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Motion Simulation and Optimization Design of FSAE Racing Car Rear Suspension

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Abstract: To improve the handling performance and stability of the racing car in the race, we analyze and optimize the rear suspension of the racing car. Firstly, we get the parameter values of the rear suspension according to the relevant experience at home and abroad. According to the results, the rear suspension model was constructed by CATIA. Then, Adams/Car was used to establish the modeling and simulation analysis, and the unsatisfactory parameters were optimized by Adams/Insight. Finally, ANSYS is used to analyze the main acceptance components of the rear suspension to optimize the suspension system. The result is a rear suspension system that complies with FSAE race rules.

1. Introduction

Formula University of China (FSAE) is a national event organized by students from more than one hundred universities in vehicle engineering and related majors. In accordance with the rules of the race and car manufacturing standards, each team will design and manufacture a small racing car with excellent performance in acceleration, braking, handling, and other aspects within one year, which can successfully complete all or part of the race links. The suspension system is the key to ensuring the racing car has a good ride and stability. The optimization design of the suspension system can make the car have good handling stability and ride comfort when it is running at high speed on an uneven road and curved road. This paper will design and optimize the suspension system based on the existing parameters of the racing frame according to the rules of the race.

2. FSAE Racing Car Rear Suspension Design

2.1. Basic Structure of FSAE Racing Suspension

As a general term for all connected devices between the frame and wheels, racing suspension usually includes elastic components, shock absorbers, guiding mechanisms, and anti-tilt bars.

2.2. FSAE Racing Suspension Type Determination

Automobile suspension can be divided into independent suspension and non-independent suspension two types, considering the racing car to high speed running stability requirements, at the same time reference at home and abroad racing suspension, this paper uses independent suspension. The independent suspension is mainly divided into single transverse (longitudinal) arm type, double transverse (longitudinal) arm type, and McPherson type. Due to the advantages of convenient manufacturing and installation of double transverse arm independent suspension, the optimized racing suspension designed in this paper adopts double transverse arm independent suspension.

Before the suspension parameters are selected, the technical parameters of the whole vehicle are given as follows:

Table 1: Vehicle Parameters of The Racing Car.

Name	Parameter
The wheelbase (mm)	1540
Before the wheel track (mm)	1128
After the wheel track (mm)	1154
Vehicle readiness quality (kg)	305
The total quality (kg)	380
Front and rear axle load distribution	45/55

2.3. Determination of Main Parameters of Rear Suspension of FSAE Racing Car

2.3.1. Frequency Deviation of Rear Suspension of FSAE Racing Car

In the suspension design parameters, the offset frequency is an important parameter considered in the suspension design. FSAE rear suspension offset frequency refers to the natural frequency of the rear half of the car body. It can be calculated by the following formula:

$$f = \frac{1}{2\pi}\sqrt{\frac{k}{m}} \tag{1}$$

k is the stiffness of the rear suspension of the automobile, and its unit is N/cm. m is the sprung mass of the rear suspension of an automobile, and its unit is kg.

As for the FSAE racing car, which is the object of this study, its main purpose is to complete the race and its requirements for ride comfort are relatively low, so the offset frequency selected in this paper is larger than that of ordinary cars. The initial selection of the rear suspension offset frequency f is $2.3Hz$.

2.3.2. Side Angle Stiffness of Rear Suspension of FSAE Racing Car

When the roll angle is small, the couple moment required at the Angle of the car is called the roll stiffness of the suspension. The roll angle stiffness of suspension affects the roll Angle and the stability of vehicle[1].

For the independent suspension system, the calculation formula of the roll stiffness of the suspension can be expressed as:

$$K_\phi = \frac{1}{2} K_W B^2 \quad (2)$$

K_W is the suspension line stiffness, and its unit is N/mm. B is the wheelbase of the car, and its unit is mm.

Too much or too little roll stiffness of suspension is not good. If the roll stiffness is too small, passengers will lack the sense of security and comfort. If the roll stiffness is too small, it is not easy to feel the car roll. For different models, the requirements of roll stiffness are different, but under the action of $0.5g$ lateral acceleration, the roll Angle of the vehicle should be controlled within the range of $2°$ to $5°$. The distribution of roll stiffness in front and rear wheels will also affect the handling stability of the car, so this aspect should also be considered in the design. It is generally required that the roll angle stiffness of automobile front suspension is larger than that of rear suspension.

2.4. Design of Steering Mechanism of Rear Suspension of FSAE Racing Car

2.4.1. Center of Inclination

The roll center is the instantaneous center of roll movement, which reflects the important characteristics of the steering mechanism, and the height of the roll center has a decisive influence on the roll Angle and lateral moment of the body. Due to the low chassis of the FSAE car, the roll center of the rear suspension was initially set at $70mm$.

2.4.2. Trim The Center

The trim center of the double-wishbone independent suspension can be obtained by drawing. The intersection of the extension lines of the rotation axes of the upper and lower wishbone can be obtained[2], as shown in Figure 1.

