Design, Kinematic Modeling and Structural Analysis of a Double Wishbone Pull-Rod Suspension System for a Formula SAE Racecar

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Abstract

The primary goal of a Formula SAE (FSAE) suspension system is to maximize the tire con- tact patch under severe dynamic loading while keeping the unsprung mass as low as possible. This paper details the engineering approach behind the double wishbone pull-rod suspen- sion system developed for our FSAE race car. We bypassed basic theoretical models and uti- lized LOTUS SHARK software to iteratively optimize the hardpoints, targeting specific kinematic behaviors like controlled camber gain, roll center migration, and anti-dive/anti- squat characteristics. For the physical architecture, we balanced weight and strength by machining the wheel hubs and uprights from Al6063-T6 aluminum, while relying on high- carbon steel for the control arms. Complete load transfer calculations were performed for longitudinal and lateral acceleration scenarios, which fed directly into our finite element analysis (FEA) using ANSYS. By structurally optimizing the load paths, we achieved a fac- tor of safety above 2.1 across all primary sus- pension components. The final setup delivers highly predictable handling, minimal chassis roll, and improved overall tractability on tight autocross circuits.

Keywords: Formula SAE, Vehicle Dynamics, Sus- pension Kinematics, Double Wishbone, FEA, Un- sprung Mass, LOTUS SHARK.

1 Introduction

In the Formula SAE environment, building a powerful engine is only half the battle; getting that power to the ground and carrying speed through tight corners is what actually cuts down lap times. The suspension system is the mechanical bridge that dictates how the vehicle handles pitch during braking, squat during acceleration, and body roll during hard cornering.

For our vehicle, we decided to implement an inde- pendent double wishbone layout actuated by a pull- rod mechanism. The double wishbone gives us precise control over alignment parameters (like camber and caster) throughout the suspension's travel. Mean- while, packaging the damper actuation via a pull-rod allows us to mount the heavy DNM Burner RCP2 dampers lower in the chassis, actively dropping the vehicle's center of gravity (COG) and improving aero- dynamic flow over the sidepods.

This paper outlines our exact workflow: establish- ing the baseline vehicle targets, calculating the dy- namic load transfers, simulating the kinematics in LOTUS SHARK, and finally using ANSYS to shave off every unnecessary gram of material from the phys- ical components without compromising structural in- tegrity.

2 Vehicle Parameters and Load Calculations

Before drawing any suspension links or hardpoints, we had to establish the baseline parameters of the car. Assuming a total vehicle mass of 225 kg (includ- ing a 65 kg driver), we targeted a 45:55 front-to-rear weight distribution to improve rear traction. This puts 101.25 kg on the front axle and 123.75 kg on the rear.

2.1 Center of Gravity (COG)

Finding the exact COG height is critical because it acts as the moment arm for all weight transfer cal- culations. Based on our chassis packaging and driver seating position, the COG height was calculated at 228.6 mm from the ground. Our wheelbase was set at 1535 mm, with a front track width of 1195 mm and a rear track width of 1166 mm to aid in turn-in response.

2.2 Dynamic Load Transfer

Tires do not experience static loads on the track. We calculated the worst-case scenario load transfers as- suming a maximum tire friction coefficient (μ) of 1.5, representing the grip limit of our 18-inch BKT racing tires.

For longitudinal acceleration (assuming a 1.029g launch), the load transfer to the rear was calculated at 300.78 N. Under maximum braking (1.5g deceler- ation), the load transfer shifting to the front axle hit 438.28 N.

Lateral load transfer is what threatens suspension structural integrity the most. Assuming a 1.5g cor- nering force on a 100-meter radius, the total lateral weight transfer was ±561.9 N. Factoring in bump forces, we engineered the front suspension to handle a worst-case vertical design load of over 2393 N per corner.

3 Kinematic Geometry and Simulation

With the forces understood, we moved to LOTUS SHARK to design the suspension geometry. The goal here was to map out the hardpoints to control how the wheels move during bump, rebound, and chassis roll.

