Turkish Online Journal of Qualitative Inquiry (TOJQI)
Volume 12, Issue 9, August 2021: 4693-4706

Design and Analysis of Suspension System for an FSAE car

K. Anirudh[a], K. Sriram[b], B. Jayanth[c], Dr. J. Anjaneyulu[d]

[a,b,c] Department of Mechanical Engineering, Vasavi College of Engineering
[d] Assistant Professor (Sr, Scale) in Department of Mechanical engineering, Vasavi College of Engineering
Email: j.anjaneyulu@staff.vce.ac.in

Abstract

The paper mainly focuses on enhancing the vehicle's overall performance while satisfying all the prerequisites of a double-wishbone suspension system with either pull rod or pushrod is used in a race car due to ease of design and lighter components. that deals with kinematic & compliance characteristics of the wheel suspension, including dynamic stability of the car. A suspension system performs the functions to maintain stability and balance during various conditions like pitching, rolling, yawing, squat, dive, to name a few, and to provide the vehicle with better control characteristics during acceleration, deceleration, and cornering.

This suspension model is simulated through the Parallel wheel and opposite wheel test and analyzed the half vehicle such that the gradient values are within the range. Later, to know the vehicle's dynamic characteristics, a complete vehicle analysis is done.

For designing the suspension models, assembly and component modeling CAD (SOLIDWORKS 2018) is used. In addition, ADAMS software has been selected for analysis for multi-body dynamics. For simulating various scenarios like Braking, constant radius cornering, drift, to name a few, a multi-body dynamic analysis is performed. In addition, the focus went on improving the suspension hardpoints so that they do not affect dynamically.

In the end, Ansys is used in analyzing the strength & life of components based on the loading characteristics of the vehicle.

Keywords: Kinematic & Compliance Characteristics, Rolling, Pull rod, Pushrod, Pitching, Strength & life of components.

Introduction

A Formula student race car is designed and built by the students for competitions like Supra SAE, FSAE. During racing, the suspension plays a significant role. Generally, a double-wishbone suspension with either pull rod or pushrod is used in a race car due to ease of design and lighter components. However, various pushrod and pull rod suspensions have been used in the front and the rear.

Generally, a Suspension system consists of springs, shock absorbers, and linkages that connect a vehicle to its wheels [1]. The four-bar linkage system is designed to increase articulation and decrease wheel wrap [3]. Initial design starts from the tire, which helps hold the vehicle on the road; then, a kinematic-based optimization is performed in Adams, which includes Parallel and Opposite wheel tests of a half vehicle.[4] [5] Next, components are modeled and then topologically optimized to reduce the weight, therefore, ultimately helps to reduce the sprung mass weight.

Adams simulates the behavior of the Full vehicle analysis during various test conditions over time and can animate its motion and compute properties such as wheel parameters, accelerations, forces, and many more. This helps in knowing the characteristic properties of the vehicle; so, if any changes are required, they can be modified and make it functional as per the racing conditions.[7][8]

Based on the loads obtained during the various tests in Adams, Ansys software is used for stress analysis of each component to know its life and sustainability as per requirements.[9]

Materials and methods

Materials

Aluminium 6061 T6 is selected for Uprights, Spring attachments, and Hubs. It is because of its high strength with low weight, and it is readily available in the market. Whereas, for A-arms, the material suitable for all the conditions, when compared its Mechanical properties (such as Flexural modulus E.I., buckling strength, etc.) and Physical properties (such as Density, U.T.S., Poisson ratio, etc.) of material which ultimately helps in improving the stiffness. This material is more potent and yet more ductile. However, it exhibits better welding properties which help in the manufacturing of the chassis with ease. Overall, it satisfies the minimum material requirement mentioned in the rule book, which ultimately motivated us to choose the material.

Design Methodology

Hardpoints determination

The initial design procedure begins with selecting proper tires, which are in direct contact with the road and chassis. Here, Hoosier 43070 was chosen, as it was one of the tires with minimal outer diameter. As a result, it results in better vehicle acceleration and has the added advantage of reducing weight. However, a tire with an even smaller size would have reduced space for suspension components; therefore, 43070 was selected.

Initially, while choosing wheelbase, as per the rules, it needs to be 1525mm, considering the larger the wheelbase, the better is handling. But, on the other hand, if the wheelbase becomes very big, the load transfer is very low, resulting in higher brake force. So, after multiple iterations, it is selected as 1560mm.

Then, track-width was chosen based on the packaging of the components, and the lateral load transfer the shorter the track, the higher is load transfer to the outer wheel during cornering. After multiple iterations, this results in a better grip for the outer wheel; track width 1150 mm is chosen.

During finding suspension Hardpoints, the motion of suspension is dependent on the planes formed by the A-arms. So, for the selection of plane, the front view IC, Sideview IC, and ball joints of upright are considered.

After selecting planes, the lines are created on the planes originating from the ball joints towards the chassis to get the A-arm chassis hardpoints, as shown in figure 1 below.