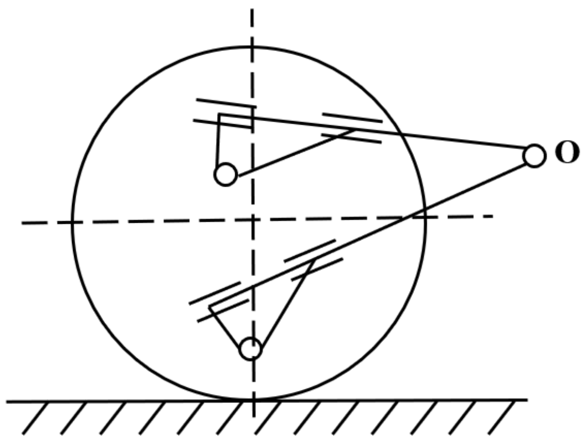


Figure 1: Determination of Trim Center of Double Wishbone Independent Suspension.

2.4.3. Upper and Lower Arm Design

In order to ensure the handling stability of the car, the length of the upper and lower arm is generally the same. In the design of the upper and lower arm, but also to consider the impact of tire changes. In order to reduce tire wear, we hope that when the wheel beats, the change of the wheel edge is small, so the length ratio of the upper and lower transverse arms is about 0.6. The ratio of the upper and lower transverse arms designed in this paper is 0.8, and the upper and lower transverse arm models established are shown in the figure below.

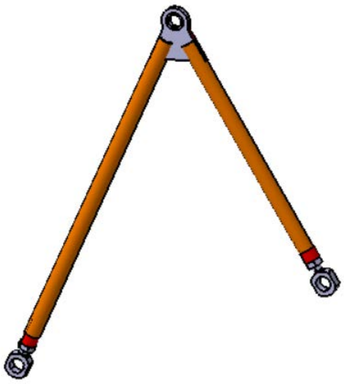


Figure 2: Right Suspension Transverse Arm.

Figure 3: Lower Transverse Arm of Right Suspension.

2.4.4. Rocker Arm and Push Rod Design

The rocker arm and push rod are mainly used to connect the shock absorber to the lower arm. The established rocker arm is shown in Figure 4, and the push rod is shown in Figure 5.

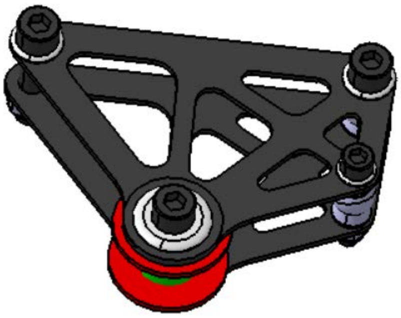


Figure 4: Rear Suspension Rocker Arm.

Figure 5: Rear Suspension Push Rod.

2.5. Elastic Components and Shock Absorbers.

The elastic components of the suspension have a variety of structural forms, the common structural forms are leaf spring, torsion bar spring, spiral spring, rubber spring, and air spring. According to the structure of double wishbone suspension and the structural characteristics of FSAE racing suspension, this paper selects spiral spring as the elastic element of the front and rear suspension.

As an important part of the suspension, the shock absorber can accelerate the attenuation of vibration between the body and the frame, so as to improve the ride of the car. The most commonly used shock absorbers in suspension are cylinder hydraulic shock absorbers, which can be divided into three types, namely single-cylinder shock absorbers, double cylinder shock absorbers, and inflatable shock absorbers. FSAE racing car is different from general cars. In order to make the racing car have good stability performance, it needs shock absorbers with better shock absorbers. In general, special damping adjustable shock absorbers of racing models are selected, which can be directly purchased in the market. The arrangement of the FSAE shock absorbers is approximately horizontal, with a symmetrical form on the left and right. The distance between front suspension shock absorbers is relatively close, while the distance between rear suspension shock absorbers may increase due to the arrangement of the engine.

2.6. Wheel Positioning Parameters.

The relationship between wheel positioning parameters changing with wheel beat and wheelbase changing with wheel beat is the main research content of suspension kinematics [3], so the design of wheel positioning parameters is also an important content of the suspension design of FSAE racing.

- **Caster angle**

The setting of the caster angle is to keep the car stable when driving on the road, its role is in the car driving when the steering wheel is caused by external force sometimes deflection, with the roadside force of the tire forming a stable torque of rotation to achieve. It is generally required that the kingpin inclination increase with the up jump of the wheel, which also offsets the tendency for the caster angle to decrease with the brake nod.

- **Kingpin inclination angle**

The kingpin inner inclination Angle and the kingpin back inclination Angle also have the steering wheel automatic righting function, cannot be too large.

- **Camber angle**

Camber angle also ensures that the car can drive in a straight line. For the car, it is hoped that the tire can provide enough tire roll force when turning, and the car with a certain camber angle can make the car make full use of the performance of the tire when turning, to get enough roll force.

- **Toe angle**

The toe angle is set to coordinate with the camber angle of the wheel to avoid the phenomenon of the wheel rolling and sliding. Keep the wheel pure rolling and straight driving, thus reducing tire wear and power consumption, improving the service life of tires and fuel economy of cars.

3. Simulation Analysis and Optimization of Rear Suspension of FSAE Racing Car

3.1. Modeling of Rear Suspension.

In order to test the rationality of the rear suspension parameters and the model, and further optimize it, according to the selected parameters, the Adams/Car module in Adams software is used to establish the rear suspension model. The hardpoints of the parts to be built are shown in Figure 6.

