Design and Lightweight based on FSC racing monocoque

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Abstract. In order to improve the handling of the suspension system in FSC racing cars under extreme operating conditions, the overall strength stiffness of the car is improved to provide reliable support and protection for the driver, engine and all other components. This paper focuses on the torsional stiffness of the load bearing members required for the transient conditions of the car entering and exiting the corners. Based on the design value, 3D modeling was carried out according to the 95th percentile manikin and driver data. Importing Hypermesh to divide the mesh for finite element analysis of each working condition, and changing the skin thickness, core material thickness or core material type for each area of the whole vehicle to continue to optimize the performance of the parts and complete the purpose of lightweighting.

1. Introduction

Reducing structural weight is of special importance for racing cars. Advanced composite materials have many excellent characteristics such as high specific strength and specific modulus, designable performance and easy monolithic forming. Using them in body structure can reduce weight by 25%-30% compared with conventional metal structure, and can significantly improve the strength and stiffness characteristics of racing cars, increase speed and structural performance, which is impossible or difficult to achieve by other materials.

Compared with the traditional steel tube truss structure, the use of composite monocoque technology can improve the structural form of the racing car frame and body, meet the requirements of the general arrangement of the car, and have more freedom of design to give the overall good handling performance of the car. The frame should also have sufficient strength and proper stiffness. Because it is in the complex driving process, there should be no interference between the assemblies and components fixed on the frame; and as a load-bearing component, it needs to bear the load of the racing car and the impact of the wheels when driving. In order to improve the light weight level of the whole car, the mass of the monocoque is required to be as small as possible; in addition, it should be arranged closer to the ground so that the center of gravity of the car can be lowered in order to help improve the driving stability of the car. In this paper, we use ergonomic concept to design and finite element to analyze the performance of the components, and finally verify the design objectives through experimental calibration.

2. Monocoque model design

The 95th percentile manikin was used as a reference for the determination of the basic man-machine space, and its H point was used as the reference point for the whole vehicle arrangement. We build a 1:1 ergonomics experiment stand as shown in Figure 1 to carry out comprehensive human body data collection, which has important reference data for the general arrangement. The general arrangement
data was obtained by weighting processing based on each rider's responsible project, and the simulation of sitting posture and field of view was carried out based on this to ensure the comfort of riders. Autodesk Alias software was chosen to carry out the 3D modeling of the monocoque. After finishing the modeling, the model was imported into catia to get the following as shown in Figure 2: maximum length: 1489.68mm maximum width: 762.68mm maximum height: 556.65mm.

Figure 1. Ergonomics experiment stand

Figure 2. 3D model of single shell

Subsequently, the 95th percentile human body model was established through the catia ergonomic design and analysis module. The posture adjustment command and the assembly module were used to assemble the human body model and the monocoque model to initially adjust the driver's driving posture and check the rationality of the monocoque design. The driver's field of view was checked by the vision command to ensure a wide driving field of view as shown in Figure 3. Finally, the ergonomic design of the monocoque is checked to meet the design objectives of driver comfort and reasonable cockpit space allocation.
3. Material selection and structural design

The sandwich structure design was chosen, which consists of two thin panels on top and bottom, core material and panel-core adhesive as shown in Figure 4, where the core material is very light in weight, and by increasing the thickness of the core material, the stiffness of the laminate can be greatly improved with only a small increase in mass. Due to the light weight and relatively large bending stiffness and strength, the sandwich structure is widely used in the aerospace field, and the usage rate in the automotive field is also gradually increasing, so it is important to analyze the sandwich structure, optimize and analyze it for the strength stiffness and light weight in this monocoque design.
The thin panel material is selected from carbon fiber prepreg, which is divided into 2554 plain fabric and 2554 unidirectional fabric, the core material is selected from 5052 type positive hexagonal aluminum honeycomb core and PVC high temperature resistant foam, and the panel-core adhesive is selected from high temperature resistant epoxy adhesive film.

The structural design and optimization selects the method of fitting the finite element simulation and physical experiment, and uses the finite element analysis to study the failure form and stiffness characteristics of the laminate in the three-point bending experiment, and verifies the analysis by experiment, so as to determine the final lay-up scheme.

