Thermal-Mechanical Coupling Finite Element Simulation and Experimental Study of Disk Brake Friction Pair

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Keywords: Friction pair, Thermal-mechanical coupling, Finite element analysis, Temperature field, Brake.

Abstract. This article mainly aims at the study of dry disc brake friction pair thermal-mechanical coupling. Through the calculation of heat source model, the mechanism of produce heat and friction heat production are obtained. Through the study of rough surface contact theory, the contact thermal resistance model of contact surface is established. The distribution of braking energy in two planes is calculated by working condition of tracked vehicle. An abaqus finite element analysis model is established under the influence of rough surface contact thermal resistance on friction pair. It is compared with the traditional finite element simulation. The second heat source heat distribution is considered in new model. The more accurate calculation results can be obtained by consideration of surface thermal resistance effect in thermal-mechanical coupling simulation. In this way, the temperature distribution of the riveted friction pair is obtained. The accuracy of the finite element model is verified by the brake bench test, which provides a meaningful reference for the design of the heat resistance of the brake.

Introduction

In most engineering applications, the calculation model of the surface temperature of friction pairs is generally adopted by Block. [1] The concept of flash temperature is proposed. Then the theory was developed, and the mathematical model of the rectangular shape of the mobile heat source on the semi-infinite surface was established [2,3].

Due to the Block - Jaeger model limited to a single point heat source or line heat source on the semi-infinite surface sliding, and for the friction brake, brake disc is limited thickness and friction lining, from micro point of view, friction heat is composed of contact area, many micro convex peak contact so after research focuses on rough surface contact to the influence of the temperature field. Such as Ling [4], A random model of surface topography is used to estimate transient temperature rise of sliding surface. The sliding friction local temperature rise of the rough surface with fractal characteristics is studied by Wang [5]. Tian and Kennedy’s study [6]. The actual sliding contact are more discrete contact and sliding pair are limited to the size of the object, flashing at contact micro convex body in addition to the local temperature rise, and the role of the nominal surface temperature rise. The frictional heat generated between the friction lining and the surface of the friction brake is not evenly distributed on the sliding surface, but depends on the distribution of local stresses [7,8]. The influence of temperature of two contact bodies on temperature rise of contact surface is considered by Barber [9], Fridrich [10] et al. calculated the anisotropy factors of the physical properties of composite materials, and proposed the definition of Pelet constant of composite materials.

The above research has continuously improved the simulation process of the friction pair temperature field, making the calculation result more and more close to the real situation, but there are still some problems. Though finite element method to simulate heat transfer on the surface of the ideal situation, but for rough surface because of the influence of the thermal resistance effect,
can make between the friction lining and dual temperature difference increase, in the name of the so you have to modify the above model. The experimental verification method of temperature field analysis and the lack of data make the research of the disc brake friction pair temperature field mostly stay in the theoretical research stage. To solve above problems, in this paper a method of combining the theory and experiment through the real surface morphology characteristic parameters of acquisition and representation is established.

Modeling

Rough Surface Contact Mechanism

State diagram as shown in figure 1 for rough surface contact, friction heat through the heat flow in the form of q respectively into fluctuation two parts of friction pair, the distribution coefficient for r, two parts temperature of friction pair of $T_1$ and $T_2$ respectively, but in the contact interface temperature generally is continuous, called contact temperature $T_c$. There is real contact area and nominal contact area in the contact area, which is related to rough surface property. The real contact area is the area where there is real solid contact, which is represented by $A_r$. The nominal contact area contains both real and non-contact areas.

Due to the existence of friction, the surface with higher hardness of the two objects in contact will be pressed into the lower hardness surface, resulting in elastic deformation or plastic deformation. Like Fig.2, The mean square root values of two surface roughness are $\sigma_1$ and $\sigma_2$ respectively. $h$ is the distance between two surface centerlines. Their contact condition can be converted to a smooth surface and another rough surface which is $\sigma = \sqrt{\sigma_1^2 + \sigma_2^2}$.