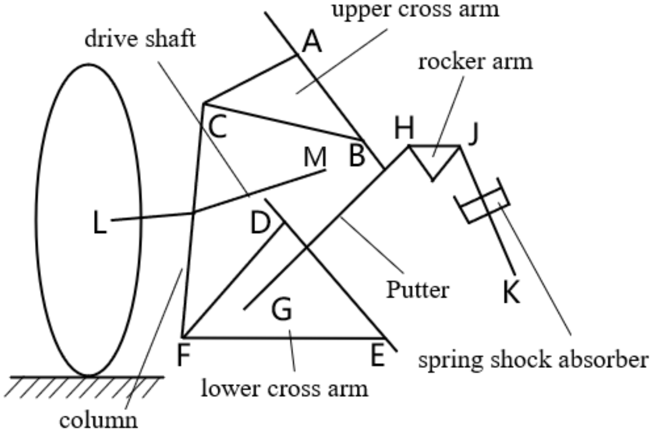


Figure 6: Structure Diagram of Rear Suspension.

The rear suspension model was established according to the above hardpoints, and the suspension assembly of the rear suspension was established to prepare for the following simulation, as shown in Figure 7.

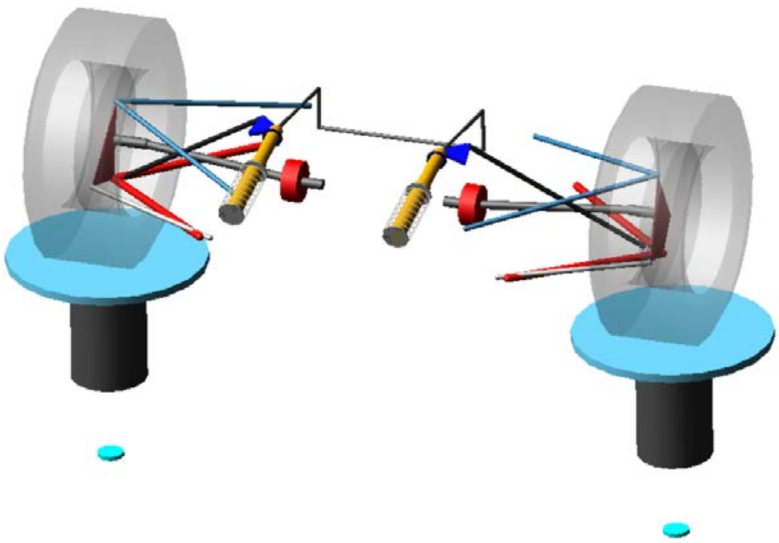


Figure 7: Rear suspension total configuration.

3.2. Simulation Analysis of Rear Suspension

When FSAE racers encounter uneven roads or obstacles, the wheels will jump up and down, so the suspension motion can be analyzed by the change of suspension parameters during tire jump simulation[4].In this paper, the characteristics of the rear suspension are analyzed by using the double wheel parallel runout simulation, and the runout is set to $30mm$ to get the simulation curve of the main characteristic parameters of the rear suspension.

- **Toe angle**

An appropriate toe angle can ensure the stability of the car when driving, and its simulation curve is shown in the figure below.

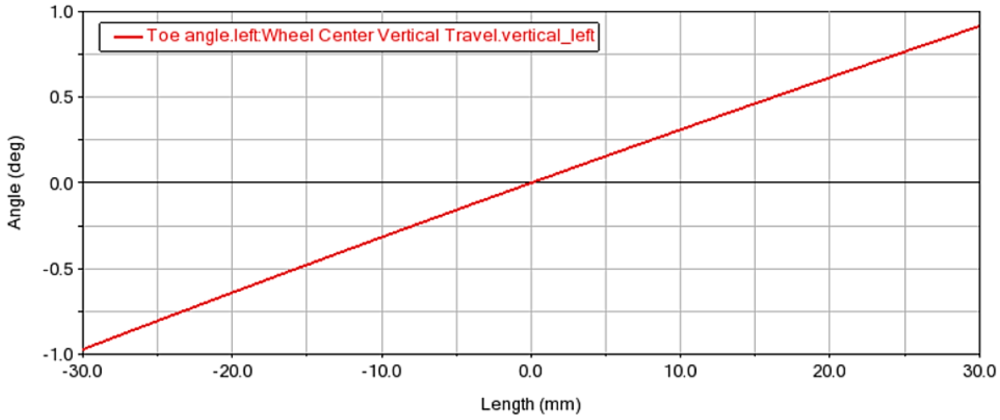


Figure 8: Change Curve of Front Beam Angle of Rear Wheel.

It is generally desirable to maintain a small range of rear front beam angles as the wheels bounce up and down. As can be seen from Figure 8, the toe angle presents an upward trend during wheel jump, with an overall variation range of $-0.95°\sim0.9°$, which is relatively large and needs to be optimized.

- **Rear-wheel camber**

As one of the main parameters of wheel positioning, the rear wheel camber affects the lateral stability of the racing car. Its simulation curve is shown in the figure below.