Due to the anisotropy of carbon fiber composites, the bending modulus of the laminate depends largely on the lay-up sequence and the number of plies, and the relationship between the bending moment and curvature and the stiffness matrix within the laminate face when the laminate is subjected to bending stress can be obtained from the classical laminate theory. The coupling stiffness matrix \( B = 0 \) for the symmetrically laid laminate without coupling due to symmetric stress distribution, so the coupling problem can be ignored in this study. In general, the modulus of elasticity of honeycomb core layer is much lower than that of panel, so the modulus of elasticity is often neglected when calculating the laminate stiffness. Accordingly, combined with the laminate shift theorem, the laminate bending stiffness calculation formula can be obtained, and the formula is shown as follows:

\[
D = \frac{E_s b t d^2}{2}
\]

Where \( D \) is the laminate bending stiffness, \( E_s \) is the panel equivalent modulus of elasticity, \( b \) is the laminate width, \( t \) is the upper panel or lower panel thickness, \( d \) is the upper and lower skin midface spacing, \( h \) is the laminate thickness.

The laminate structure should be designed to meet the principle of symmetrical lay-up to avoid coupling deformation. At the same time to ensure that the mechanical properties of the laminate meet the requirements, the number of lay-up direction needs to be reduced as much as possible, choosing to use plain cloth \( 0^\circ \) and \( 45^\circ \) lay-up. The principle of minimum ply ratio is satisfied, even if each direction of the substrate is not subject to load, while ensuring that the ply ratio in any direction should be greater than 10%. When manufacturing, the thickness ratio of panel and honeycomb is as small as possible, and the ratio of its single panel layer thickness to sandwich layer thickness should not exceed 0.2. Combined with the above requirements the weight threshold of laminate in the case is set to 0.5 Kg, so the constraints can be organized as:

\[
\frac{(x_1 + x_2) t_{sf}}{h_c} \leq 0.2
\]

\[
\frac{x_1}{x_1 + x_2} > 0.1
\]

\[
\frac{x_2}{x_1 + x_2} > 0.1
\]

Where \( x_1 \) and \( x_2 \) are the number of layers of \( 0^\circ \) plain fabric and \( 45^\circ \) plain fabric, respectively, and the variable interval is from 4 to 16; the thickness of honeycomb core layer adopts the common standard thickness of 15mm, 20mm or 25mm. is the thickness of prepreg single layer molding.

In order to obtain the optimal number of layers of anisotropic prepreg for different core thicknesses, an optimization model based on the interior point method was developed and solved in Matlab mathematical software. The results of several computer iterations show that \( 0^\circ \) layup has better flexural performance compared with \( 45^\circ \) layup, and increasing the thickness of aluminum honeycomb layer can effectively reduce the number of layups while ensuring the bending stiffness of the laminate and reducing the mass of the laminate.

According to the above laws and optimization results, various lay-up forms are set for different areas of the monocoque shell. In order to achieve different strength and stiffness and energy absorption needs of each area and light weight.

The main function of the front bulkhead area is to simulate the front-end crash intrusion and ensure
the driver's driving safety. Therefore, in order to ensure the shear performance and reduce the weight, the traditional empirical pavement method is used for the pavement design of this area. Under shear conditions, the shear force is mainly borne by the upper plate layer, and the main form of stress is surface tension. Accordingly, the fiber direction will be as much as possible to want the same direction of surface tension, so that the 0° direction and 90° direction of the single item of cloth is used to resist internal tensile stress, while in the surface layer is still selected 45° plain cloth for shear performance requirements. The design of the front bulkhead support and drive system protection structure area should focus on the design of its flexural stiffness, and the use of asymmetric lay-up to reduce weight. In order to improve the collision energy absorption in the side impact area, PMI foam with better cushioning performance under collision conditions is used as the core material. The bottom edge of the side impact area and the front cabin floor area have no special requirements. To reduce the calculation workload and weight, the front bulkhead support area ply design is used. The front ring diagonal brace area resists the effect of torsional force, and PMI foam, which is easy to bend and shape, is chosen as the core material due to its shape. The main ring diagonal support area mainly resists the torsional force. After the completion of the pavement design, a simple laminate was made to do a three-point bending experiment to obtain the measured flexural stiffness to verify the pavement results. In summary, the design of each area is as follows:

