Optimized Design Based on FSC Racing Wheels and Brake Discs

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Abstract. As we all know, there are many factors that affect the lap time of a racing car, and the performance of the braking system is an important part to ensure the delayed braking and driver safety. In this paper, we plan to design a wheelside braking system for FSC racing cars through CATIA, ANSYS and other modeling and simulation software, and verify the reliability of the design through calibration and calculation for various working conditions, to ensure that the brake system can function properly under extreme conditions.

1. Introduction
The China Formula Student Car Competition ("China FSC") is a car design and manufacturing competition in which teams of students from higher education institutions majoring in automotive engineering or automotive-related disciplines participate. [1] Each team designs and builds a small single-seater car within one year according to the rules and standards of car manufacturing. The key to the design of the braking system is to ensure that the wheel edge part has enough strength to transmit the force from the ground to the tires, and to determine the reasonable front and rear braking force distribution coefficient according to the load ratio of the whole vehicle, so as to infer the specific parameters of the brake caliper and brake master cylinder, and to select the type according to the demand, so as to ensure that the vehicle will hold all four wheels at the same time under strong braking, which will neither make the front wheels hold in advance, resulting in This will not cause the front wheels to lose steering ability, nor will it cause the rear wheels to lock in advance and skid.[2]

2. Calculation of braking force distribution factor
When braking, it is necessary to ensure that all four wheels are held at the same time. On a road with a coefficient of adhesion, the conditions that need to be met for the front and rear wheels to be held at the same time are: the sum of the braking forces of the front and rear wheels is equal to the adhesion force, and the braking forces of the front and rear wheels are equal to their respective adhesion forces, that is

$$F_{\mu 2} = \frac{1}{2} \left[ \frac{G}{h_g} \sqrt{b^2 + \frac{4h_g^2L}{G}} - \frac{Gb}{h_g} + F_{\mu 1} \right]$$

In the formula; $F_{u1}$ —— Front wheel brake braking force (N); $F_{u2}$ —— Rear wheel brake braking force (N); $G$ —— Vehicle Gravity (N); $L$ —— Vehicle Wheelbase (m); $b$ —— Distance from the centerline of the rear axle of the vehicle to the center of mass (m); $h_g$ —— Vehicle center of mass height (m) [3].

[1] Publication details: ICAMMT 2021, Journal of Physics: Conference Series 1885 (2021) 052018, doi:10.1088/1742-6596/1885/5/052018
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The ideal braking force distribution curve for the front and rear brakes can be obtained from equation (1), referred to as the I curve. β represents the braking force distribution coefficient:

\[ \beta = \frac{F_{\mu 1}}{F_{\mu 1} + F_{\mu 2}} \]  

(2)

According to the actual situation of the road and tire parameters to take the adhesion coefficient of 1.5, the maximum deceleration of the vehicle is 1.5g, drawing different braking intensity z oblique intersection line, and then take z = 1.5 when its intersection with the i curve, and the origin can be connected to obtain \( \beta \) curve as follows, the data from the figure can determine the braking force distribution coefficient is about 0.7, in the brake system assembly, you can get the preset braking force distribution through the balance bar coefficient [4].

![Figure 1 Racing brake I curve and β curve](image)

3. Wheel Design

One side of the wheel is connected to the racing rim tire through wheel bolts, the middle is fixed with brake discs through rivets wave spacers and so on, and the other end is matched with the column through bearings, the force situation is complicated and the load is large, anything in the design and processing stage can lead to serious accidents and threaten the safety of racing cars and drivers, so the wheel must be carefully calibrated for the force in the design stage.
3.1. Wheel force calculation

According to the design experience, there are three main limit conditions for racing cars, acceleration (1.2g), steering (1.5g), and braking (-1.57g). However, there is no single operating condition in the actual driving of the car, so we superimpose on this basis, so there are: acceleration and steering (1.2g, 1.5g), and braking and steering (-1.57g, 1.5g). Here we take the front axle wheels, which are subjected to greater load transfer during braking, as an example, and analyze the braking and steering conditions where the acceleration and steering and load transfer are greater. Since the left and right wheels are symmetrical to each other, we only need to calculate and analyze one of the sides to know the safety performance of the design. Therefore, this design is all based on the right-hand wheel of the race car as an example.