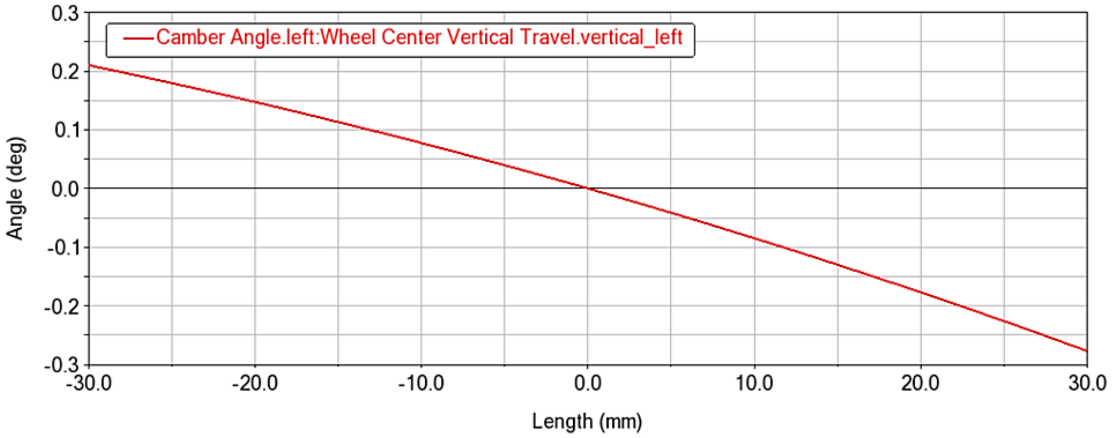


Figure 9: Variation Curve of Rear Wheel Camber.

It is generally expected that the rear wheel camber will decrease with the drop of the wheel and increase with the drop of the wheel. It can be seen from Figure 9 that the variation of rear-wheel camber of the rear suspension model constructed in this paper conforms to this change rule. Inside and outside Angle change is $0.49°$ during the whole stroke of up and down, which is ideal.

- **Rear wheelbase**

The simulation curve of the rear wheelbase is as follows. Too much change of rear wheelbase will lead to wear of rear wheel and reduce the service life of the tire. It can be seen from Figure 10 that the total change of rear suspension left rear wheelbase with the wheel jumping up and down is within a reasonable range of $1mm$.

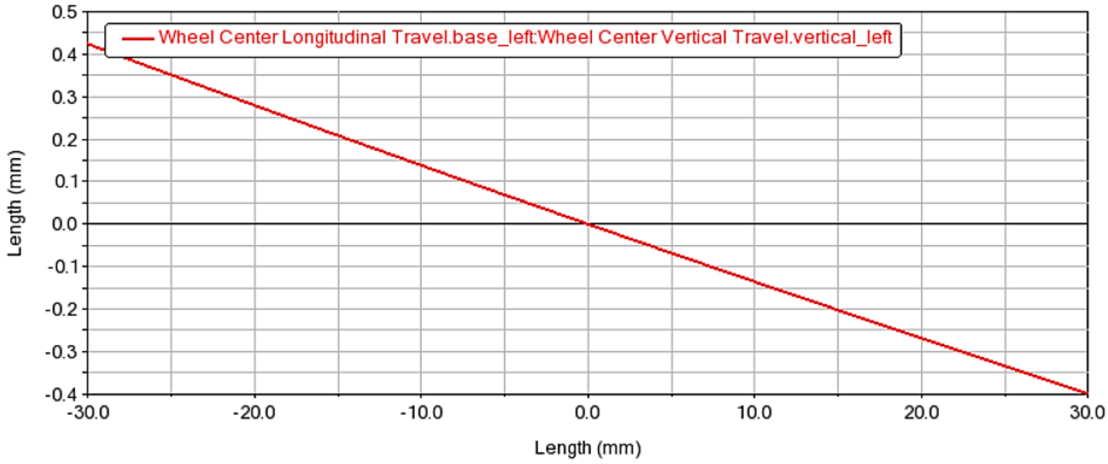


Figure 10: Change Curve of Rear Wheel Base.

- **Rear inclination center height of suspension roll**

Figure 11 shows that the roll center height of the rear suspension at rest is $70mm$. To make the change of the wheelbase not too large when the car body tilts and reduce tire wear, the roll center may be required to be small, so the height of the roll center needs to be further optimized.

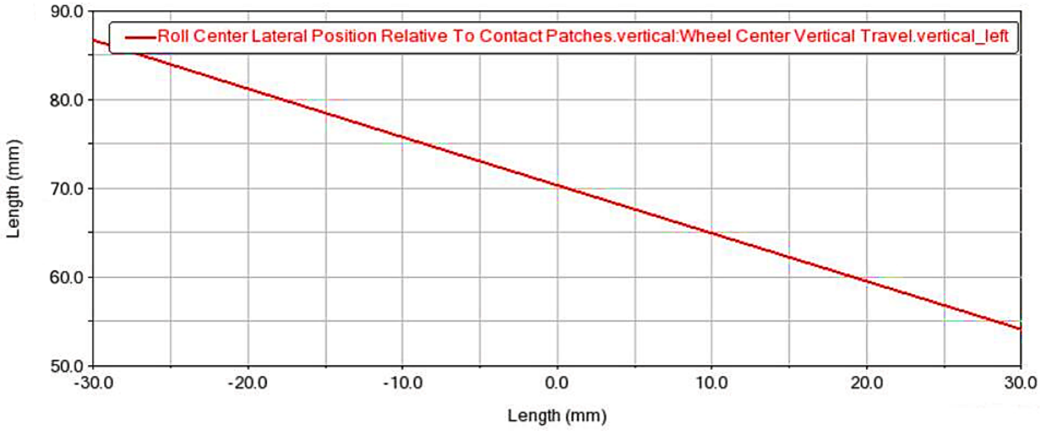
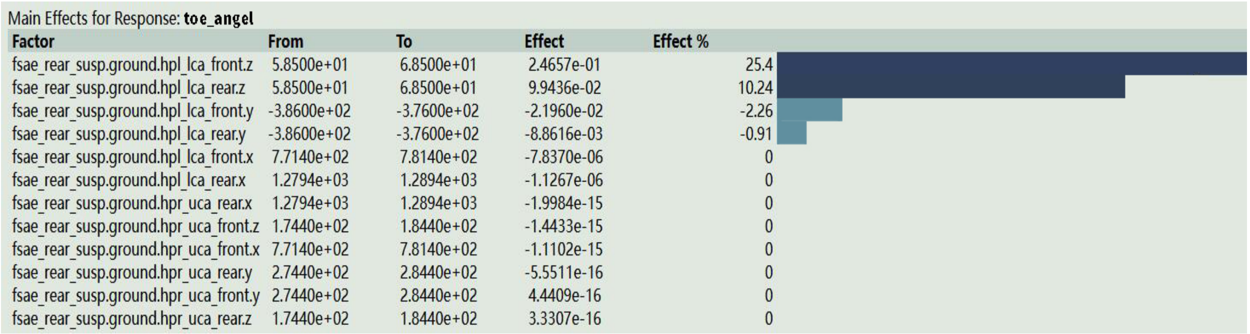