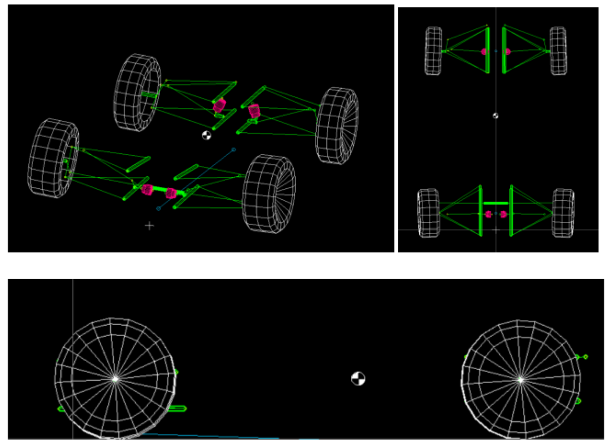

Figure 1: Front and rear suspension hardpoint geom- etry simulated in LOTUS SHARK.

3.1 Camber Gain

If the tire leans with the chassis during a corner, the contact patch shrinks and grip is lost. We designed the unequal length wishbones to induce negative cam- ber as the suspension compresses (bump). Our LO- TUS simulation verified a front camber gain ranging from -0.30° to -0.62° at 40 mm of bump. The rear, which handles the aggressive acceleration loads, fea- tures a steeper camber gain curve, reaching -1.18° at maximum compression.

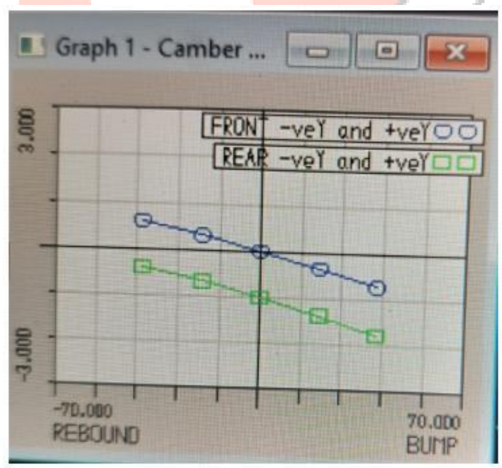

Figure 2: Camber angle vs. Bump/Rebound travel for front and rear axles.

3.2 Anti-Dive and Anti-Squat

To prevent the car's nose from diving violently under heavy braking, we angled the front wishbone mount- ing points to create a mechanical anti-dive geometry of 22–33%. Similarly, to prevent the rear from squat- ting and throwing off the aerodynamic balance during acceleration, the rear geometry was tuned to approx- imately 42% anti-squat.

Other critical alignment figures locked in during this phase included a caster angle of 4°, a kingpin inclination (KPI) of 8°, and a scrub radius tightly controlled at 36.64 mm to ensure stable, predictable steering feedback.

4 Component Design and Material Selection

Unsprung mass—the weight of the components not supported by the springs (wheels, hubs, uprights, brakes)—is the enemy of a responsive suspension. We tackled this by carefully selecting materials tailored to the specific stress profiles of each component.

4.1 Uprights and Wheel Hubs

The uprights (knuckles) and wheel hubs connect the rotating tire to the static wishbones. Because these parts represent a massive chunk of the unsprung mass, we machined them out of Aluminum 6063- T6. This aerospace-grade alloy provides an excellent strength-to-weight ratio and natural corrosion resis- tance.

4.2 Wishbones and Pull-Rods

For the control arms and the pull-rods themselves, rigidity is more important than absolute minimum weight to prevent buckling under compression. We utilized High Carbon Steel tubing. The bell cranks, which transfer the linear pull-rod motion to the damper at a 1:1.2 motion ratio, were fabricated from plain carbon steel due to its excellent fatigue resis- tance.