Condition 1, -1.57g acceleration braking: Due to inertia, the load will be transferred from the rear axle to the front axle when the car is braked, which can be obtained from the formula:

\[
\Delta F_1 = M a_{y_2} \cdot \left( h / L \right) = 831.6 \text{N}
\]  

\[
F_{f1} = 0.5 \left( (b / L) \cdot Mg + \Delta F_1 \right) = 1088.3 \text{N}
\]

\[
F_{BF} = 0.7 \cdot 0.5 \cdot M a_{y_2} = 3850.9 \text{N}
\]

Where \(\Delta F_1\) is the total load transferred from the rear axle to the front axle during braking; \(F_{f1}\) is the vertical load applied to a single front wheel during braking; \(F_{BF}\) is the braking force generated by the ground on a single front wheel; \(M\) is the total mass of the car; \(a_{y_2}\) is the maximum braking acceleration of the race car; \(g\) is the acceleration of gravity, taken as 9.8; \(h\) is the height of the center of mass; \(L\) is the axis distance; and \(b\) is the distance from the rear axis to the center of mass.
Assuming that all of the external forces on the braking acceleration generated by the race car are generated by the brake disc, we have:

\[ F_{BMF} = F_{Br}R = 375.5N \cdot m \]  \hspace{1cm} (6)
\[ F_{Br} = 2F_{BMF} / d_f = 3850.9N \]  \hspace{1cm} (7)

Where \( F_{BMF} \) is the braking torque of the front single wheel; \( F_{Br} \) is the braking force of a single front caliper; \( R \) is the wheel radius; \( d_f \) is the center diameter of the brake disc.

Condition 2, +1.5g acceleration steering (left turn): It is assumed that the load is only transferred between the inside and outside wheels when the car is steering, and because the lateral force is mainly carried by the outside wheels when the car is turning at the limit. For conservative consideration, this design assumes that the lateral forces are all carried by the outer wheels and ignores the actual proportion, so that we can obtain:

\[ \Delta F_2 = (b / L) \cdot Ma_s \cdot \left( h / T_f \right) = 459.3N \]  \hspace{1cm} (8)
\[ F_{xf} = 0.5 \cdot (b / L) \cdot Mg + \Delta F_2 = 1131.8N \]  \hspace{1cm} (9)
\[ F_y = 0.5(b / L)Ma_s = 1008.8N \]  \hspace{1cm} (10)

Where \( \Delta F_2 \) is the amount of load transferred from the inner wheel of the front axle to the outer side during limit steering; \( F_{xf} \) is the vertical load on the outer wheel of the front suspension during limit steering; and \( F_y \) is the required lateral force on the front axle during limit steering.

Condition 3, combined condition (braking + steering): assuming that the car is braking and steering in the corner (ignoring the tire friction limit), that is, we have

\[ F_{xo} = 0.5 \cdot (b / L) \cdot Mg + 0.5F_1 + \Delta F_2 = 1547.6N \]  \hspace{1cm} (11)
\[ F_{Br} = 1642.5N \]  \hspace{1cm} (12)
\[ F_{BMF} = 375.5N \cdot m \]  \hspace{1cm} (13)
\[ F_{Br} = 3850.9N \]  \hspace{1cm} (14)
\[ F_y = 1008.8N \]  \hspace{1cm} (15)

Where \( F_{xo} \) is the vertical load on the outside front axle wheel of the car during the combined front axle condition (superposition of braking and steering conditions). \( F_{Br} \) is the braking force generated by the ground on a single front wheel; \( F_{BMF} \) is the braking torque of a single front wheel; \( F_{Br} \) is the braking force of a single front caliper; and \( F_y \) is the lateral force required by the front axle during extreme steering.