Figure 11: Change Curve of Center of Rear Suspension Roll.

3.3. Optimization of Rear Suspension Characteristic Parameters.

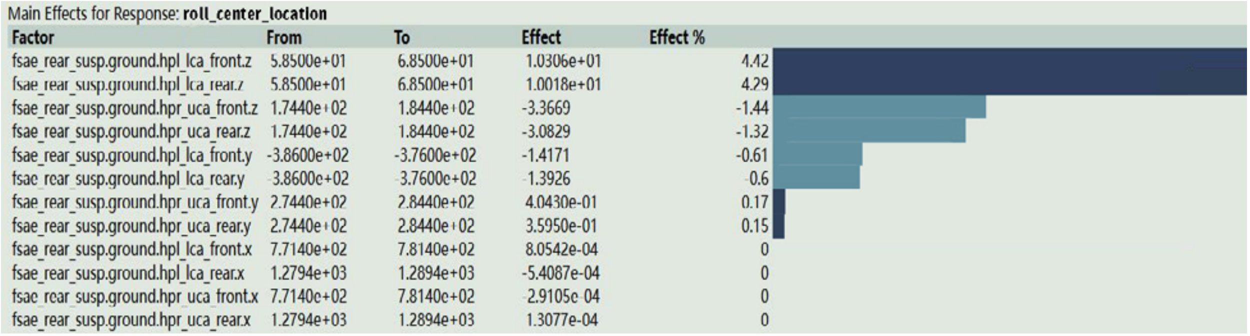
Through the above simulation analysis, it can be seen that the rear wheel front beam Angle varies widely, and the roll center height of the rear suspension is high when it is at rest. Therefore, the rear wheel front beam Angle and the roll center height of the rear suspension is selected as the optimization objectives. The Adams/Insight module in Adams software was used for analysis and optimization, and the coordinates of the four hinge points of the rear suspension, the lower transverse arm, and the frame were selected as the design variables, with a total of 12 coordinates, namely 12 design variables. Therefore, the 2^{12} iterative calculation was carried out to obtain the influence coefficients of each variable on the optimization table, as shown in the figure below.



Main Effects for Response: **toe_angel**

Factor	From	To	Effect	Effect %
fsae_rear_susp.ground.hpl_lca_front.z	5.8500e+01	6.8500e+01	2.4657e-01	25.4
fsae_rear_susp.ground.hpl_lca_rear.z	5.8500e+01	6.8500e+01	9.9436e-02	10.24
fsae_rear_susp.ground.hpl_lca_front.y	-3.8600e+02	-3.7600e+02	-2.1960e-02	-2.26
fsae_rear_susp.ground.hpl_lca_rear.y	-3.8600e+02	-3.7600e+02	-8.8616e-03	-0.91
fsae_rear_susp.ground.hpl_lca_front.x	7.7140e+02	7.8140e+02	-7.8370e-06	0
fsae_rear_susp.ground.hpl_lca_rear.x	1.2794e+03	1.2894e+03	-1.1267e-06	0
fsae_rear_susp.ground.hpr_uca_rear.x	1.2794e+03	1.2894e+03	-1.9984e-15	0
fsae_rear_susp.ground.hpr_uca_front.z	1.7440e+02	1.8440e+02	-1.4433e-15	0
fsae_rear_susp.ground.hpr_uca_front.x	7.7140e+02	7.8140e+02	-1.1102e-15	0
fsae_rear_susp.ground.hpr_uca_rear.y	2.7440e+02	2.8440e+02	-5.5511e-16	0
fsae_rear_susp.ground.hpr_uca_front.y	2.7440e+02	2.8440e+02	4.4409e-16	0
fsae_rear_susp.ground.hpr_uca_rear.z	1.7440e+02	1.8440e+02	3.3307e-16	0

Figure 12: Influence Coefficient of Design Variable on toe angle.