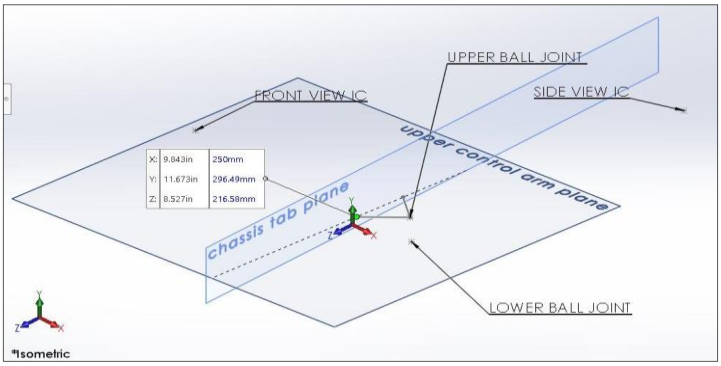


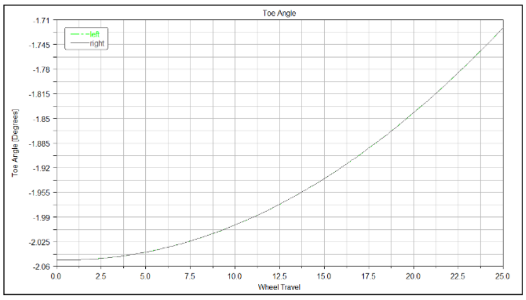
Figure 1 Determination of Hardpoints

Kinematic Analysis

Here, hardpoints are taken as an input from the Solidworks so that the points in the front and rear sub-assemblies are tested individually, as shown below.

Parallel Wheel Test

In the parallel wheel test, the maximum spring travel in straight line vehicle motion, which is considered as either 25mm Jounce or 25mm Rebound (as per rulebook) and simulated the subassemblies. But when tested the vehicle dynamically, the maximum spring travel obtained is 15mm during the skid pad test at a speed of 60kmp/hr (which is an extreme level of testing for the vehicle) so, iterations are made in such a way to get a minimum toe, camber variation. Here in the below graphs starting from Toe vs. Wheel Travel, the gradient



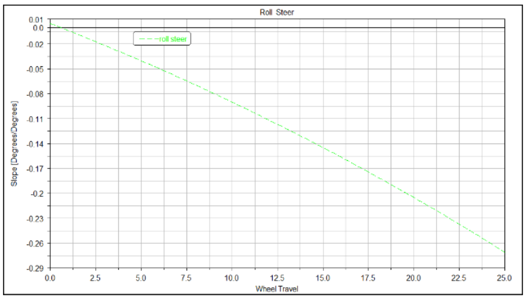
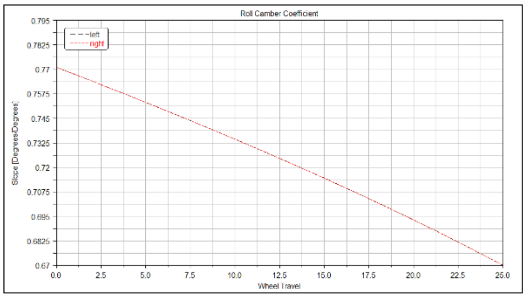


Figure 2A Wheel Travel vs. Toe Angle (Degrees) **Figure 2B** Wheel Travel vs. Slope (Roll Steer)



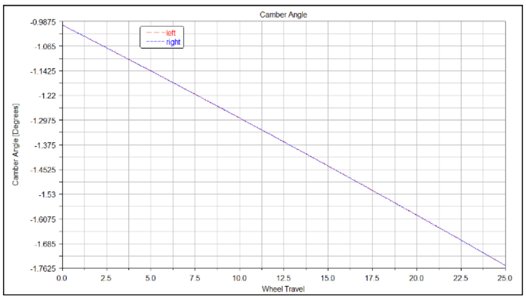
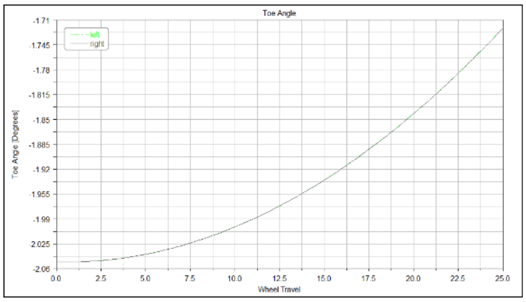


Figure 2C Wheel Travel vs. Slope (Roll Camber Coeff) **Figure 2D** Wheel Travel vs. Camber (Degrees)

Here, the minimum variation toe and camber angle helps in straight-line motion and braking conditions.

Opposite Wheel Test

It is considered that the maximum spring travel in cornering the vehicle motion is either 25mm as Jounce or 25mm Rebounce (as per rulebook) and simulated the subassemblies in the opposite wheel test. But the maximum spring travel obtained is 15mm during the skid pad test at a speed of 60kmp/hr (which is an extreme level of testing). So, here are the below graphs



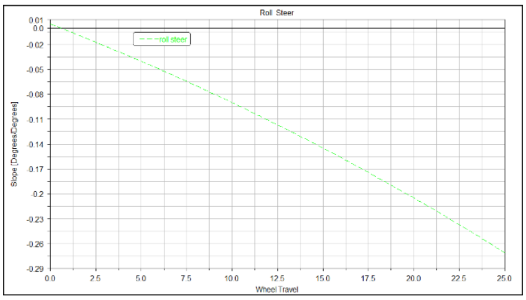
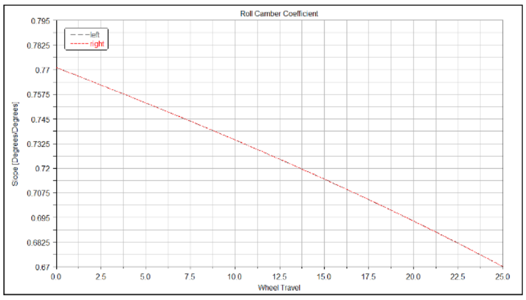


Figure 3A Wheel Travel vs. Toe Angle (Degrees) **Figure 3B** Wheel Travel vs. Slope (Roll Steer)



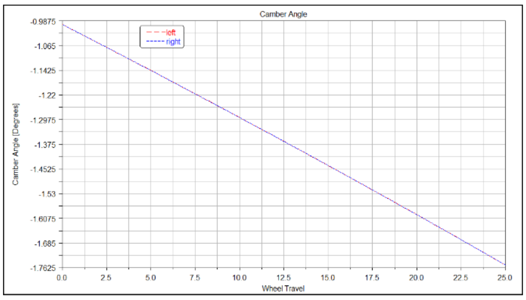


Figure 3C Wheel Travel vs. Slope (Roll Camber Coeff) **Figure 3D** Wheel Travel vs. Camber (Degrees)

As the wheel travel increases, the toe variation is nearing zero. Therefore, it helps in cornering the stability of the vehicle. Here the graphs of the slope are minimum which is a good sign of stability of the vehicle; in addition, the camber variation is also very minimum, which can be accommodated.