In theory the contact thermal resistance calculation of rough surface, the key is the actual contact points of the given pressure, the contact point average heat channel radius, with a single point contact thermal resistance of the pressure formula obtained the general contact thermal resistance. Fig.3 shows that the contact heat resistance of the rough surface can produce shrinkage effect, and the thermal resistance model of the contact point can’t be calculated using the single point thermal resistance model.
Figure 3. Rough contact surface heat flow shrinkage.

The ratio of the surface temperature difference $\Delta T_c$ and the average heat flux density $q$ on the contact surface:

$$R_c = \frac{\Delta T_c}{q}$$

By simplifying the contact heat resistance calculation formula under single point contact, the formula is given:

$$R_c = \frac{A}{\pi \alpha \lambda_s} f(\epsilon)$$

(1)

(2)

However, the rough contact surface is different from single point contact, and the rough surface is equivalent to multiple single point contact thermal resistance parallel. The mean radius of contact points is:

$$R_s = \sqrt{A_s / \pi} / ds$$

(3)

where $A_s$ is actual contact area. Contact radius and contact force calculation formula:

$$R_{\text{eff}} = \sqrt{R \cdot \delta}$$

(4)

$$P(\delta) = A \cdot ds \cdot P_c(\delta) = A \cdot ds \frac{4}{3} E^3 R^{3/2} \delta^{3/2}$$

(5)

where $R$ is asperity radius, $\delta$ is the pressing depth of asperity, $A$ is nominal contact area, $ds$ is the distribution density of asperities. The average thermal resistance of each contact point is obtained:

$$\frac{1}{R_c} = \sum_{i=1}^{n} \frac{1}{R_{ci}}$$

(6)

**Determination of Main Parameters**

Because of a smaller radius of convex peak contact so ignore caused by elastic deformation curvature change, calculation of heat energy must be explicitly vehicle parameters, as shown in table 1 for tracked vehicle parameters. The friction material is different from the dual material, and the friction material characteristics of the riveted friction plate are listed in table 2.

<table>
<thead>
<tr>
<th>Number of friction pieces of a single brake</th>
<th>Individual brake friction pairs</th>
<th>kerb weight (kg)</th>
<th>Initial speed (km/h)</th>
<th>contact area (mm²)</th>
</tr>
</thead>
<tbody>
<tr>
<td>2</td>
<td>2</td>
<td>4</td>
<td>60000</td>
<td>50</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Materials</th>
<th>$\lambda$ W/m°C</th>
<th>$\rho$ kg/m³</th>
<th>$c$ J/kg°C</th>
<th>$r$ J/mm²</th>
<th>$q$ °Cmm²/J</th>
<th>$R_c$</th>
</tr>
</thead>
<tbody>
<tr>
<td>Friction/Dual</td>
<td>6/10</td>
<td>6000/6000</td>
<td>552/700</td>
<td>0.3873</td>
<td>2495.4</td>
<td>0.01929</td>
</tr>
</tbody>
</table>
Studies were not given the coefficient of friction, but the friction plate and dual exert respective surface heat flux density of load, and ensure that the friction and dual pills to exchange heat under the transient temperature field calculation. The calculation of heat flux density in typical working conditions of brake is shown in table 3, and six typical braking conditions are selected, and the heat flux density of the friction pair is obtained according to the previous calculation.

<table>
<thead>
<tr>
<th>Working condition</th>
<th>The vehicle speed (km/h)</th>
<th>braking time (s)</th>
<th>Average heat flux density (W/m²)</th>
<th>Friction/dual heat flux density (W/m²)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>65</td>
<td>12</td>
<td>0.8*10^6</td>
<td>0.61<em>10^6/0.89</em>10^6</td>
</tr>
<tr>
<td>2</td>
<td>50</td>
<td>18</td>
<td>0.31*10^6</td>
<td>0.25<em>10^6/0.355</em>10^6</td>
</tr>
<tr>
<td>3</td>
<td>50</td>
<td>11</td>
<td>0.51*10^6</td>
<td>0.4<em>10^6/0.58</em>10^6</td>
</tr>
<tr>
<td>4</td>
<td>35</td>
<td>13</td>
<td>0.215*10^6</td>
<td>0.165<em>10^6/0.25</em>10^6</td>
</tr>
<tr>
<td>5</td>
<td>35</td>
<td>8</td>
<td>0.35*10^6</td>
<td>0.268<em>10^6/0.4</em>10^6</td>
</tr>
<tr>
<td>6</td>
<td>20</td>
<td>4</td>
<td>0.226*10^6</td>
<td>0.175<em>10^6/0.256</em>10^6</td>
</tr>
</tbody>
</table>