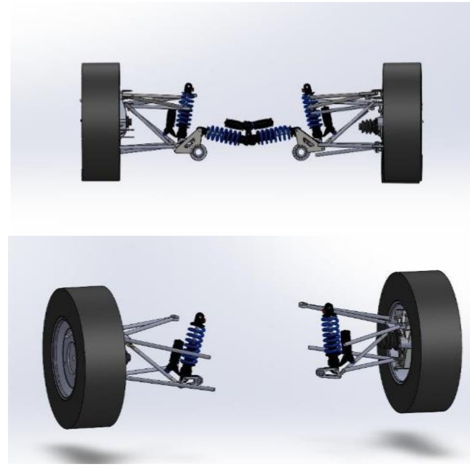

Figure 3: Complete CAD assembly of the pull-rod suspension system.

5 Structural Analysis (FEA)

Theoretical designs often fail in the real world due to stress concentrations. We imported our SOLID-WORKS models into ANSYS to simulate the extreme load cases calculated earlier.

The initial iterations of the uprights showed high stress concentrations near the brake caliper mounts and the lower ball joint pickups. By realigning the load paths and increasing the fillet radii in these crit- ical areas, we successfully dropped the peak stresses well below the material yield limits.

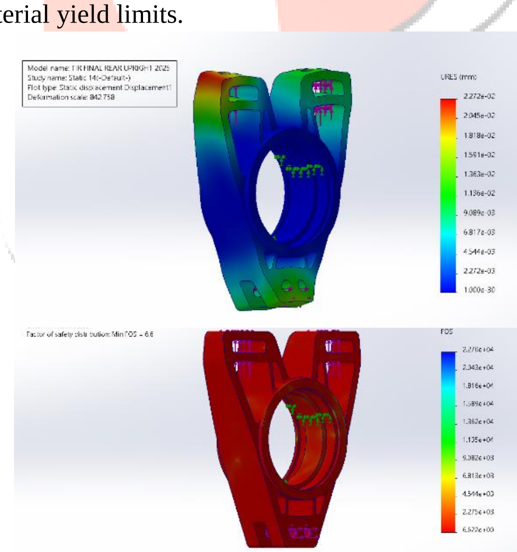

Figure 4: Von Mises stress distribution on the opti- mized rear upright under dynamic loading.

For the wheel hubs, the torsional load from the drivetrain and braking forces was the primary con- cern. FEA results for the Al6063-T6 front and rear hubs demonstrated uniform stress distribution, resulting in a Factor of Safety (FOS) of 2.3 and 2.2, respectively.

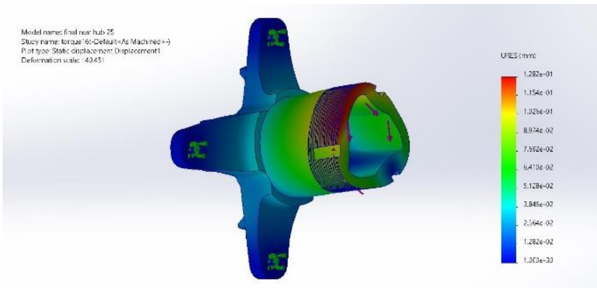

Figure 5: Structural analysis of the front hub showing minimal deformation.

The bell cranks were subjected to high pivot-point forces due to the leverage ratio of the pull-rod setup. Post-optimization, the front and rear bell cranks achieved an FOS of 2.3 and 2.5. Overall, through relentless topology adjustments, we kept the total weight of the front wheel assembly down to 9.5 kg per side, and the rear down to 6.5 kg.

6 Conclusion

The development of this double wishbone pull-rod suspension system represents a massive step forward in vehicle dynamics for our FSAE package. By heav- ily relying on LOTUS SHARK for kinematic map- ping, we dialed in the ideal camber curves, roll cen- ters, and anti-dive/squat geometries needed to keep the BKT tires planted on the tarmac. Pairing this ge- ometry with lightweight Aluminum 6063-T6 uprights and strictly validated FEA load paths ensures that the system isn't just theoretically fast, but physi- cally durable enough to survive the brutal endurance events of a Formula Student competition.

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