Organized by SAEINDIA Society of Automotive Engineers INDIA

BAJA SAEINDIA 2022

Under the aegis of NATRAX CHITKARA UNIVERSITY CAMBRIDGE GROUP OF INSTITUTIONS

CAE Evaluation Report

Team ID 22153 || Team Racers IITJ || IIT Jodhpur

racers@iitj.ac.in

The purpose of this report is to analyse the strength, durability and applications used to build our vehicle. Report consists of a CAE analysis and optimization study. CAE analysis is to make safe designs obeying all the rules and regulations set by the SAE BAJA. Optimization was performed in terms of design, the strength of the vehicle and driver safety. SolidWorks has been used to design the vehicle, Lotus has been used to perform the multi body dynamics and static structural module of Ansys has been used for the purpose of simulations.

1. Introduction

The purpose of designing and manufacturing a Baja car was to create a prototype recreational off-road vehicle that could provide a fun, safe, and reliable experience for a weekend off- road vehicle enthusiast. To accomplish this task we design, develop and fabricate an All-Terrain Vehicle in accordance with the rulebook of BAJA 2022 given by SAE. Material of roll cage, knuckle, hub and gearbox is selected based on the strength, cost and availability. The roll cage is designed to incorporate all the automotive subsystems. The design and development process of the roll cage involves various factors; namely material selection, frame design, cross-section determination and finite element analysis.

Suspension is designed to provide a high level of comfort to the driver. Spring, damper, suspension linkages together form a suspension system. Shock absorber plays an important role in eliminating shocks and providing comfort.

Braking system works by converting the kinetic energy of the vehicle into heat energy. Usage of inboard braking systems, helped us in achieving complete locking of the four wheels. The Brake Circuit consisted of the Master Cylinder, Stainless Steel braided Brake lines and the calipers. The Master cylinder was chosen considering the respective brake calculations and the requirements of the vehicle. Stainless steel braided brake lines were chosen to minimize pressure loss throughout the circuit as they ensure no sharp bends in the brake line. The calipers were chosen with respect to the brake calculations.

Our goal is to design a transmission system which provides high torque and moderately high speed. To achieve this two criteria have to be fulfilled which include ability of vehicle to climb 35 degrees gradient and achieve maximum speed of 40 kmph.

The power train of ATV which includes BLDC motor, controller, battery and transmission, provides the required tractive effort as well as acceleration under restricted speed conditions. BLDC motor should not exceeds of 6KW, together with of battery 48V, 110 Ah Li-ion battery (as per the rules of competition), space constraints and the various conditions of terrain type, a suitable transmission system is selected its parameters are designed, which is largely responsible for major vehicle performance parameters.

2. Objective

1. Main aim is to obtain the factor of safety (FOS) for:
    - All impact cases
    - Fatigue & stress analysis of knuckle, hub and wishbone
    - Total deformation, fatigue & stress analysis of gearbox
2. To show the strength and durability of our vehicle.
3. To show all the calculations considering all the possible scenarios.

3. Methodology

- Solidworks is used for designing the roll cage because of it's easy and compatible interface.
- 3D points were then imported from Solidworks into excel as a .txt file which was then fed to Design Modeler.
- In Design Modeler, points were joined to get the required cross-section and 1D meshing was carried out and all impact cases were tested.
- Interactions were performed by analyzing contours of max. deformation & max. combined stress to get the FOS of 1.2 or above.
- All 4 steps are shown in fig. 1 a to d

3.1 Meshing

We have done 1D meshing of the roll cage, and the results from the 1D meshing are quite satisfactory. Mesh size of 2mm is selected based on the convergence study as shown in fig. 2.

- Meshing of the knuckle, hub and wishbone are done using triangular mesh.
- Meshing of brake discs is done using tetrahedral mesh.
- Roll cage meshing is done with the rectangular mesh.

We used tetrahedral meshing because for 3D meshing tetras are preferred over hexas for complex geometries. The triangular element is called a constant strain triangle (CST) element. The value of the stress tensor in the whole element will be constant. This may not be true in real situations. Thus, 'tri', are preferred if the geometry is very complex, due to difficulty in meshing with 'quad' elements.
Hexahedral mesh is preferred near the regions of high-stress gradients i.e. near the holes. For this, we have used the inflation layer near the holes.
Total number of nodes and elements in rollcage, knuckle, and hub are shown in Table 1 under the conclusions. The elements are obtained by doing a convergence study for all the components. Fig. 3. shows the convergence study for the mesh for the knuckle. The mesh for the hub with an inflation layer near the holes is shown in fig. 4.

3.2 Material Selection

As per the rulebook, the roll cage material must have at least 0.18% carbon content. After an exhaustive market survey, commercially available and usable materials are shortlisted. A comparative study of these shortlisted materials is done on the basis of strength, availability and cost. The shortlisted materials are as follows.