Results and Discussion

Finite Element Calculation Results

In the braking process, the brake friction plate generally is the process of the speed from high to low. In the braking process, the input of energy is not a constant. In the initial stage of the brake, due to the high speed, the friction components obtain more energy, and the brake needs to convert the higher mechanical energy into heat energy. But at the end of the braking, the speed is low, the brake input energy is limited, so the heat flow density also decreases. Therefore, the linear loading method is used to simulate the load input process of the vehicle in the braking process. Figure 4 shows the temperature field distribution cloud map of the friction pair in 0.1 second time.

Because of the decrease of heat flux density in linear loading, the initial stage of temperature rise is relatively fast and gradually slows down. The peak temperature is not much different, but the temperature of the highest point does not appear at the end of the braking, but in the braking process. This is because the temperature of the friction interface decreases with time as the heat flux decreases as the heat flux is less than the heat transferred by the metal heat transfer. Figure 5 shows the temperature variation of the friction plate and the contact point of the dual interface in 60 seconds after the braking. It can be seen that the temperature of the friction plate in each working condition is lower than that of the dual film.
Friction Pair Temperature Measurement Test

Different conditions of temperature difference is obvious, the temperature of the working condition of 1 up to about 400 °C. In order to verify the accuracy of Abaqus finite element simulation, the brake friction pair temperature measurement test was designed and carried out. The thermocouple temperature measurement method is used to obtain the temperature change process of the point. As shown in FIG. 7, because of the rotation of friction disc of brake, the method of measuring dual film is used to measure.

Then, the temperature curve of the node element in the finite element model is derived by Abaqus, and the two are compared. The performance parameters of thermocouple and thermal conductive adhesive are shown in table 4. The test process is carried out on the brake test bench, as shown in FIG. 6, the left diagram is the brake test bench, and the right picture is the brake thermocouple fixed on the dual film of the brake.

Table 4. Thermocouple test condition table

<table>
<thead>
<tr>
<th>Model of thermocouple</th>
<th>Specification</th>
<th>sample frequency(Hz)</th>
<th>measurement range(°C)</th>
<th>precision (°C)</th>
<th>test time(s)</th>
</tr>
</thead>
<tbody>
<tr>
<td>WRKK-103T</td>
<td>Ф1.5×50+5</td>
<td>100Hz</td>
<td>-40-1300</td>
<td>±1.5</td>
<td>25</td>
</tr>
</tbody>
</table>

The test process includes the start before the braking and the braking when the speed is stable. The moment of inertia of the test bed is:134kgm². The braking time is about 12 seconds. Due to the short time, the influence of air convection heat transfer on temperature is ignored. The main advantage of the new simulation method proposed in this paper is to consider the influence of the surface state of the friction pair on the temperature. Figure 7 shows the difference between this method and the general method.

Both of them are not very different at the beginning of the simulation, but for the calculation of the peak temperature, the finite element method considering the surface topography influences the higher temperature. Due to the effect of obstacles in the transfer of the surface morphology for the temperature, the effect will be greater restrictions on the friction pair surface heat, so as to increase the friction pair of substrate material and the surface temperature difference.
It can be seen from the comparison of experimental values that the calculation results of the finite element simulation method proposed in this paper are more consistent with the experimental values. The effect of surface topography on temperature rise can’t be ignored, which affects the temperature of the highest point of friction. It is of great significance for the design of the friction pair structure to be able to calculate the maximum temperature of friction pairs more accurately.

Conclusions

The meaningful conclusions obtained through the above analysis are as follows:

1. The surface topography of the friction pair will affect the heat flow distribution and generate thermal resistance, which will increase the temperature of the highest point of friction.

2. The accuracy of finite element calculation can be improved by establishing multi-point thermal resistance model and establishing finite element model with heat flow distribution.

3. The thermocouple method can accurately measure the temperature variation law of the braking process, and the test value is in good agreement with the predicted value of the finite element calculation.

References