Design Methodology

The following are the suspension components to be designed.

- Upright
- Hub
- Spring attachments
- A-arms

The initial phase begins with the essential input given for the initial modelling of the upright is the suspension hardpoints. So, a solid elemental block was put initially with covers all the ball joints.

The groove is cut for placing two Taper roller bearings inside the upright. Here, based on the tolerance ranges of taper roller bearing, barrel (hole) diameter is taken and made it interference fit so that it tightly holds the upright.

Now cut the portion of the bracket to hold the a-arms at the upper and lower ball joint. Here in the design, the upper bracket is entirely detachable, which can be easily maintained and replaced at less cost. Here the base is also topological optimized to reduce its weight.

Coming to hub PCD of attachment holes of rims and brake disc is input to design it. To the barrel portion of the hub diameter, tolerances are given to obtain Interference fit for practical use. In the end, assembly is made with proper clearance between components and manoeuvrability for replacing the parts. The entire design is an iterative process of checking the functional aspect of the calliper, brake disc, and hub. So, based on it, the final design of assembly is obtained

Initially, three points of spring attachment are taken as input. Here are spherical bearings are used in the bell crank near the attachment portion.

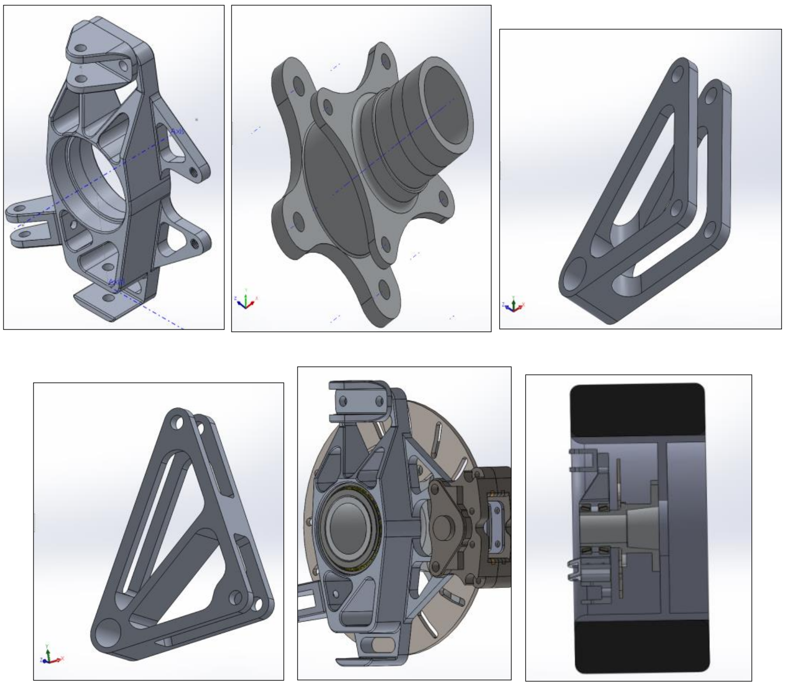


Figure 4 Suspension Components 3D models and Assembly

Numerical load calculations

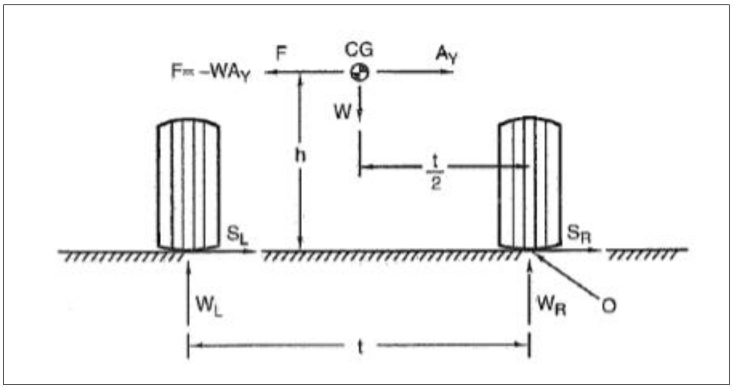
Many load cases wheel experiences during static and dynamics conditions; some of them are listed here:

- Static weight distribution
- Lateral load transfer (cornering effects)
- Longitudinal load transfer (acceleration and deceleration)
- Bump forces
- Frictional force
- Centrifugal force
- Grade `
- Aerodynamic forces

Static Weight Distribution: -

Lateral Load Transfer: -

Load on the outer wheels increases; this is due to the inertia of the mass of the vehicle. The increase on outer wheels reduces the load on inner wheels. this acts in the Y direction.