- AISI 1018
- AISI 4130
- AISI 1026

Material property	**AISI 1018**	**AISI 4130**
Yield strength	370 MPa	460 MPa
Ultimate strength	440 MPa	670 MPa
Hardness	126	217
Density	7.87 g/cm^3	7.85 g/cm^3
Cost	Cheap	Expensive
Welding	SMAW	TIG

Table 1. Comparison of material properties

From the table:

- AISI 4130 provides better strength than AISI 1018
- Density of AISI 4130 is less than that of AISI 1018

Hence, it gives better strength with less weight and we can also use SMAW on AISI 4130.

4. Static analysis

For impact force calculations:

- Impact time for collision with rigid body: 0.2 sec
- Impact time for collision with rigid body: 0.4 sec

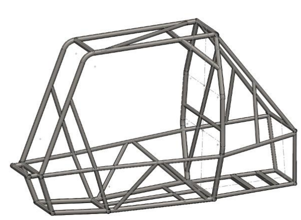

Fig. 1a. Designing model in SolidWorks

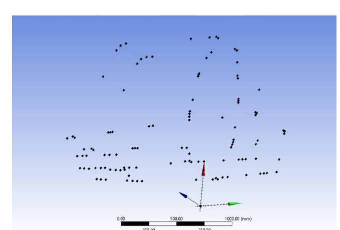

Fig. 1b. Importing points in Design Modeler

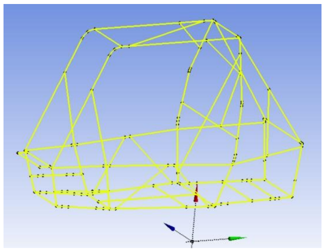

Fig. 1c. Making lines from points

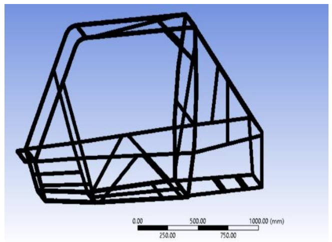

Fig. 1d. Meshing the model

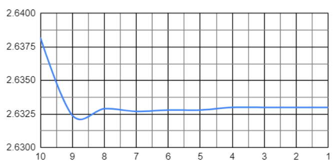

Fig. 2. Convergence study for meshing roll cage

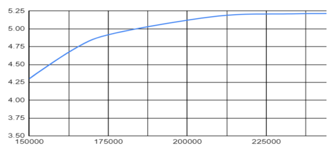

Fig 3. Convergence study for meshing of knuckle

4.1 Front impact

4.1.1 Force Calculations

Let m be the mass of the vehicle with the driver, v_1 be velocity before impact, v_2 be the velocity after impact, and t is the impact time.

m=265 kg, v_1=40 km/h =11.1 m/s, v2=0m/s, t=0.2s

In this case vehicle at 11.1 m/s is considered to hit a wall, and comes to rest in 0.13s

∴ **Impact Force, F** = m*a = m*(v2-v1)/t

= 14707.5 N

= **6.81 G**

4.1.2 Boundary Conditions and Constraints

Rear suspension points are kept fixed, front suspension points are kept free in the longitudinal and lateral direction and fixed in the z-direction. Force has applied on the foremost member. Fig. 5 shows all the solver settings.

4.1.3 Results

Fig. 6 and 7 shows contours of total deformation and maximum combined stress. Based on the maximum value of combined stress a **FOS of 1.58** is obtained.

(FOS = maximum stress/Yield strength; yield strength of AISI 4130 is 460 MPa).

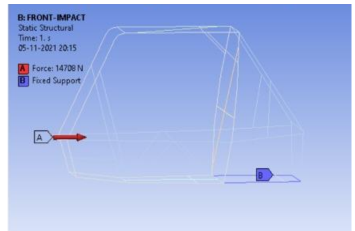

Fig. 5. Boundary Conditions (Front Impact)

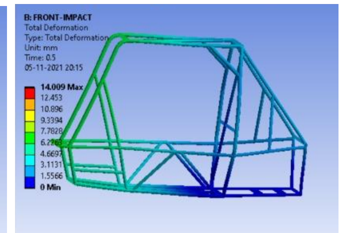

Fig. 6. Total Deformation (Front Impact)

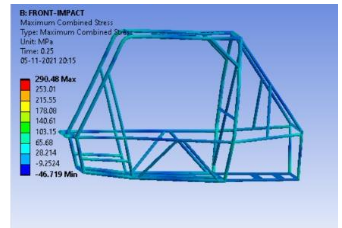

Fig. 7. Combined Stress (Front Impact)

4.2 Rear impact

4.2.1 Force Calculations

Here impact time is 0.4s, as a vehicle at 11.1 m/s will hit the rollcage from behind.

∴ **Impact Force**, F = m*a = m*(v2-v1)/t

= 9805 N

= **4.54 G**

4.2.2 Boundary Conditions and Constraints

Front suspension points are kept fixed, rear suspension points are kept free in the longitudinal and lateral direction and fixed in the z-direction. Force is applied to the rearmost members. Fig. 8 shows all the solver settings.