Main Effects for Response: **roll_center_location**

Factor	From	To	Effect	Effect %
fsae_rear_susp.ground.hpl_lca_front.z	5.8500e+01	6.8500e+01	1.0306e+01	4.42
fsae_rear_susp.ground.hpl_lca_rear.z	5.8500e+01	6.8500e+01	1.0018e+01	4.29
fsae_rear_susp.ground.hpr_uca_front.z	1.7440e+02	1.8440e+02	-3.3669	-1.44
fsae_rear_susp.ground.hpr_uca_rear.z	1.7440e+02	1.8440e+02	-3.0829	-1.32
fsae_rear_susp.ground.hpl_lca_front.y	-3.8600e+02	-3.7600e+02	-1.4171	-0.61
fsae_rear_susp.ground.hpl_lca_rear.y	-3.8600e+02	-3.7600e+02	-1.3926	-0.6
fsae_rear_susp.ground.hpr_uca_front.y	2.7440e+02	2.8440e+02	4.0430e-01	0.17
fsae_rear_susp.ground.hpr_uca_rear.y	2.7440e+02	2.8440e+02	3.5950e-01	0.15
fsae_rear_susp.ground.hpl_lca_front.x	7.7140e+02	7.8140e+02	8.0542e-04	0
fsae_rear_susp.ground.hpl_lca_rear.x	1.2794e+03	1.2894e+03	-5.4087e-04	0
fsae_rear_susp.ground.hpr_uca_front.x	7.7140e+02	7.8140e+02	-2.9105e-04	0
fsae_rear_susp.ground.hpr_uca_rear.x	1.2794e+03	1.2894e+03	1.3077e-04	0

Figure 13: Influence Coefficient of Design Variable on Roll Center Height.

According to Figure 12 and 13, it can be seen that the influence degree of Z coordinate of the four hard points on the optimization goal is significantly greater than that of X coordinate and Y coordinate. Therefore, the Z coordinates of the four hard points articulated by the upper and lower transverse arms and the frame were taken to optimize the two optimization objectives. The hard point coordinates before and after the optimization were shown in Table 2.

Table 2: Hard Point Coordinates Before and After Rear Suspension Optimization.

The design variables	Coordinates before optimization (mm)	Optimized coordinates (mm)
lca_front.z	63.5	61.1
lca_rear.z	63.5	61.1
uca_front.z	179.4	177.0
uca_rear.z	179.4	177.0

The optimized suspension model is simulated again and the optimized simulation curve is obtained. Figures 14 and 15 show the comparison of simulation curves of toe angle and roll center height before and after optimization. In the figure, the solid red line is the simulation curve before optimization, and the dotted blue line is the simulation curve after optimization.

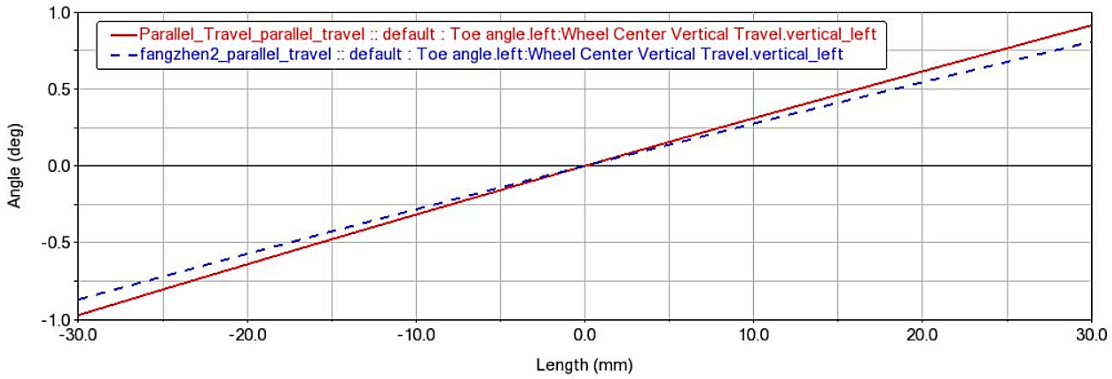


Figure 14: The Change Curves of Toe Angle Before and After Optimization.

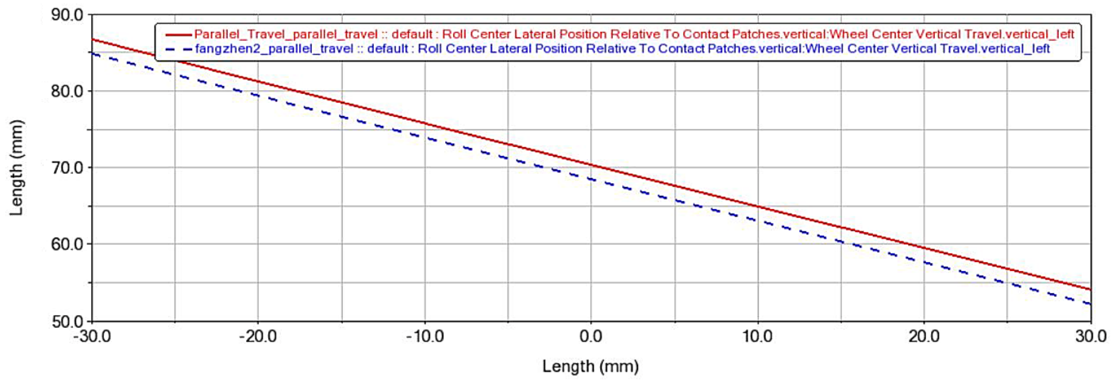


Figure 15: Optimize The Change Curve of Center Height of Forward and Rear Roll.