3.2. Wheel model building

The wheel hub is mounted on the column through the bearing connection, and the initial model of the front wheel hub is established by combining the custom parameters of the racing car and the diameter of the brake disc 195mm and the inner diameter 41.75mm, and the model is obtained after topology optimization as Figure 3.
3.3. Finite element static analysis of wheel hub

The structural simulation is optimized according to the extreme working conditions of the wheelside. This analysis mainly refers to 3 factors: equivalent deformation, equivalent force, and safety factor. For the consideration of light weight, the wheel material is selected as 7075-T6 aluminum alloy, and 1mm mesh is used according to the accuracy need. From the above calculation, the corresponding load is applied to the wheel model in ANSYS simulation software as shown in Figure 4.

![Wheel model in ANSYS simulation software](image)

The simulation results of equivalent deformation and equivalent stress show that the maximum equivalent elastic deformation is 0.0031715mm, the maximum stress is 221.5MPa, which is less than the maximum allowable stress of 370MPa, and the minimum safety factor is 2.5734, which is greater...
than the minimum safety factor of 1.8. All three parameters are within the safety range, so the hub design meets the requirements.

4. Brake disc design

4.1. Determine design parameters

Brake parameters mainly include the diameter of the brake disc, the effective radius and the caliper piston area, [5] and after determining the structural parameters of the brake, the appropriate brake disc size can be calculated. One of the main materials of brake discs is 2Cr13 steel, which has good machinability, excellent corrosion resistance after heat treatment and good toughness, and can be used for parts subjected to high stress loads.

| Table 1  | Rim and brake disc related parameters |
|----------|--------------------------------------|
| Rim diameter (mm) | 254 |
| Brake disc diameter interval D (mm) | 195≤D≤263.31 |
| Selected front wheel disc diameter D (mm) | 195 |
| Selected rear wheel brake disc diameter D (mm) | 195 |
| Front wheel brake disc inner diameter (mm) | 41.75 |
| Rear wheel brake disc inner diameter (mm) | 51.25 |
| Brake disc temperature setting (℃) | 300 |
| Materials | 2Cr13 |

| Table 2  | Brake Lining Block Parameters |
|----------|-------------------------------|
| Friction liner radial dimension (mm) | 30 |
| Friction coefficient f (generally taken from 0.3 to 0.35 for design calculation) | 0.35 |
| Working area of front wheel friction lining block A (cm²) | 17 |
| Working area of rear wheel friction lining block A (cm²) | 17 |
| Front wheel friction lining block m value (≥0.65) | 0.69 |

4.2. Brake disc modeling

In previous years, the brake disc model is shown in Figure 9, while according to the design disc diameter and hub design parameters, the improved spiral through-hole model is shown in Figure 10:

In CATIA material addition module for previous years brake disc and this year's brake disc added material for steel, density of 7860kg/m3, its density is similar to 2Cr13 steel, the use of measurement inertia module to derive four brake disc mass as shown in the figure below.

Front disc: 0.65kg in previous years, optimized to 0.592kg. 8.9% lighter
Rear disc: 0.568kg in previous years, 0.512kg after optimization. 9.9% lighter
4.3. Brake Caliper Selection

The selection of brake calipers should take into account the price, space within the rim, friction area, piston bore, piston number and many other factors, the following figure shows the three mainstream brake calipers on the market.

| Type                      | Willwood GP200 | ISR22-043 | Willwood PS1 |
|---------------------------|----------------|-----------|--------------|
| Unit Price                | 1240           | 3072      | 1200         |
| Single cylinder bore      | 1.25in         | 0.98in    | 1in          |
| Number of piston          | 2              | 4         | 1            |
| Total wheel cylinder cross-sectional area | 1587.1mm²   | 1963mm²  | 510mm²      |
| Total weight              | 0.9LBs         | 1.6LBs    | 1.1LBs       |
| Total friction area       | 1.83in²        | 2.56in²  | 0.79in²      |

The Wilwood GP200's single-chamber bore and total wheel cylinder cross-sectional area are moderate, and its two-piston arrangement eliminates some of the axial error problems in disc and caliper assembly, while its lighter weight helps reduce unsprung mass.