4.2.3 Results

Fig. 9 and 10 shows contours of total deformation and maximum combined stress. Based on the maximum value of combined stress a **FOS of 1.84** is obtained.

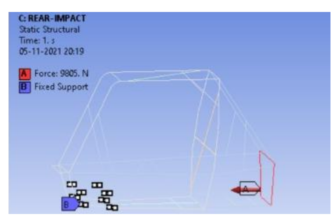

Fig. 8. Boundary Conditions (Rear Impact)

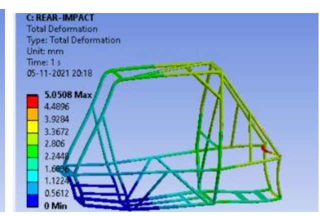

Fig. 9. Total Deformation ((Rear Impact)

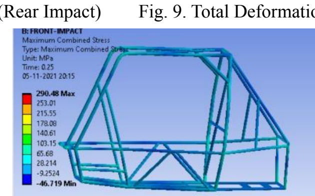

Fig. 10. Combined Stress ((Rear Impact)

4.3 Side impact

4.3.1 Force Calculations

Here impact time is 0.4s, as a vehicle at 12.5m/s will hit the rollcage from behind.

∴ Impact Force, F = m*a = m*(v2-v1)/t = 6875N = 3.19G

4.3.2 Boundary Conditions and Constraints

Opposite side suspension points are kept fixed, force side suspension points are kept free in the longitudinal and lateral direction and fixed in the z-direction. Force is applied on SIM and RRH. Fig. 11 shows all the solver settings**.**

4.3.3 Results

Fig. 12 and 13 shows contours of total deformation and maximum combined stress. Based on the maximum value of combined stress a **FOS of 0.77** is obtained.

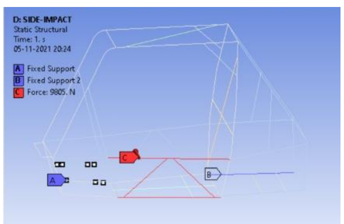

Fig. 11. Boundary Conditions (Side Impact)

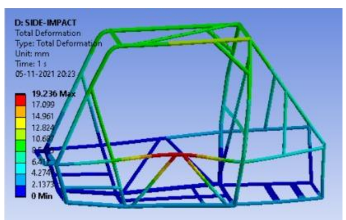

Fig. 12. Total Deformation (Side Impact)

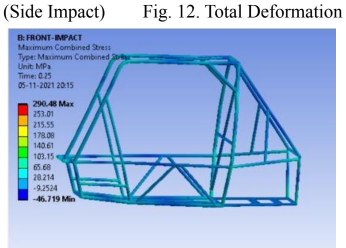

Fig. 13. Combined Stress (Side Impact)

4.4 Roll over

4.4.1 Force Calculations

In this case, the vehicle is considered to fall from a height, h=3.048m=10 feet at 45 degrees angle with the ground. Assuming, conservation of mechanical energy; m*g*h=0.5*m*v2 ⇔ v=7.73m/s; t=0.2s

∴ Impact force, F = m*v/t = 8503N = 6G

4.4.2 Boundary Conditions and Constraints

All the suspension points are kept fixed, and force is applied on RHO at a 45-degree angle. Fig. 14 shows all the solver settings.

4.4.3 Results

Fig. 15 and 16 shows contours of total deformation and maximum combined stress. Based on the maximum value of combined stress a **FOS of 1.60** is obtained.

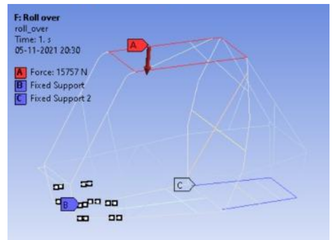

Fig. 14. Boundary Conditions (Roll over)

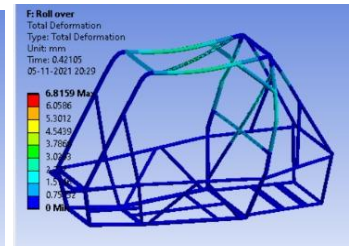

Fig. 15. Total Deformation (Roll over)

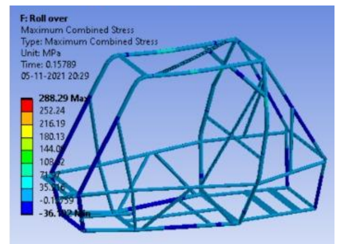

Fig. 16. Combined Stress (Roll over)

4.5 Brake pedal

Boundary conditions:

- Pivot point and pushrod joint are fixed
- The force of magnitude 2000N(mentioned in the SAE BAJA rulebook) is applied on foot to the pedal in a direction perpendicular to the surface of the foot of the pedal.