Through the comparative analysis before and after optimization, it can be seen that when the wheel is jumping up and down, the change range of the angle of the optimized rear wheel becomes smaller, the car is more stable, and the height of roll center decreases, so the performance of rear suspension motion after optimization is improved to some extent.

4. Finite Element Analysis of Rear Suspension of FSAE Racing Car.

4.1. Strength Analysis of Upper and Lower Arm of Rear Suspension.

4.1.1. Modeling of Upper and Lower Transverse Arms of Rear Suspension.

The upper and lower arm of the rear suspension are A-type fork arm structures, respectively in SolidWorks to establish the upper and lower arm model, and some simplification, in SolidWorks, to establish the model sequence is the point, line, surface, and then to the body. Then the Boolean operation in ANSYS modeling is used to process the model to prepare for the mesh division of the next model. The simplified model is shown in Figures 16 and 17:

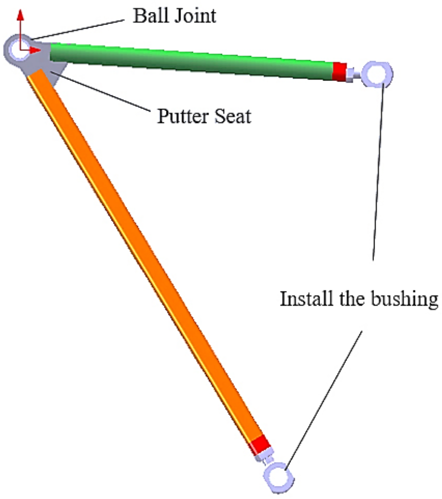


Figure 16: Model of the rear suspension arm.

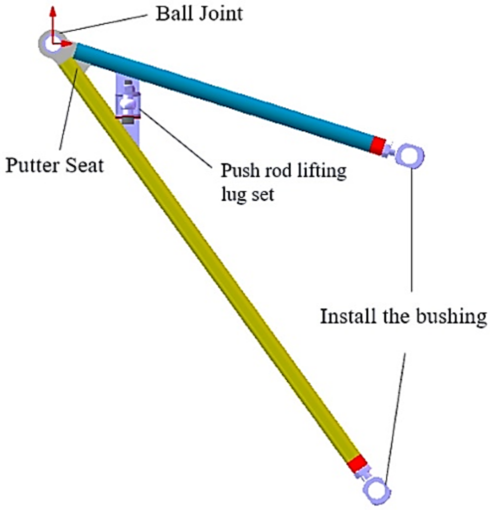


Figure 17: Model of the lower arm of rear suspension

4.1.2. Division of Upper and Lower Boom Grid of Rear Suspension.

Before meshing, material properties need to be set. The upper and lower arm of the rear suspension in this paper is made of carbon fiber, which is characterized by high strength, high modulus, low density, and lightweight, and is suitable for the lower arm of the FSAE racing suspension. The material properties of carbon fiber selected in this paper are: elastic modulus is $305GPa$; Poisson's ratio is 0.3; Density: $1.7g/cm3$; The yield strength is greater than $1200MPa$.

For the grid division of upper and lower transverse arms, the cell type is $SOLID92$, and free division is adopted. The side length of the cell is manually set to 5mm so that the grid division is more uniform and fewer singular units are generated. The finite element model is shown in Figures 18 and 19.

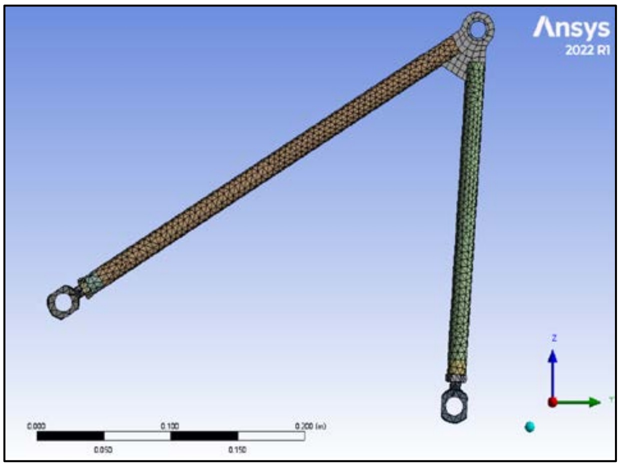


Figure 18: Finite element model of a transverse arm of rear suspension.

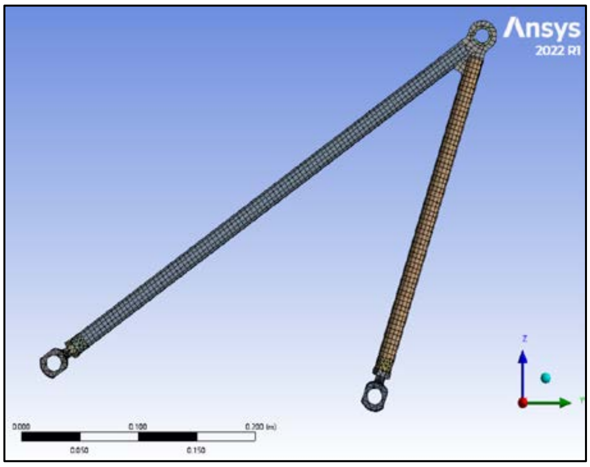


Figure 19: Finite element model of lower arm of rear suspension.