- [7]

Figure 5 Lateral Load Transfer

First, we need to determine the centrifugal force, which is given by the formulae $Fc = M*V2/R = 280*(12.5)^2/5 = 8750$ N

Now taking moments about point "0"

$W_l*t - W*(t/2) - F*h = 0$ - [7]

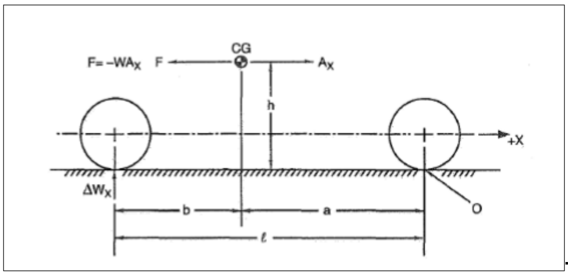
$W_L - (W/2) = \Delta W = 1467.35$ N

(ΔW_L) lateral load transfer= 964.12 N

(ΔWr) lateral load transfer = 1671.28 N

Longitudinal load Transfer: -

Here longitudinal load transfer happens during both acceleration and deceleration, but acceleration has lower value compared to deceleration as both cannot be simultaneously happening in a vehicle, we will consider both braking here +x is towards forward of vehicle this acts in Y direction.



- [7]

Figure 6 Longitudinal Load Transfer

Here $A_x = A_d = -1.5\ g = -14.715$ m/S2

Taking moments about "O"

$\Delta W_x * l = h * M * A_x$ - [7]

$\Delta W_x = 245.44 * 280 * (-14.715)/1540$

$(\Delta W_b)_{rear} = -412.7599$ N

$(\Delta W_b)_{front} = 412.7599$ N

Similarly, while accelerating

$(\Delta W_a)_{rear} = 609.75$ N

$(\Delta W_a)_{front} = -609.75$ N

Bump Force: -

The resisting force that the wheel produces when it encounters a bump is the bump force. this acts in the Y direction.

$(WF)\ bump = k * S_T (F) = 78.88 * 35 = 2760.8$ N - [7]

$(WR)\ bump = k * S_T (R) = 78.88 * 30 = 2366.4$ N - [7]

Frictional Force: -

The force induced when the wheels are locked is dependent on the braking torque; this acts in the X-direction.

$(FF_f) = T_f / R = 404.6 * 1000/207.1 = 1953.94$ N - [7]

$(FF_r) = T_r / R = 434.86 * 1000/207.1 = 2099.80$ N - [7]

Centrifugal Force: -

For this force, it is considered as the wheels are stationary and the sprung mass of the body is producing centrifugal force due to flexible connection between these acts in the Z direction

$$(F_{c})_f = (M_s + M_d) *V2/R = 3900N$$

$$(F_{c})_r = (M_s + M_d) *V2/R = 4225N$$

Taking all the forces acting in the Z direction are taken to get total force in the vertical direction.

$$\sum Z_{\text{front}} = (W1)_{\text{static}} + (\Delta W1)_{\text{lateral load transfer}} + (\Delta Wx)_{\text{front}} + (WF)_{\text{bump}} \quad - [7]$$

$$\sum Z_{\text{rear}} = (W3)_{\text{static}} + (\Delta Wl)_{\text{lateral load transfer}} + (\Delta Wx)_{\text{front}} + (WF)_{\text{bump}} \quad - [7]$$

A centrifugal force acts in the Y direction vehicle, and in the X direction, frictional forces act oppositely. Braking torque is taken as input from the braking team

Load type	Front	Rear	Direction to the global coordinate system
Static	1248.00 N	1352.00 N	Z
Braking load transfer	323.50 N	-323.50 N	Z
Bump Force	2760.80 N	2366.40 N	Z
Lateral load transfer	964.12 N	1671.28 N	Z
Frictional force	1953.94 N	2099.80 N	X
Centrifugal force	3900 N	4225 N	Y
Braking torque	190 N-m	110 N-m	My
Longitudinal Load Transfer	609.75 N	-609.75 N	Z

Table 1 Theoretical loads acting on wheels

Direction	∑ Front	∑ Rear
X	832.68 N	842.75 N
Y	3900	4225 N
Z	5582.67	5999.43 N
My	190 N-m	110 N-m

Table 2 Summation of loads in three directions

Results and discussion

Dynamic Analysis

The initial phase begins by providing inputs to the Full vehicle in Adams. So, to evaluate the vehicle's performance in various test conditions that FSAE performs during the event and to know its performance, tested a few of them, like, Fishhook test (Skid pad), Constant radius cornering, Brake test, Braking during cornering. This evaluation method helped know the critical parameters such as wheel kinematics, loads, acceleration, and many more. Furthermore, it helped change the parameters as per the requirements and improve its performance to perform the best during the event.

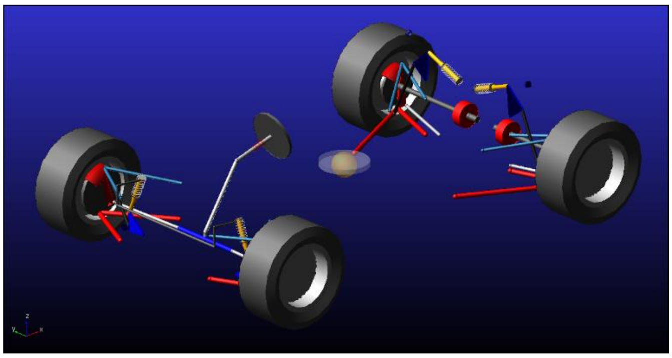


Figure 7 Adams Full Vehicle Assembly

This method helped to know the characteristics parameters of the vehicle and its extremities; this helps to train the driver during vehicle testing.