Mass	729.3 gr
material	Structural steel
thickness	26 mm
Max deformation	0.001 m
Max equivalent stress	314876 Pa

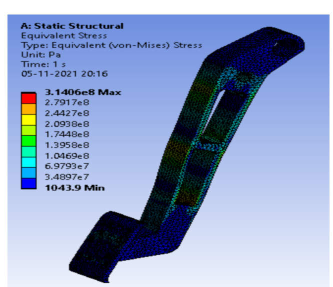

Fig. 17. Equivalent stress (Brake pedal)

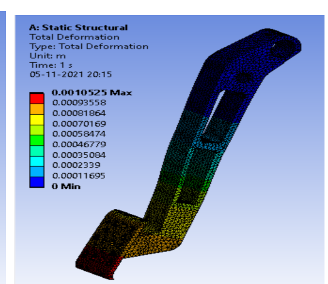

Fig. 18. Total Deformation (Brake pedal)

4.6 Gearbox

Material	Al7075
Mesh size(Element size)	0.003 m
Equivalent stress	$5.9712*10^{7}$ Pa
Total deformation	$7.4289 * 10^{-5}$ m
Fatigue tool	FOS: 2.3027

Boundary Condition:

- The clamping points of casing are fixed
- Force of 182 N was a applied on caliper mounting
- For gearbox, output gear was fixed

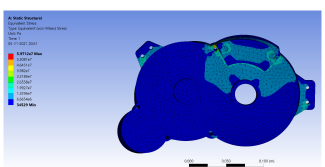

Fig. 19. Equivalent Stress

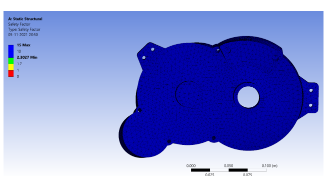

Fig. 20. FOS

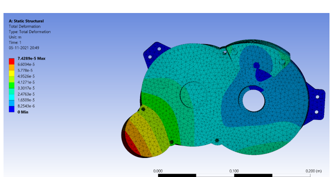

Fig. 21. Total Deformation

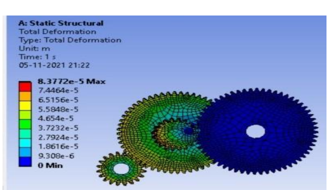

Fig. 22. Total Deformation

Fig. 23. Equivalent Stress

4.7 Knuckle

Material: Aluminum 7075 T6

Because it has suitable mechanical properties such as lightweight, high strength, and fair machinability.

Two cases of Static structural analysis:

i) When dropped from 1m height, in this we analysed Maximum Equivalent Stress(Von-mises) and total Deformation.

Input (Front)
Force @ UBJ = 0, 0, -4000N
Force @ LBJ = 0, 0, -8000N

Result (Front)
Total Deformation: 0.54mm
Equivalent Stress (Von-mises): 100 MPa

Input(Rear)
Force @ Lower wishbone pivots point = 0, 0, -8000N

Result(Rear)
Total Deformation: 0.27mm
Equivalent Stress (Von-mises): 84.5 MPa

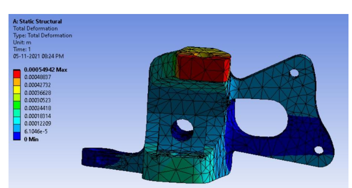

Fig. 24. Maximum total deformation (Front)

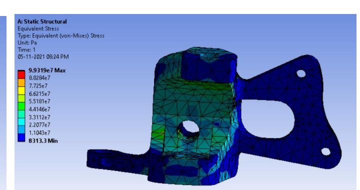

Fig. 25. Maximum Equivalent Stress (Front)

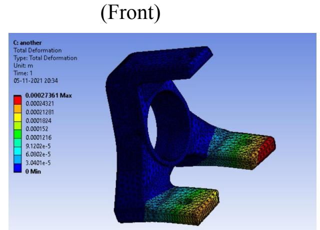

Fig. 26. Maximum total deformation (Rear)

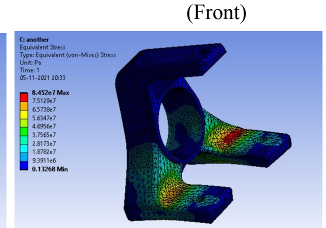

Fig. 27. Maximum Equivalent Stress (Rear)

ii) Static Structural analysis for the lateral and longitudinal forces generated during dynamic analysis using lotus were applied at UBJ, LBJ, steering arm along with braking force, last two are for front only. And for fatigue analysis we are using Gerber theory.