4.1.3. Results Analysis of Upper and Lower Boom of Rear Suspension.

The boundary conditions of the upper and lower arms of the rear suspension are the same, only the rotation motion in the x direction is retained at the installation of the bushing, and only the displacement constraint along the x、y、z direction is applied at the hinged seat. The finite element analysis assumes that the car body is moving, and the load forces exerted by the rear suspension arm at the bushing and the pushrod seat are $4800N$ and $5000N$, and the corresponding forces of the lower arm of the rear suspension are $5000N$ and $5200N$. After solving, the displacement deformation diagram and strain diagram of the rear suspension arm is shown in Figure. 20 and 21. It can be seen from the following figure that the maximum deformation displacement of the rear suspension arm is very small, and the maximum stress is $78.367MPa$ at the ball-hinged seat. Most of the stress of the upper arm is below $20MPa$, which meets the design requirements.

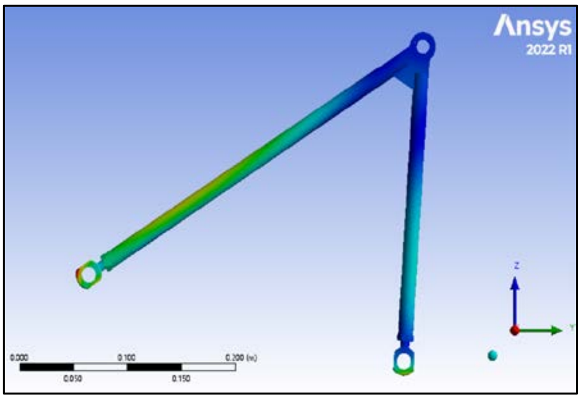


Figure 20: Deformation diagram of the rear suspension arm.

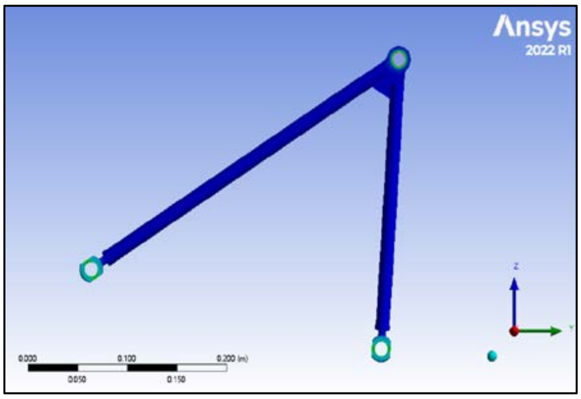


Figure 21: Strain diagram of the transverse arm of the rear suspension

The analysis results of the lower boom of the rear suspension are shown in Figures. 22 and 23. It can be seen from the figure that the displacement deformation of the lower boom of the rear suspension is also very small, with a maximum deformation of$0.005678mm$, which can be almost ignored. The maximum stress is $87.656MPa$, and most of the lower boom stress is less than $24Mpa$, in line with the design requirements.

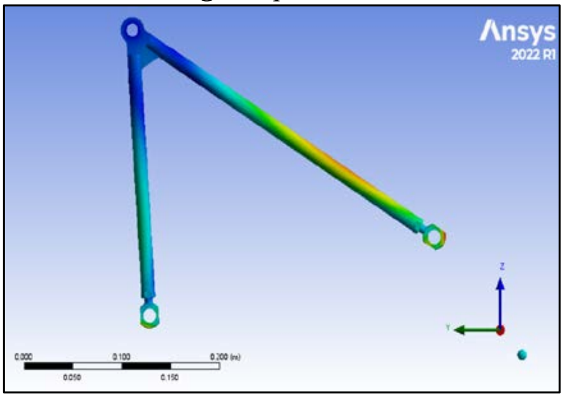


Figure 22: Deformation diagram of Lower arm of rear suspension

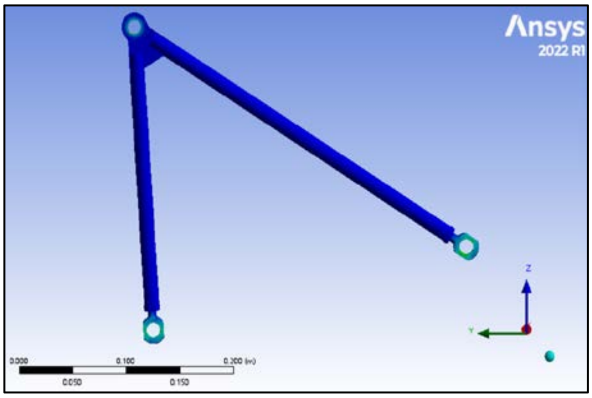


Figure 23: Strain diagram of lower arm of rear suspension

5. Conclusion

In this paper, the rear suspension model was firstly established by CATIA, and then the front beam angle, camber angle, wheelbase and lateral center height of this model were simulated and analyzed by Adams, and the front beam angle and lateral center height were optimized by Adams/Insight to improve the handling performance and stability performance of the car. Finally, the finite element analysis of the main stressed components of the rear suspension model is carried out by ANSYS to ensure that it meets the structural requirements in terms of both deformation and stress.

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