| Area Name          | Pavement solution                                                                 | Measured flexural stiffness (N/m²) |
|--------------------|------------------------------------------------------------------------------------|-----------------------------------|
| Front bulkhead     | 45F/04/90/04/90/04/90/04/45/20mmhoneycomb/45F/0/0/0/0/0/0/0/45F                  | 2.11E+04                          |
| Front bulkhead support | 45F/0/0/0F/0/0/45F/25mmhoneycomb/45F/0/0 F/45F                        | 9.18E+03                          |
| Front ring diagonal brace | 45F/0/0F/0/45F/20mmPMI/45F/0/0/45F                       | 1.90E+03                          |
| Side impact side   | 45F/0/0/0F/0/0/0/0F/0/0/0/0/0/0/45F/25mmPMI/45F/0/0F/0/0/0/45F   | 4.75E+04                          |
| Front bulkhead     | 45F/0/0/0F/0/45F/20mmhoneycomb/45F/0/0/0/0/45F                  | 3.11E+03                          |
| Side impact bottom plate | 45F/0/0F/0/45F/20mmhoneycomb/45F/0/0/45F                        | 3.11E+03                          |
| Main ring diagonal brace | 45F/0F/0F/45F/15mmhoneycomb/45F/0F/0F/45F                    | 2.74E+03                          |

4. Stiffness Simulation
The model is imported through Hypermesh's Input Solver Deck command, and the interface properties are set using Beam Section for steel tube components such as anti-roll bars and suspension bars. The outer diameter of the anti-roll bar is 25.4mm and the wall thickness is 2.4mm, while the outer diameter of the suspension bar is 14mm and the wall thickness is 1mm.

Two types of cells are used for meshing, one is used to classify the 2D cells of the body, and the cells are classified in a mixed way, with the cell side length of 8mm, the number of cells is 74654, and the number of nodes is 74774. Another class is used to divide the anti-roll bar and suspension bar 1D unit, the unit type is selected CBAR, the unit is a one-dimensional bending beam unit, can withstand axial tension, axial torsion, in-plane bending and shear. The unit size is 10mm, the number of suspension bar units is 516, the number of nodes is 524, the number of anti-roll bar units is 850, the number of nodes is 860.

All the parts in the model are connected to each other. Among them, the suspension bars, the anti-roll bar and the body are connected by lugs, and the connection can be considered as a rigid connection. In the finite element model, the connection is made using RBE2, and the six degrees of
freedom of all the connected nodes of this unit are completely synchronized, i.e. there is no relative deformation between the nodes. The connection is made between the upper and lower outer hard points of the suspension bar and the wheel column, which can also be considered as a rigid connection.

The loads and boundary conditions of the model can be determined according to the torsional conditions as follows: (1) Set the restraint load at the rigid connections of the upper and lower bars of the rear suspension to restrain all six degrees of freedom; (2) Apply a force of 2000N at the rigid connections of the upper and lower bars on both sides of the front suspension respectively in the opposite direction of the Z-axis to apply torsional moments on the body.

The torsional moment acting on the body can be calculated by the following equation:

\[ M_n = \frac{(R_R - R_L)}{2} \cdot L \]  

Where, \( R_R - R_L \) is the sum of the hard point reaction force difference between the left and right suspensions. \( L \) is the average value of the distance between the hard points of the left and right suspension.

Body turning angle can be calculated according to the following formula:

\[ \theta = \frac{\delta}{\pi L} \times 180^\circ \]  

Where, \( \theta \) is the front axle angle of rotation, \( \delta \) is the front suspension hard point of the difference between the left and right Z-axis direction displacement.

The torsional stiffness \( K \) of the body can be calculated by the following equation:

\[ K = \frac{M_n}{\theta} \]  

In Hyperworks, a simulation based on the pavement design was established as shown in Figure 5, and the comprehensive torsional stiffness of 4708 N-m/deg was calculated, which meets the design requirements.

5. Conclusion
The design of the laminate is mainly based on the torsional stiffness problem using the traditional design method and the collaborative optimization method, and the combination of these two design methods makes the laminate design more reasonable and efficient, and the final mass of the
monocoque is 23 kg. The simulated torsional stiffness of the monocoque is 4708 N-m/deg, and the maximum deformation of the simulated lateral collision (185.5N applied to the lateral collision area) is 14.57mm, generating 129.5J of energy, which is less than the maximum energy absorption of 196mJ measured in the sample parts in this area, and the maximum stress of the lateral collision in this test condition is 136Mpa, so the safety of the driver in the cockpit can be guaranteed.

Acknowledgments
This work was supported by National innovation and entrepreneurship training program for college students 202010497059.

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