Here are the test results: -

Fishhook test

This simulation replicates the skid pad dynamic event where the vehicle is driven virtually in the track of shape eight. This test is a comprehensive experiment of evaluating the vehicle dynamic anti-rollover propensity. The fishhook test method: the vehicle is driven straight at various speeds starting from 20km/hr to 60km/hr. At each speed, the main motto is to improve the vehicle's performance. Here, the saturation points where the vehicle is getting unstable is at the speed of 40km/h. However, it is tested until 60km/hr to know the extent of the vehicle's instability. So, 40km/hr as an input speed on the proving ground, with first steer input in one direction, after 3seconds of the first turn then, reversed the steering angle by 180° and continues for 5 seconds for the completion of one lap on the skid pad track.

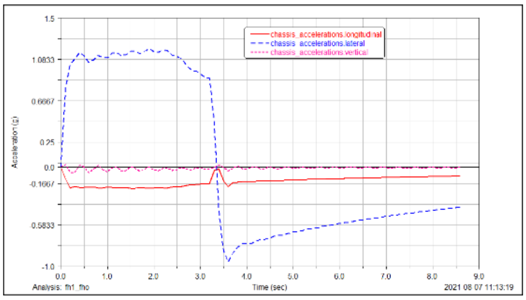


Figure 8A Chassis Accerleration in 3-Directions

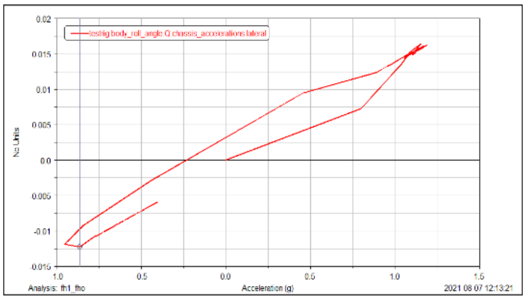


Figure 8B Roll angle vs. Lateral Accerleration

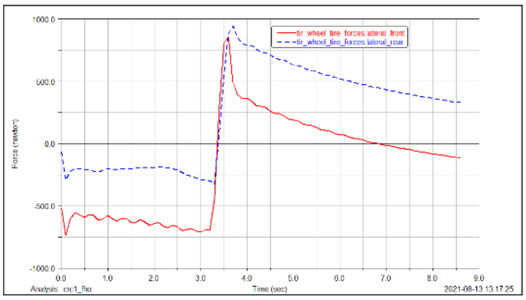


Figure 8C Maximum Lateral force on Wheels

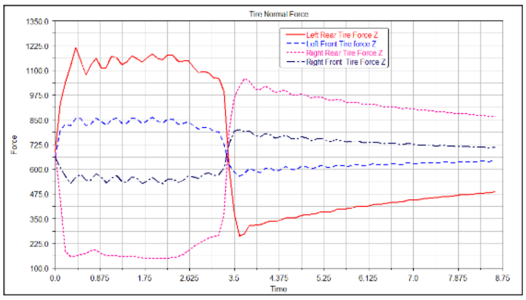


Figure 8D Maximum Normal Force on Wheels

As mentioned above, the fishhook test is performed at various speeds; here, during 3-5 seconds of time interval, the vehicle is getting oversteered (which tends to skid the vehicle), or else it is understeered. So, it was restricted such that the speed was decreased to 30km/hr as shown below.

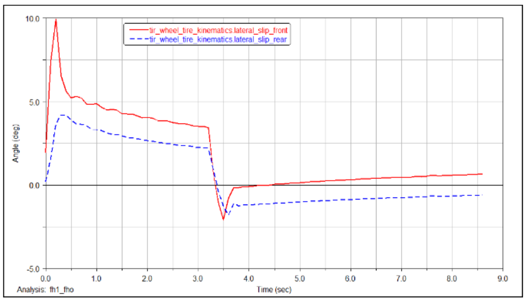


Figure 8E Lateral Slip Speed 50 km/hr

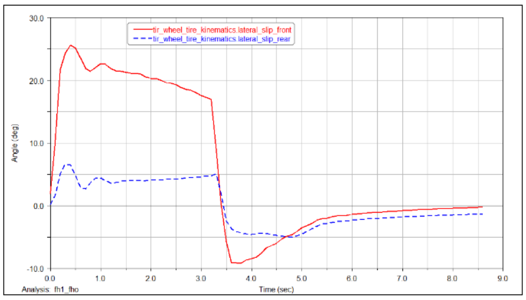


Figure 8F Lateral Slip Speed 30 km/hr

Constant Radius Cornering

The simulation is perhaps the most important because it recreates the cornering where the suspension mechanism is forced to the limit. The simulation is of the vehicle under a constant radius cornering. The performance parameters considered were the vehicle's side slip angle and tire's normal forces.

This simulation is considered because it examines the stability, it deals with the lateral load shifts of the vehicle during cornering, steering characteristics of the vehicle. The below are the normal loads acting on the tire.

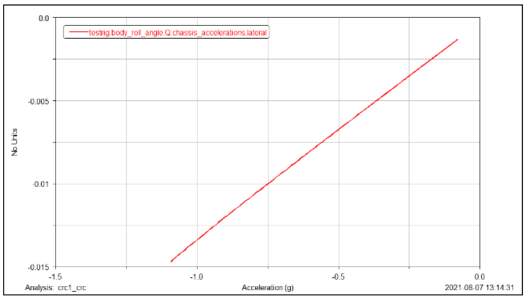


Figure 9A Lateral Accerleration

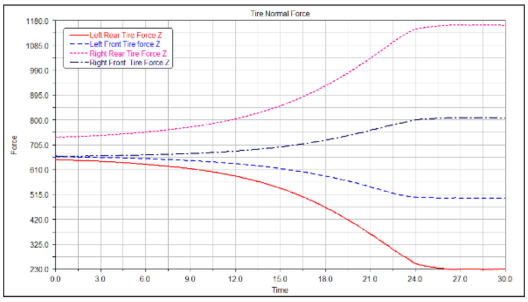
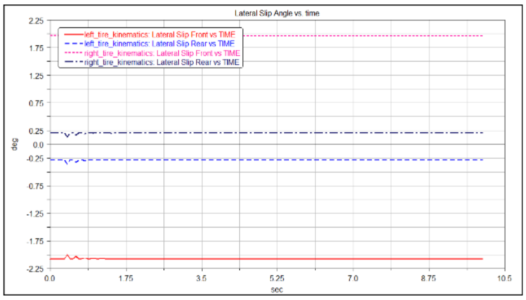