Input (Front)
Force @ UBJ = 415.95, 4511.50, 5320.11
Force @ LBJ = -363.13, -3495.11, -546.40
Breaking torque @ caliper mount = 154.3 Nm
Steering force @ Steering arm = 160N

Result (Front)
Total Deformation: 0.22mm
Equivalent Stress (Von-mises): 64MPa
Safety Factor: 2.14
Equivalent Alternating Stress : 64 Mpa

Input(Rear)
Force @ Lower wishbone outer front pivot = -313.47, 1118.38, 963.85
Force @ Lower wishbone outer rear pivot = -320.42, 857.26, 2708.72
Force @ Upper link outer ball Joint = 633.89, -1975.64, -265.13

Result(Rear)
Total Deformation: 0.16mm
Equivalent Stress (Von-mises): 54.6MPa
Safety Factor: 2.51
Equivalent Alternating Stress: 54.6 Mpa

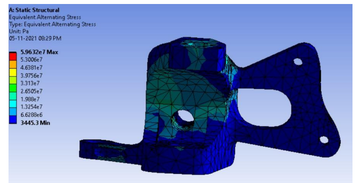

Fig. 38. Maximum alternating Equivalent Stress(Front)

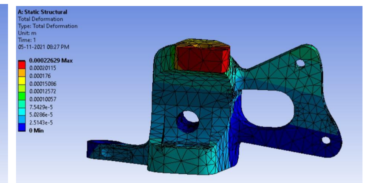

Fig. 29. Total deformation (Front)

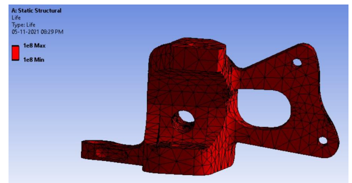

Fig. 30. Life of knuckle (Front)

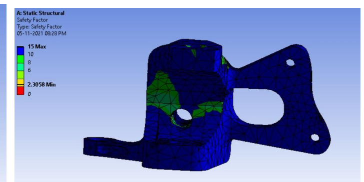

Fig. 31. FOS (Front)

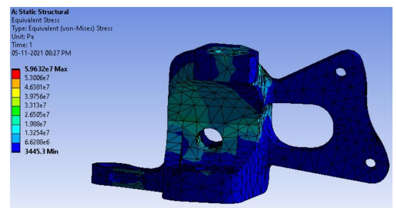

Fig. 30. Maximum equivalent stress (Front)

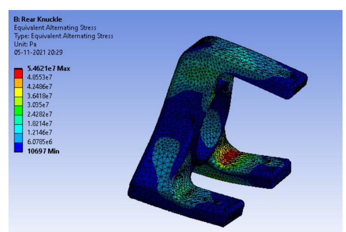

Fig. 31. Maximum alternating Equivalent Stress(Rear)

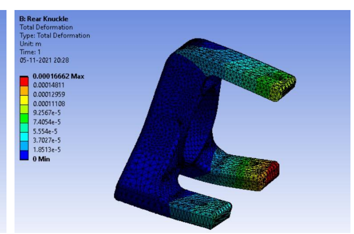

Fig. 32. Total deformation (Rear)

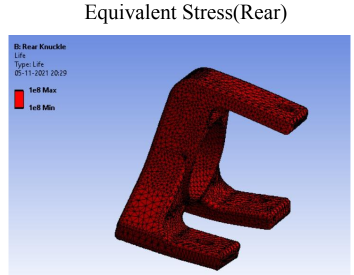

Fig. 33. Life of knuckle (Rear)

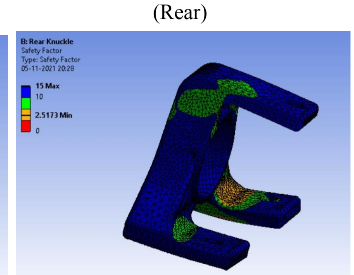

Fig. 34. FOS (Rear)

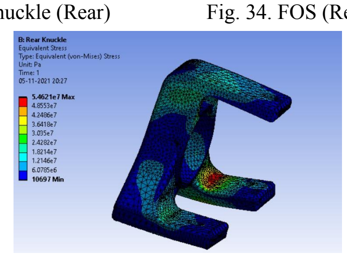

Fig. 35. Maximum equivalent stress (Rear)

4.8 Hub

With the consideration of combined loads during cornering, lateral weight transfer, and bump, we carried fatigue analysis. We also included acceleration and break loads. Material used for hub is same as knuckle i.e. Aluminum 7075 T6 for the above mentioned properties.

Input
Lateral Force at Spindle: 3530N
Bump force at spindle : 8000N
Braking Torque at Brake disc Mount : 154 Nm
Rolling wheel Torque : 27Nm
Remote Displacement : 4^0 about lateral axis

Results
Total Deformation: 0.32mm
Equivalent Stress (Von-mises): 88MPa
Safety Factor: 8.98

