angle.

The calculation formulas for the average rolling and sliding speeds of gears are formula (2) and formula (3):

$$V_s = 0.026118n_1g_s\left(z_1+z_2\right)\left(z_1/z_2\right)$$

(2)

$$V_r = 0.2094n_1\left[r\sin\alpha - 0.125g_s\left(z_1+z_2\right)/z_2\right]$$

(3)

In the formula, $n_1$ is the input shaft speed; $g_s$ is the length of the meshing line.

The calculation formulas for the average sliding and rolling heating power are formula (4) and formula (5):

$$P_s = \frac{\gamma f h V_s}{1000}$$

(4)

$$P_r = \frac{\gamma 90000 V_r h_o e}{\cos\beta}$$

(5)

In the formula, $\gamma$ is the coefficient of converting friction energy into heat energy, which is taken as 0.95; $f$ is the friction coefficient; $b$ is the tooth width; $h_o$ is the average oil film thickness; $e$ is the coincidence degree.

The total heating power of gear meshing can be expressed as formula (6):

$$\sum P = P_{s1} + P_{r1} + P_{s2} + P_{r2}$$

(6)

In the formula, $P_{s1}$ and $P_{r1}$ are the average sliding and rolling heating power of the first-stage gear respectively; $P_{s2}$ and $P_{r2}$ are the average sliding and rolling heating power of the second-stage gear respectively.

2.1.2 Heat generated by bearing friction. In this paper, Palmgren formula [4] is used to calculate the frictional heat generation of the bearing.

The frictional moment $M$ of the bearing can be expressed as formula (7):

$$M = M_0 + M_1$$

(7)

In the formula, $M_0$ is the friction torque related to bearing type, lubricating oil properties and speed; $M_1$ is the friction torque related to bearing load.

When $v n \geq 2000$, $M_0$ can be expressed as formula (8):

$$M_0 = 10^{-7} f_0 \left(v n\right)^{2/3} D_m^3$$

(8)

When $v n < 2000$, $M_0$ can be expressed as formula (9):

$$M_0 = 160 \times 10^{-7} f_0 D_m^3$$

(9)

In the formula, $D_m$ is the average diameter of the bearing $D_m = 0.5(d + D)$; $n$ is the speed; $f_0$ is
the coefficient related to the bearing type and lubrication; \(v\) is the kinematic viscosity of the lubricating oil.

\[ M_1 = f_1 F_1 D_m \]  

(10)

In the formula, \(f_1\) is the coefficient related to the bearing type and load; \(F_1\) is the calculated load to determine the friction moment of the bearing, and the cylindrical roller bearing \(F_1 = F_r\) (radial force).

The bearing friction heating power can be expressed as formula (11):

\[ P = 1.047 \times 10^{-4} n M \]  

(11)

2.1.3 Heat produced by stirring oil. The gear box adopts splash lubrication, and the large spiral bevel gear and the oil slinger are responsible for stirring the oil. The method of Soo H. Jeon[8] is used to calculate the heating power of stirring oil for spiral bevel gears, and the formulas are equations (12) and (13):

\[ T = 0.5C \rho_o \omega^2 r^2 b S \]  

(12)

\[ C = 10 \left( \frac{h}{r} \right)^{0.1786} \left( \frac{V_o}{r^3} \right)^{-0.2195} \] \[ Re^{-0.4169} \times Fr^{-0.4482} \left( \frac{v}{v_a} \right)^{-0.1777} \]  

(13)

In the formula, \(C\) is the resistance coefficient; \(\rho_o\) is the density of the lubricating oil; \(\omega\) is the angular velocity; \(S\) is the gear oil immersion area; \(h\) is the oil immersion depth; \(V_o\) is the lubricating oil volume; \(v_a\) is the kinematic viscosity of air; \(Fr\) is the Frode number; \(Re\) is the Reynolds number.

The method mentioned in ISO/TR 14179[9] is used to calculate the heating power of stirring oil on the side of the gear and the oil slinger. The formula is formula (14):

\[ P = \frac{1.4744f_g \cdot v \cdot n \cdot 3 \cdot D \cdot 5.7}{A_g \cdot 10^{26}} \]  

(14)

In the formula, \(f_g\) is the gear infiltration factor, when the components are fully immersed, \(f_g = 1\); \(D\) is the outer diameter of the components; \(A_g\) is the configuration constant, which is 0.2.

The formula for the total heating power of stirring oil is equation (15):

\[ \sum P = P_C + P_S \]  

(15)

In the formula, \(P_C\) is the heating power of gear stirring oil; \(P_S\) is the heating power of oil slinger stirring oil.