Figure 9B Normal Force on Tires

Straight Line Brake test

The complete vehicle braking evaluates the dynamic response of the suspension under a straight-line braking event. The deceleration effect while Braking is 1.5G. During this simulation, critical parameters are regular forces for front and rear tires and the oversteer/understeer effect. In addition, it also helps in knowing the braking torque and stopping distance of the vehicle.



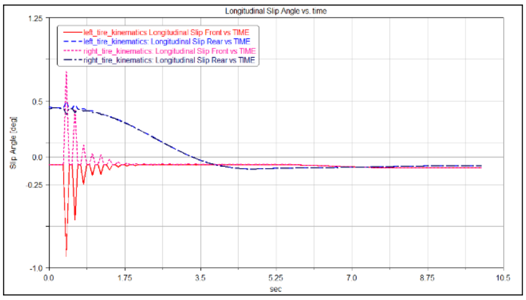
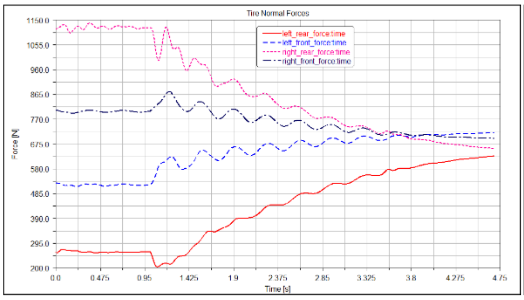


Figure 10A Lateral Slip of All Tires

Figure 10B Longitudinal Slip of All Tires

Braking during Cornering

This test is a combination of cornering and Braking. Here cornering acceleration is taken as 1G and the deceleration effect is 1.5G during the cornering, where the extreme loads are fallen on the wheel with extreme kinematic variations. Here, parameters are varying in this test; the variation is not as much when compared to the fishhook test, so loads are only shown below,



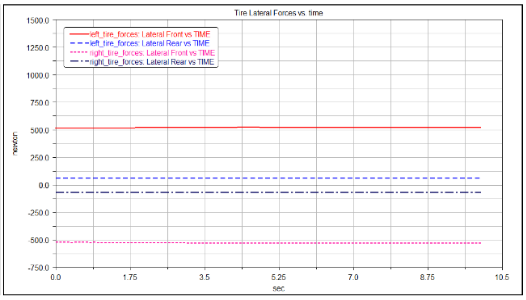


Figure 11A Normal Force on Tires

Figure 11B Lateral Force on Tires

Analysis

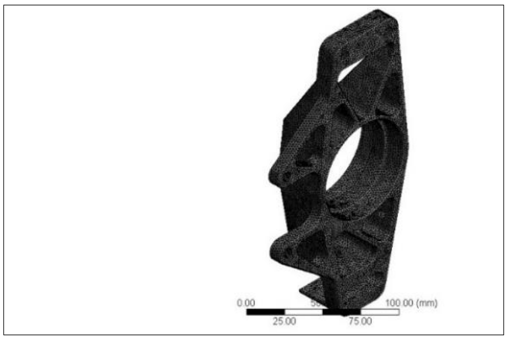
Upright Analysis

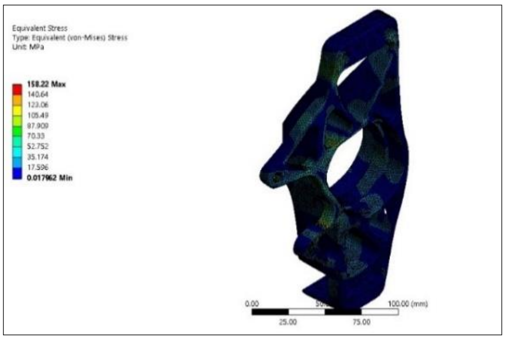
An upright base component is a non-rotary component that connects the hubs to the rest of the wheel assembly; it has two bearing faces for the bearings and holes for connecting the clevis and the A-arms.

Fixtures – Fix the Clevis mounting holes

Forces – The two bearing forces are subjected to forces obtained

Torque – The brake calliper mounting holes





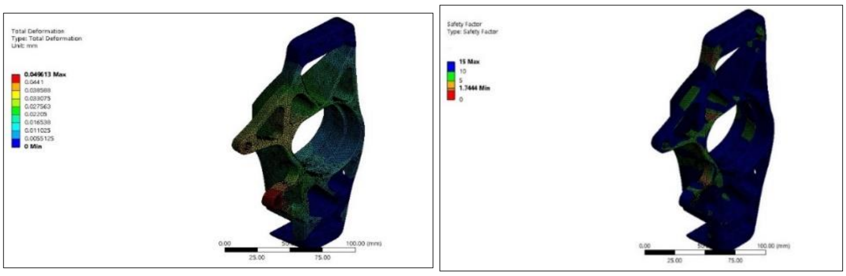


Figure 12 Upright ANSYS Results

Hub Analysis

A Hub is a rotating component, the first connection between the rims and the wheel assembly, so it is of utmost importance to be strong and light to reduce unsprung mass and improve the vehicle's handling. Moreover, it's a rotating component; its life (in cycles) is found.