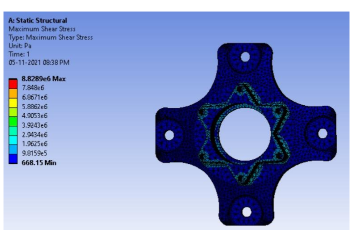

Fig. 36. Maximum Equivalent Stress

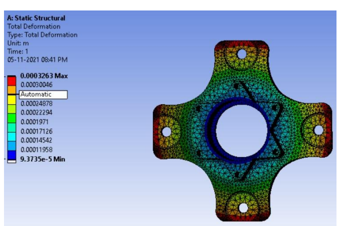

Fig. 37. Total deformation

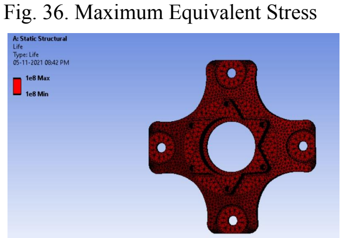

Fig. 38. Life of knuckle

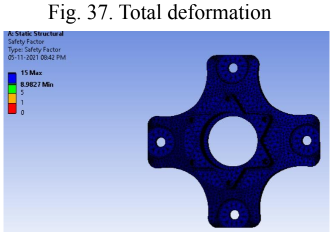

Fig. 39. FOS

Results

So we can see here that FOS in every case is greater than 2, Maximum equivalent stress is less than UTS of the materia i.e. (110MPa) and Life of every element is 10^8 cycles.

4.9 Suspension arms

Bending and axial forces are being applied on suspension arms during bump and cornering respectively. While axial force is much lesser than Bending we are neglecting axial loads. Material we are using for them is the same as the roll cage one. Mounting points of wishbone are chosen as fixed supports.

Input
Force on ball joints: 2000N
Force on shocker mountings: 1500N

Output
Max Equivalent stress: $3.158 * 10^7$N
Total deformation: 0.0027m
Max Life cycles: 10^8
Safety Factor: 1.604

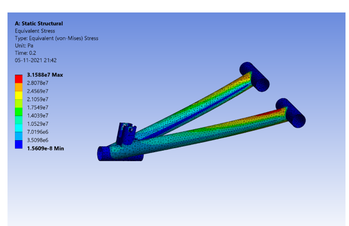

Fig. 40. Maximum Equivalent Stress

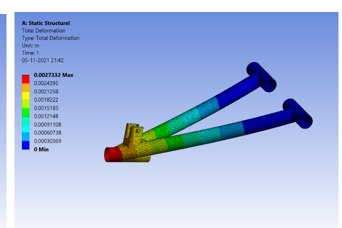

Fig. 41. Total deformation

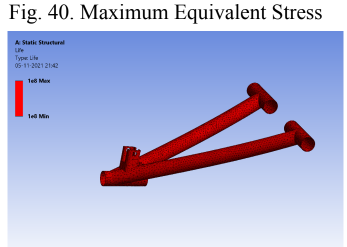

Fig. 42. Life of knuckle

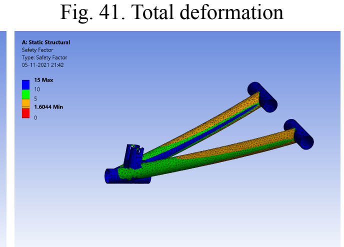

Fig. 43. FOS

5. <u>Torsional and bending analysis</u>

Cornering of the vehicle results in the weight transfer in the lateral direction at the outward side. Due to this, the outward shockers compress and inward shockers elongate. This creates a couple. The resistance of the vehicle to this coupling force is called the **torsional rigidity of the vehicle**. The higher the torsional rigidity, the more comfortable the ride will be. For calculating torsional rigidity, we have to calculate the weight transfer.

Weight transfer, W=m*a*h/t;
m = mass of the vehicle,
a = lateral acceleration = v^2 /r;
h = height of COG of the vehicle,
t = track width,
v = turning velocity, and
r = turning radius.

Putting values as m=265kg, v =13.89m/s, r =2.3m, h=0.5116m, t=1.544m,

we get,

W=3887 N **=1.5G**.

For calculating the torsional rigidity, the maximum deformation in the z-direction is taken as Δz which is 7.18mm as seen from fig. 18.

Distance between the coupling force, y=0.3302m, and F=3887N.

Now,

tanθ≃θ=2Δz/y radian ;

Torque, T=F*y=kθ ⇔ k=Fy^2 /2Δz.

Putting values we get, **k=1772Nm/deg**.

5.1 Boundary Conditions and Constraints

We have to apply this couple at the shocker joints, which is shown in fig. 35. Rear shocker joints are kept fixed.

5.2 Results

Fig. 36 and 37 show the contours of total deformation in Z direction and maximum combined stress respectively. **FOS of 1.79** is obtained.