Wind resistance loss accounts for a small proportion of the total loss of this gearbox and can be ignored. The heating power of each working condition is summarized in Table 2.

| Working condition | First gear /W | Secondary gear /W | Bearing 1/W | Bearing 2/W | Bearing 3/W | Bearing 4/W | Bearing 5/W | Bearing 6/W | Bearing 7/W | Bearing 8/W | Stir oil /W |
|-------------------|---------------|-------------------|-------------|-------------|-------------|-------------|-------------|-------------|-------------|-------------|-------------|
| 1                 | 665.2         | 600.8             | 172.1       | 109.8       | 143         | 22.1        | 25.6        | 52.5        | 33.3        | 43.5        | 219         |
| 2                 | 665.2         | 600.8             | 172.1       | 109.8       | 143         | 22.1        | 25.6        | 52.5        | 33.3        | 43.5        | 219         |
| 3                 | 665.2         | 600.8             | 172.1       | 109.8       | 143         | 22.1        | 25.6        | 52.5        | 33.3        | 43.5        | 171         |
| 4                 | 665.2         | 600.8             | 172.1       | 109.8       | 143         | 22.1        | 25.6        | 52.5        | 33.3        | 43.5        | 171         |

2.2 Gearbox heat transfer analysis

The heat generated during the operation of the gearbox is mainly transferred through heat conduction, heat convection, and heat radiation[10]. The contact parts of gears, bearing inner and outer rings, box bodies and the oil and gas mixture are considered according to convection heat transfer, and the contact parts between other solid components are considered according to heat conduction. Due to the relatively low temperature of the gearbox, heat radiation is not considered here.

2.2.1 Physical parameters of two-phase flow. Use the method in[11] to determine the physical parameters \(\zeta_f\) of the oil-gas mixture, and the calculation formula is equation (16):

\[ \zeta_f = [\alpha_a + (1 - d)\alpha_o] \zeta_a + \alpha_o d \zeta_o \]  

(16)
In the formula, $\zeta_a$ and $\zeta_o$ are the performance parameters of air and lubricating oil respectively; $\alpha_a$ and $\alpha_o$ are the volume fractions of air and lubricating oil respectively, $\alpha_a + \alpha_o = 1$; $d$ is the proportional coefficient affected by experimental factors.

### 2.2.2 Convection heat transfer coefficient of gear tooth surface.

Use Handschuh’s method\(^{[12]}\) to calculate the convective heat transfer coefficient of the gear tooth surface.

The formula for calculating the heat transfer coefficient of the first-stage spiral bevel gear tooth face-to-flow is equation (17):

$$h_1 = 0.228 R_{ef}^{0.731} P_r^{1/3} \lambda_f \left[ \frac{v_f}{\cos \theta} \right]^{-0.5}$$

The formula for calculating the face-to-flow heat transfer coefficient of the two-stage cylindrical helical gear tooth is equation (18):

$$h_2 = \frac{0.228 R_{ef}^{0.731} P_r^{1/3} \lambda_f}{L}$$

In the formula, $R_{ef}$, $P_r$, $\lambda_f$, $v_f$ are the Reynolds number, Prandtl coefficient, thermal conductivity, and kinematic viscosity of the oil and gas mixture respectively; $\delta$ is the cone angle of the spiral bevel gear; $L$ is the characteristic length (take gear indexing Circle diameter).

### 2.2.3 Convection heat transfer coefficient of gear end face, bearing end face and oil slinger face.

The gear end surface, bearing end surface and oil slinger surface are regarded as rotating disc surfaces. The calculation formula of convective heat transfer coefficient\(^{[13]}\) is formula (19):

$$h = 0.308 \lambda_f (m + 2)^0.5 P_r^{0.5} (\omega/v_f)^{0.5}$$

In the formula; $m$ is the radial temperature distribution exponential constant, taking 2.

### 2.2.4 Convection heat transfer coefficient of the inner wall of the box.

The calculation formula of the convective heat transfer coefficient between the inner wall of the box and the oil-gas mixture\(^{[14]}\) is formula (20):

$$h = 0.037 R_{ef}^{A/5} P_r^{1/3} \lambda_f / x$$

In the formula, $x$ is the characteristic length of the inner cavity of the box.

### 2.2.5 Convection heat transfer coefficient of the cylindrical surface of the rotating shaft.

The calculation formula\(^{[15]}\) for the convective heat transfer coefficient between the cylindrical surface of the intermediate shaft and the output shaft and the oil-gas mixture is equation (21):

$$h = \frac{0.133 R_{ef}^{2/3} P_r^{1/3} \lambda_f}{D}$$

Using the above formulas, the convective heat transfer coefficients at different positions of the gearbox under different working conditions are calculated.