Fixtures – Two bearing faces are fixed
Forces – The wheel petal holes are subjected to forces obtained
Torque – The brake disc mounting holes

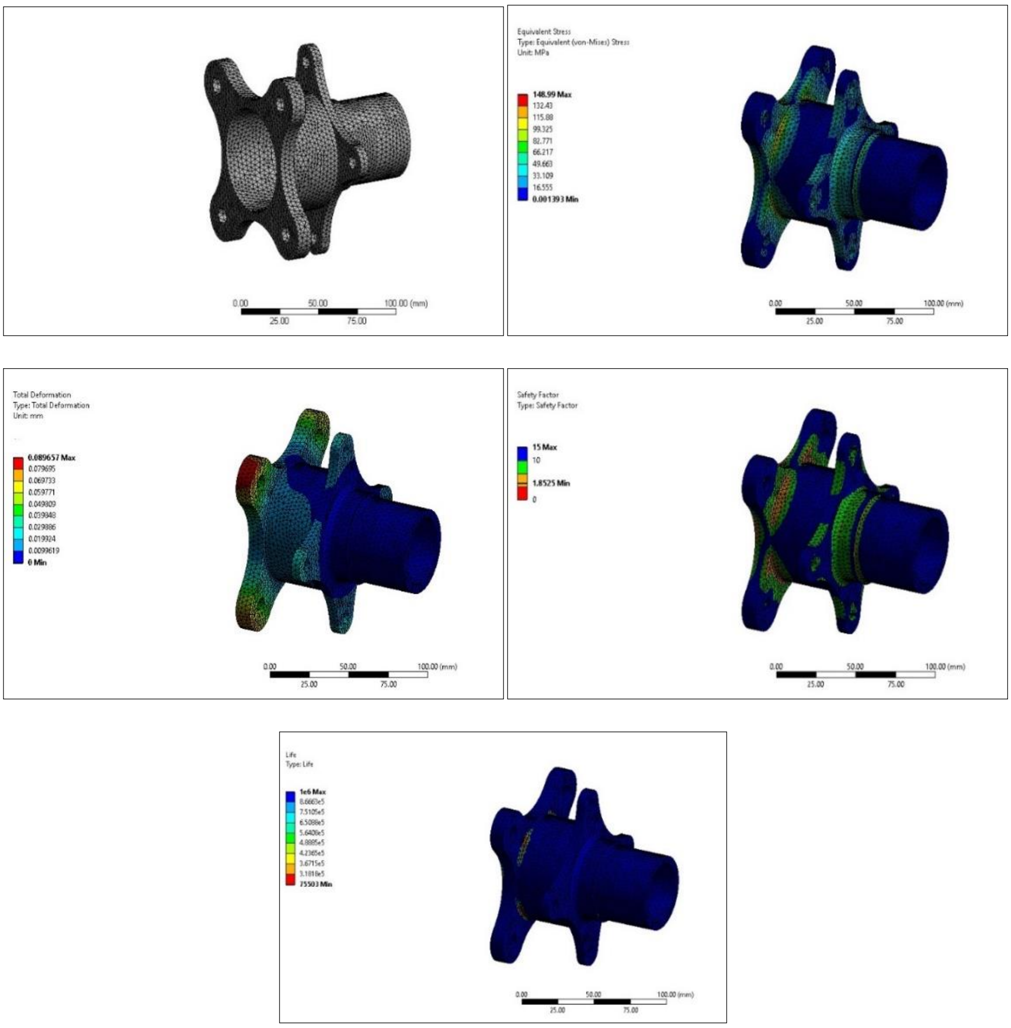


Figure 13 Hub ANSYS Results

Spring Attachments Analysis

The bell cranks transfer forces from the wheels to the springs and returns the force to rebound the wheel, so it needs to withstand both compression and tension loads from the wheel and the spring.

Fixtures – The bearing bore is kept as cylindrical support, the push/pull rod mounting hole is kept as Fixed support.
Forces – The spring mount is kept as the force location along the spring orientation direction

Front –

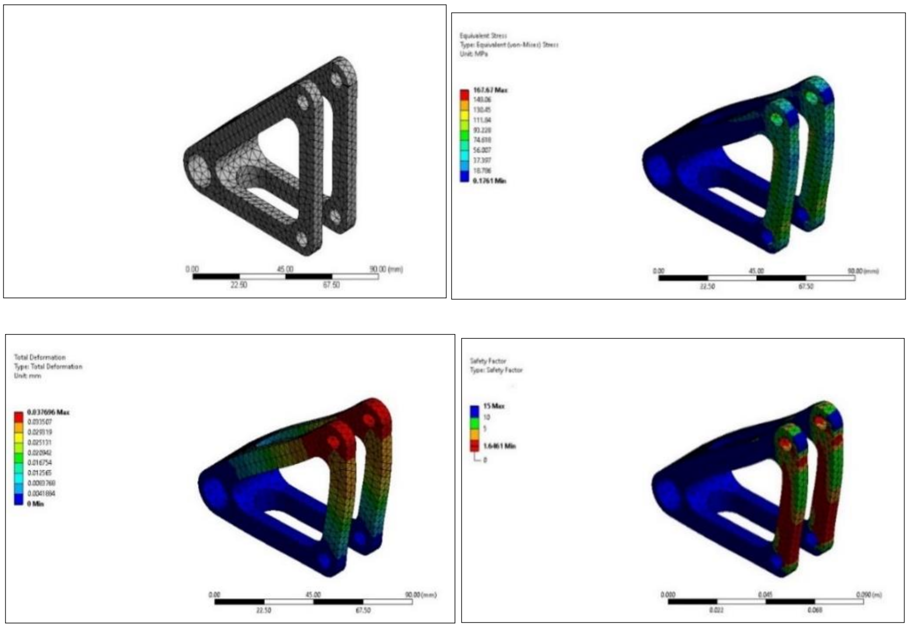


Figure 14 Spring Attachment ANSYS results (Front)