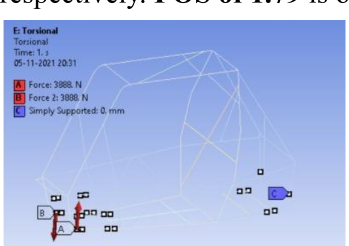

Fig. 44. Boundary Conditions (Roll over)

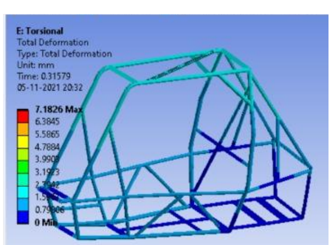

Fig. 45. Total Deformation (Roll over)

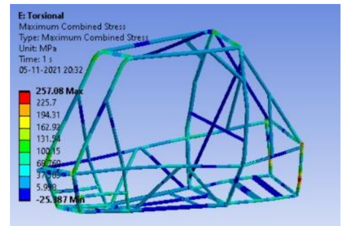

Fig. 46. Combined Stress (Roll over)

6. <u>Thermal Analysis for Brake Disc</u>

Material of Disc	Grey cast iron
Specific heat capacity	490 J/Kg*K
Thermal conductivity	53.5 W/m*K

Boundary conditions:

- For structural and thermal analysis of the rotor, six connecting bolt holes are fixed.
- Net torque of 253.5 Nm is applied on the brake disc
- (using the coefficient of friction of 0.7).
- Torque is applied as a result of frictional force between the brake rotor and caliper pads

Property	values
Linear velocity of vehicle	50 kmph
Mass of vehicle	265
Mass of brake disc rear	469.3 gr
Mass of brake disc front	354.9 gr
Mesh size used	3mm
Film coefficient	33.12 W/ m*2 C
Change in kinetic energy	25559 J
Equivalent von 0 mises stress (max)	6.628 * 10^7 pa
Deformation of (max)	0.0025 m

Result:

Temperature : 137℃ to 264 ℃

Heat flux range: 375- 38553 W/m^2

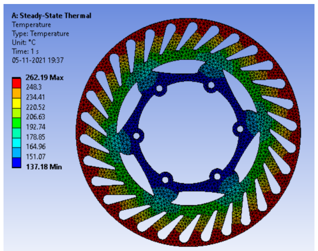

Fig. 47.Temperature static-state thermal

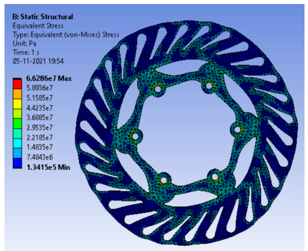

Fig. 48. Equivalent stress-static structural

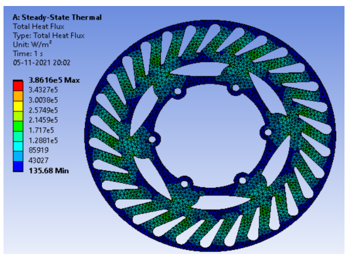

Fig. 49.Total heat flux- steady state thermal

7. <u>Multibody Dynamic Analysis</u>

Dynamic analysis has been performed to find out the variations of different parameters which can have a significant effect on the performance of the vehicle. This analysis is performed with the help of the lotus suspension analysis tool. The analysis considers mainly 3 scenarios namely bump travel, roll and steer. Firstly half car analysis is separately performed for the front and back and then full car analysis is performed for finding the forces acting on the wheels.

7.1 Bump analysis:

Analysis has been performed to find out the variation of parameters like camber, toe, caster, Ackermann percentage, and roll center height with a wheel travel of 8in bump and 4in droop in the front and 5in bump 3in droop travel for the rear. Variations are represented in graphs below for both front and rear wheels. After checking the range of variations we modified our design and made sure that the range of variations is acceptable. We also found out the variation of forces acting on the wheels(front and rear) during wheel travel. the forces which are mainly considered were traction force, lateral force, and longitudinal force according to the SAE convention system.

7.2 Roll analysis:

In this analysis, we measure the variation of different parameters as mentioned above w.r.t variation of the roll angle of the car and then modify the geometry until the range of these parameters is in acceptable regions. Here roll angle is varied from -7 deg to +7 deg. Variations of parameters are shown below for both front and rear wheels.

7.3 Steer:

In this analysis, we measure the variation of parameters w.r.t rack displacement. From our calculations, rack travel came out to be 100mm for our model. The variations of parameters for the front wheel are shown below. We verified whether the steer angles(toe change) with our calculations and modified them until they matched.

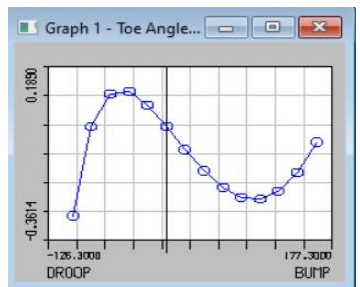

Fig. 50. Toe angle

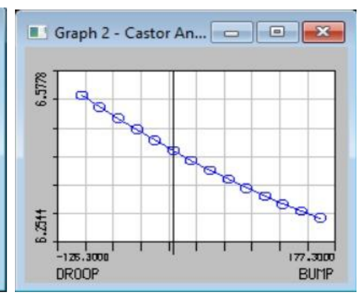

Fig. 51. Castor angle

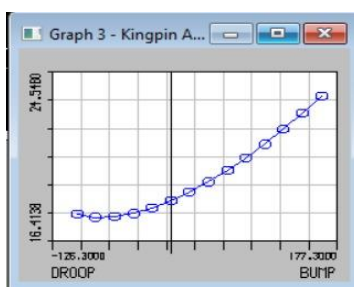

Fig. 52. Kingpin angle

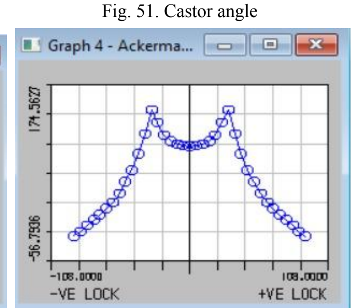

Fig. 53. Ackerman (steering)