### 3. Gearbox thermal analysis model

#### 3.1 Gearbox modeling

Use SolidWorks software to model and simplify the three-dimensional gearbox, remove unnecessary rounded corners and chamfers and other unimportant structural factors. Set the wind field outside the box to simulate the environmental conditions of the gearbox during operation. The external wind field model is shown in Figure 3. Import the processed model into Fluent, and use the Poly-Hexcore meshing method of Fluent Meshing to generate the mesh of the gearbox. The grid model, the number of grids is 468 5,042, the number of nodes is 628 7794, the gearbox grid model is shown in Figure 4.
3.2 Solution settings and material parameters

Use the above model, heating power and convective heat transfer coefficient to carry out steady-state simulation calculation of temperature field, adopt 3D solver double-precision parallel calculation, select pressure base, steady-state solution, use RNG k-epsilon turbulence model, standard wall equation, use SIMPLEC Algorithm discrete solution.

The performance parameters of the gearbox components are shown in Table 3.

| Parts         | Materials   | Density /Kg/m³ | Specific heat /J/(Kg·K) | Heat conduction coefficient /W/(m·K) |
|---------------|-------------|----------------|--------------------------|--------------------------------------|
| Gear          | 17CrNiMo6   | 7850           | 0.44×10³                 | 50.2                                 |
| Drive shaft   | 30CrNi3     | 7850           | 0.44×10³                 | 50.2                                 |
| Gearbox shell | AlSi7Mg0.3  | 2250           | 0.21×10³                 | 231                                  |
| Bearing       | AlSi7Mg0.3  | 2250           | 0.39×10³                 | 45.5                                 |

4. Simulation results and analysis

When the external wind speed is 15m/s and the ambient temperature is 25°, the maximum temperature of each component under each working condition is shown in Table 4. In the table, T_max represents the maximum temperature.

| Working condition | Gear box /T_max/K | First gear /T_max/K | Secondary gear /T_max/K | Input shaft bearing /T_max/K | Intermediate shaft bearing /T_max/K | Output shaft bearing /T_max/K |
|-------------------|-------------------|---------------------|------------------------|----------------------------|-----------------------------------|-----------------------------|
| 1                 | 361.349           | 392.139             | 375.34                 | 362.901                    | 365.749                           | 356.018                     |
| 2                 | 362.637           | 400.452             | 379.09                 | 364.85                     | 367.607                           | 357.79                      |
| 3                 | 355.409           | 382.57              | 374.613                | 357.033                    | 362.287                           | 346.89                      |
| 4                 | 356.826           | 395.208             | 376.777                | 359.524                    | 364.065                           | 349.817                     |

It can be seen from Table 4 that the maximum temperature of each component in working condition 2 is higher than other working conditions, which is 400.452K. The highest temperature is located on the meshing tooth surface of the first-stage spiral bevel gear driving wheel. The temperature distribution cloud diagram of main components is shown in Figure 5. The temperature distribution law of the gearbox is that the gear mesh and the friction between the bearing rollers and the inner and outer rings are the center of the heat source, and it gradually decreases outward.

The temperature cloud diagram of the box is shown in Figure 6. The windward side of the box and the end cover of the output shaft have lower temperatures; the bearing seat of the input shaft and the intermediate shaft end cover are on the leeward side, and the heat dissipation effect is not good, which is affected by the internal heat source. The temperature is high; there is lubricating oil at the bottom of the tank, and the temperature is high.
Fig 5 Temperature cloud map of main parts

Comparing and analyzing the data in Table 4, it can be seen that when the gear steering is constant, the maximum temperature of each component is higher when the oil level is high than when the oil level is low. It also increases accordingly; when the oil level is constant, the maximum temperature of each component is higher when the gear is reversed than when the gear is rotating. The reason is that when the large bevel gear rotates forward, the oil is stirred more fully and a large amount of lubricating oil is brought in. To the first gear meshing area and the input shaft bearing, a better cooling effect is achieved.

5. Conclusion
(1) Under the same oil immersion depth, the cooling and lubrication effect of the large spiral bevel gear is better than that of the reverse rotation when the large spiral bevel gear is rotated. This should be fully considered in the design of the runner structure.
(2) When the spiral bevel gear turns to a certain degree, the internal equilibrium temperature of the gearbox under high oil level conditions is higher than the temperature at low oil level.
(3) The finite element analysis method used in this article can effectively predict the distribution of the temperature field inside the gearbox, and obtain the temperature of the area that is difficult to monitor by conventional temperature measurement methods, which is for the reliable operation of the gearbox and the structure of the lubricating oil flow channel inside the gearbox. The improved design provides effective theoretical support.

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