Although the ISR22-043 model has excellent performance in friction area, it is too heavy, increasing the undersprung mass of the car, and the excessive wheel cylinder cross-sectional area will increase pedal stroke, which is not conducive to faster braking response. And ISR price is too high, not in line with the concept of cost saving.

Wilwood PS1 single piston caliper, due to the experience of previous years, the parallelism between the disc and the caliper is not very high, which can lead to empty wear of the disc, while the multi-piston caliper can fit the disc better, while the single piston cannot solve such problems.

Therefore, for a FSC racing car, GP200 is a more suitable choice.

4.4. Selection of brake master cylinder

According to the formula:
In the formula, $d$ for the friction pad and brake disc distance to take 1mm, $a$ for the pipeline expansion rate, due to the difference between the front and rear pipeline length, so the front wheel to take 1.2, the rear wheel to take 1.4. $D$ for the caliper piston diameter, $n$ is the total number of pistons.

This data is substituted to calculate the front wheel master cylinder compression fluid volume of 5214mm³; rear wheel master cylinder compression fluid volume of 6829mm³.

So the Tilton 78 master cylinder of two different bore master cylinder, the front wheel use the smallest bore Tilton78-625 (full stroke to push the brake fluid volume of 5533 ) The rear wheels use the slightly larger Tilton 78-700 (full stroke to push the brake fluid volume of 6936.2).

4.5. Finite element analysis of brake disc

We choose the caliper for gp200 caliper, the front axle braking torque in the braking process as shown in Table 4, the braking torque to consider 30% heat recession.

| Table 4  Front axle braking torque |
|-----------------------------------|
| Braking torque Tf1’ generated by the front axle single-side wheel brake (N-m) | 375.5  |
| Front axle single side of the wheel friction lining block on the brake disc generated by the friction force (N) | 2776.05 |

And in the process of racing car braking, the brake disc and friction pad friction generates huge heat, which leads to brake disc heat recession, [6] so the main influence of brake disc heat dissipation are thermal stress as well as temperature, according to the material parameters we selected the brake disc material is 2Cr13, according to the accuracy requirements we use the accuracy of 3mm mesh brake disc through the caliper for the braking force generated by the brake disc for the brake disc Finite element analysis, in Workbench Transient Structural module, set the friction coefficient of 0.3, speed 60rad / s, friction pad surface pressure 3000N. and in the above simulation premise for heat dissipation simulation. Then, in the steady-state thermal module, the temperature in the boundary conditions is set to 120℃ (derived from the above transient structural analysis), and the air natural convection is 20W (m²*K), which simulates the heat dissipation of the vehicle under natural convection, and according to the analysis pictures, it can be seen that the heat is conducted faster in the brake disc, and the temperature is more dispersed, and in the actual driving process According to the analysis pictures, it can be seen that the heat conduction in the brake disc is faster, the temperature is more dispersed, and in the actual driving process, the temperature demand can be fully reached through the holes and natural fluid heat dissipation, so the set of brake disc model is reasonable.

5. Conclusion

In this paper, through the simulation and analysis of ANSYS static and thermodynamic modules, it is concluded that the equivalent force and equivalent deformation of the wheel model are within a reasonable range, and it is also verified that the optimized brake disc model can meet the requirements in terms of heat dissipation while achieving the lightweight target, and it has certain reference value for the selection of brake master cylinder components. For the design process of FSC racing wheels and brake discs, a reasonable reference scheme is given in this paper.

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