Rear –

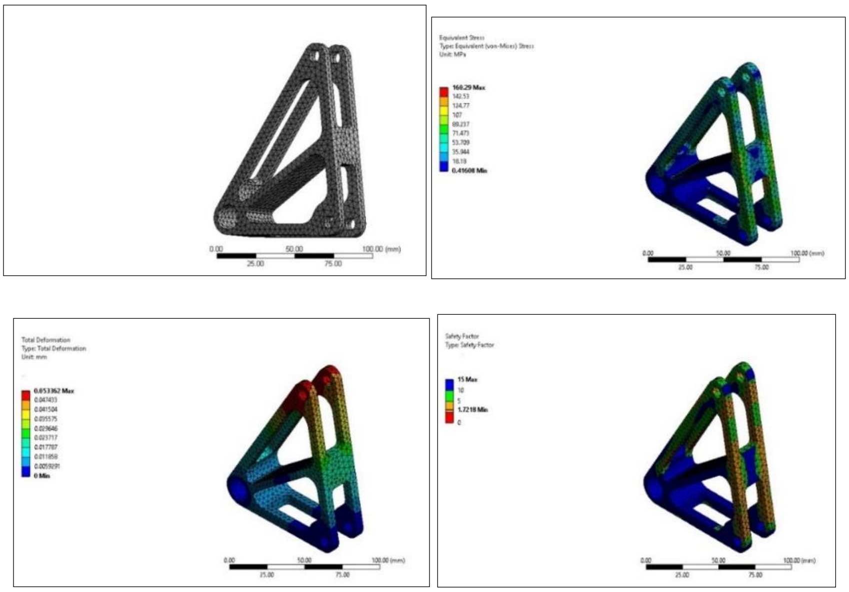


Figure 15 Spring Attachment ANSYS results (Rear)

Conclusions

- The optimum weight of wheel assembly (i.e., 18kgs) is obtained. Furthermore, the safety factor is improved from 1.27 to 1.5 (Minimum value) by upgrading weaker sections.
- The good life of components is obtained, which ultimately helps while testing the vehicle.
- The height of the Centre of gravity was reduced.
- During the skid pad test, controlled oversteer is obtained, which helps in training the driver accordingly.
- Proper stopping distance is obtained during the brake test.
- Clevis brackets are used for tuning wheel aligning parameters.

Hence, the project of designing and analysing the double-wishbone suspension system has been systematically executed. During the literature survey, the type of suspension system and the actuation have been thoughtfully chosen. Par modelling and assembling have been done. After designing, the dynamic simulation has been carried out in the Adams software 2021 version by considering all the required loads, and the vehicle body dynamics have been analysed. Besides, it is to be noted that "Iteration is the key to perfection." The design is modelled and analysed through various static and dynamic simulations by the problem statement such that it maintains the following parameters:

- They are preventing road shocks from being transferred to the vehicle.
- They are preserving the stability of the vehicle during cornering.
- It safeguards the driver from road shocks.
- It was maintaining good traction during driving, cornering, and Braking.

Acknowledgments

We want to express a deep sense of gratitude and heartfelt thanks to our guide Dr. J. Anjaneyulu, Assistant Professor of Mechanical Engineering Department, for his valuable suggestions and directions towards the execution of this project.

References

1. Samant Saurabh Y., Santosh Kumar, Kaushal Kamal Jain, Sudhanshu Kumar Behera, Dhiraj Gandhi, Sivapuram Raghavendra, Karuna Kalita*, Design of Suspension System for Formula Student Race Car, 12th International Conference on Vibration Problems, ICOVP 2015, p.1138-1149.
2. Michael Gifford, Tanner Landing, Cody Wood, Design and Manufacture of an Adaptive Suspension System, April 30, 2015, p.15-30.
3. Canyi Du, Baochai Zhu, Hengbo Wang, Xingye Mai, Taixing Qin, Design and Optimization of Front Suspension of FSAE based on ADAMS Simulation, Proceedings of the 2016 4th International Conference on Sensors, Mechatronics and Automation ICSMA 2016, p.1-6.
4. RahulSindhwani AyanBhatnagar AbhiSoni AyushmanSisodia Punj LataSingh VipinKaushik SumitSharma, Design and optimization of suspension for formula Society of Automotive Engineers (FSAE) vehicle, Material today, Vol 38, Part 1, 2021, p.229-233.
5. C. Kavitha, S. Abinav Shankar, B. Ashok, S. Denis Ashok, Hafiz Ahmed, Muhammad Usman, Adaptive suspension strategy for a double wishbone suspension through camber and toe optimization, Engineering Science and Technology, an International Journal, 2018, p.1-8.
6. Thomas D. Gillespie, Fundamentals of Vehicle Dynamics, Society of Automobile Engineers, 1995, p.195-237.
7. William F. Milliken, Douglas Milliken, Race Car Vehicle Dynamics, 1995, p.665- 704.
8. SUPRA SAEINDIA, STUDENT FORMULA SAE Rule Book, 2019, p.14-44.
9. Smith, C., Racing Chassis and Suspension Design, society of Automotive Engineers, 2004,