Thermal-mechanical Coupling Analysis of Rear Wheel Brake of Sightseeing Vehicle Based on Workbench

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Abstract: Taking the rear wheel brake of a certain brand of sightseeing vehicle as an example, floating caliper disc brakes and leading trailing shoe brake were simulated and compared for determining rear wheel brake types. The simplified model is established by UG software, imported into Workbench for thermal-mechanical coupling analysis under 10 braking conditions. The results show that the two brakes can meet the requirements of the vehicle, the former has better performance, and the latter has lower cost.

1. Introduction
It is meaningful for manufacturers to choose suitable brakes for vehicles. There is a huge difference in the structure and manufacturing cost of different brake combinations. Workbench can perform finite element analysis on different brakes [1], and according to the simulation results, the manufacturer can choose the suitable brake type.

2. Main technical parameters of the vehicle
The rear brake of a 14-seater sightseeing vehicle of a certain brand is simulated in this article, and the relevant technical parameters are shown in Table 1.

| Vehicle parameters                                      | Symbol | Value  |
|--------------------------------------------------------|--------|--------|
| Distance from center of mass of vehicle to front axle  | a      | 1692.9 |
| Distance from center of mass of vehicle to rear axle   | b      | 1442.1 |
| Core height (mm)                                       | h<sub>g</sub> | 650    |
| Vehicle full load weight (kg)                          | m      | 2525   |
| Rim diameter (mm)                                      | R      | 309    |

3. The main parameters of the rear wheel brake of the vehicle
The primary types of rear brakes for sightseeing vehicle are: floating caliper disc brakes and leading trailing shoe brakes, both brakes can meet the requirements of rear wheel brakes. The model of the former is shown in Fig.1, and the latter is shown in Fig.2. Bring the technical parameters in Table 1 into Eq.1 and Eq.2, it can be obtained that the front and rear braking forces are \( F_{\mu 1} = 12389.71 \text{N} \), \( F_{\mu 2} = 7406.29 \text{N} \) on the road with adhesion coefficient \( \phi = 0.8 \). According to Eq.3, the braking force distribution coefficient \( \beta = 0.626 \).
\[
F_{\mu 1} = \frac{G}{L} (b + zh \ g) \varphi 
\]
\[
F_{\mu 2} = \frac{G}{L} (a - zh \ g) \varphi 
\]
\[
\beta = \frac{F_{\mu 1}}{F_{\mu 2}}
\]

Where: \(F_{\mu 1}, F_{\mu 2}\) - Front and rear brake braking force[N]; \(\beta\) - Road adhesion coefficient; \(z\) - Severity of braking, when the front and rear wheels are locked, \(z=\beta\);

Fig. 1 Model of floating caliper disc brakes

Fig. 2 Model of leading trailing shoe brake

3.1. Determination of the main parameters of the disc brake
The calculation of various brake-related parameters has been introduced in detail in the work\(^2\) of Professor Liu from Tsinghua University. The calculation process in this work is used as a reference to calculate the main parameters of disc brakes: disc brake diameter \(D=280\text{mm}\); brake disc thickness \(h=25\text{mm}\); the inner and outer radius \(R_1\) and \(R_2\) of the brake lining are \(136\text{mm}\) and \(90\text{mm}\); braking efficiency factor \(BF=0.8\); wheel cylinder diameter \(d=46\text{mm}\);

3.2. Determination of the main parameters of the drum brake
The main parameters of the drum brake: the inner diameter \(D\) of the brake drum is \(260\text{mm}\); brake shoe wrap angle \(\beta=95^\circ\); brake shoe width \(B=75\text{mm}\); braking efficiency factor \(BF=2.2\); wheel cylinder diameter \(d=25\text{mm}\);

4. Finite element simulation analysis
Brake discs and brake drums are analyzed by thermal-mechanical coupling in Workbench.

4.1. Thermal-mechanical coupling analysis of brake disc
According to the parameters calculated above, the simplified model is established by UG software. Gray cast iron HT250 is selected as the material. The brake disc model is imported into the software for mesh division, as shown in Fig. 3.

In the braking simulation process, heat radiation and heat conduction are ignored for the convenience of calculation, because they have relatively little influence on the temperature rise results. According to the heat flux density generated during the braking process and the heat dissipation coefficient of the surface of the brake disc parts, the thermal load is added to the brake disc. The initial temperature of the brake disc and the environment is set to \(22^\circ\text{C}\).

Theoretically, the equation for calculating the total heat flux density due to frictional heat in the braking process\(^3\) is:
\[
q = F_{\mu} V
\]
Where: \(q\) - Total heat flux[W]; \(F_{\mu}\) - Total braking force[N]; \(v\) - The speed of the vehicle[m/s];
From Eq.4, the heat flux calculation equation on the rear wheel brake is:

\[ q_r = \frac{1 - \beta}{2} (1 - s)q \]  

(5)

Where: 
- \( q_r \) - Heat flux on the rear brake [W];
- \( s \) - Slip rate, take 10% in the calculation;
- \( \beta \) - Factor related to the wheel-to-road friction;
- \( q \) - General heat flux [W].

| Brake time (s) | Velocity (m/s) | General heat flux \( q \) (W) | Heat flux for rear brake \( q_r \) (W) |
|----------------|---------------|-------------------------------|-----------------------------------|
| 0              | 8.33          | 153897.61                     | 24042.63                          |
| 0.1            | 7.56          | 139679.52                     | 21821.41                          |
| 0.2            | 6.79          | 125461.43                     | 19600.20                          |
| 0.3            | 6.02          | 111243.34                     | 17378.97                          |
| 0.4            | 5.25          | 97025.25                      | 15157.75                          |
| 0.5            | 4.48          | 82807.16                      | 11825.92                          |
| 0.6            | 3.71          | 68589.07                      | 10715.32                          |
| 0.7            | 2.94          | 54370.98                      | 8494.09                           |
| 0.8            | 2.17          | 40152.90                      | 6272.88                           |
| 0.9            | 1.40          | 25934.91                      | 4051.66                           |
| 1.0            | 0.63          | 11716.72                      | 1830.44                           |
| 1.08           | 0            | 0                             | 0                                 |

About 95% of the heat is absorbed by the brake disc or drum [4]. Based on the low-speed characteristics of the sightseeing vehicle, the initial braking speed is set to 30 km/h. The heat flux density at each moment can be calculated based on the calculations of Eq.4 and Eq.5, and the specific values are shown in Table 2.

![Fig.3 Mesh model of brake disc](image1)

![Fig.4 Heat distribution cloud map of brake disc](image2)

![Fig.5 Maximum temperature curve of brake disc](image3)

![Fig.6 Stress distribution of brake disc](image4)
The heat dissipation coefficient of the end surface of the brake disc is set to 40 W/(m²·C), and the internal ventilation channel surface is set to 35 W/(m²·C)[5]. The thermal load is added, the ambient temperature and the initial temperature of the brake disc are set to 22°C. The vehicle takes a total of 1.08s of braking time from initial speed to stop, 10 times braking processes are simulated, and the interval between two braking processes is 1s. The heat distribution cloud map of the brake disc is calculated by Workbench, as shown in Fig.4, and the maximum temperature change curve of the brake disc is shown in Fig.5. Thermal-mechanical coupling analysis was carried out on the brake disc. 10 MPa pressures (the legal limit pressure under normal braking conditions) are added to the two working surfaces of the brake disc. Cylindrical support is added to the model to simulate the fixed connection between the brake disc and the rim, and the simulation result is shown in Fig.6.

It can be seen from Fig.4 and Fig.5 that the temperature of the friction surface gradually increases with the braking time, and the highest temperature appears on the edge of the working surface of the brake disc. The internal temperature gradually decreases, indicating that the heat gradually diffuses in the radial direction. After 10 times of continuous braking, the maximum temperature of the brake disc reached 81.631°C from the initial 22°C. It can be seen from Fig.6 that the maximum stress value appears at the contact position. Between the working surface of the brake disc and the fixed surface of the wheel hub, and the magnitude is 92.465 MPa.

4.2. Thermal-mechanical coupling analysis of brake drum

The brake drum of the drum brake is analyzed by thermal-mechanical coupling.

![Fig.7 Mesh model of brake drum](image1)
![Fig.8 Heat distribution cloud map of brake drum](image2)
![Fig.9 Maximum temperature curve of brake drum](image3)
![Fig.10 Stress distribution of brake drum](image4)

The heat dissipation factor is 10 W/(m²·C), the grid model is shown in Fig.7. 10 MPa pressures are added to the friction surface for thermodynamic analysis. The heat distribution cloud diagram of the brake drum is shown in Fig.8, the maximum temperature change curve is shown in Fig.9, and the stress
analysis result is shown in Fig. 10.

The friction surface of the brake drum has the highest temperature, and the internal temperature gradually decreases, indicating that the heat gradually diffuses in the radial direction, as can be seen from Fig. 8. As shown in Fig. 9, the brake drum has been braked continuously for 10 times, during which the highest temperature reached 85.7°C from the initial 22°C. The maximum stress value appears on the edge of the friction surface of the brake drum, about 167.4 MPa.

5. Conclusion
(1) After the thermal coupling analysis, the highest temperature of the brake disc is 81.6°C, and the brake drum is 85.76°C. The maximum temperature of the brake disc is 4.13°C lower than that of the brake drum. However, the temperature rise after each braking for both of them is less than 15°C, which is in a reasonable range.

(2) The maximum stress value of the brake disc is 92.465 MPa, and the brake drum is 167.4 MPa. The maximum stress value of the brake drum is 74.935 MPa larger than that of the brake disc, but both are less than the allowable stress 250 MPa of HT250 material.

(3) From the simulation analysis data of brake discs and brake drums, it can be seen that the performance of disc brakes is better than that of drum brakes. From the actual production cost, the manufacturing cost of disc brakes is 2 to 3 times that of drum brakes. According to the analysis of simulation results, both brakes can meet the requirements of safe use.

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