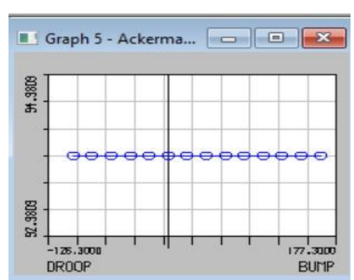

Fig. 54. Ackerman (Bump Travel)

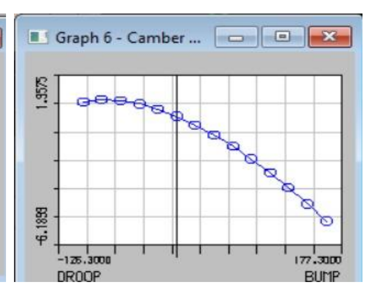

Fig. 55. Camber

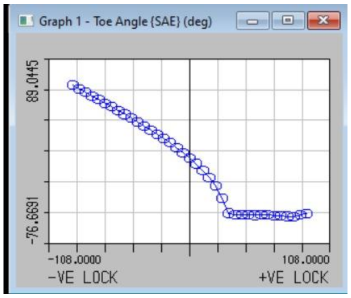

Fig. 56. Toe angle vs Rack travel

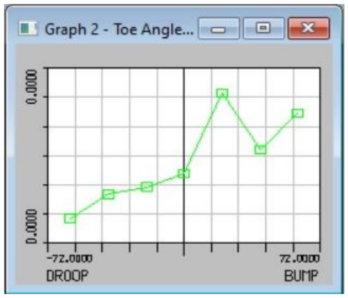

Fig. 57. Toe (rear)

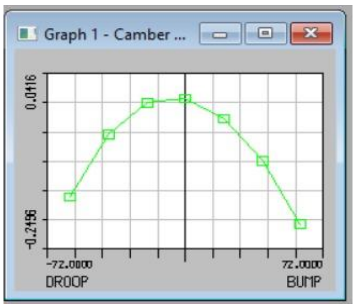

Fig. 46. Camber (Rear)

8. Conclusions

- Fine mesh is used as it completely defines the solution of the problems faced in the analysis.
- Regions of high-stress gradients i.e. holes,edges are meshed using hexahedral mesh having aspect ratio as high as 16 and low-stress regions are meshed using tetrahedral meshing.
- Parameters like skewness, aspect ratio, jacobian & orthogonality are considered while meshing all the components.
- Parameters for knuckle, hub and roll cage are shown in Table 1 and the bracket value shows max. Value beyond which meshing collapses.

	Roll cage	Front knuckle	Rear knuckle	Hub
Skewness	0	0.40	0.41	0.41
Aspect ratio	0	2.39	2.31	2.33
Jacobian	0	0.90	0.95	0.93
Orthogonality	0	0.59	0.58	0.58
Nodes	32405	11996	41611	51944
Elements	16221	6870	23176	29394

9. References:

1) Race Car Vehicle Dynamics by William F. Milliken and Douglas L. Milliken
2) Mr.U.Yashvanth ,Mr. Ganapa Raghu Vamsi Reddy,Mr. Mohammad Raffi, Structural Analysis of a ATV Wheel Hub , Vol. 8 Issue 03, March-2019
3) Chetan Chaudhari, Aditya Pasalkar, Akshay Gavade, Ashish Jagtap, Design of Two-Stage Single Speed Gearbox for Transmission System of SAE BAJA Vehicle ,Volume: 05 Issue: 10 | Oct 2018
4) Raj kumar yadav, Adani group of Institutions, BRAKE DISC ANALYSIS WITH THE HELP OF ANSYS SOFTWARE,November 2015.
5) A. S. Kulkarni, K. C. Hegade, O. V. Karale & P. S. Joshi, Coupled Thermal and Structural Analysis of Brake Disc Rotor Manufactured from Aluminum Metal Matrix Composite (AMMC) reinforced with Silicon Carbide
6) Adams, H., “Chassis engineering,”.
7) Ebsch, E.Kudla, JO'Brien, C. and Quick, B, “ME 450 Gear Reduction, Baja Gear Reduction,”University of Michigan Baja team, 2012-13.
8) Budynas, R. G. and Nisbett, J.K., “Shigley’s Mechanical Engineering Design, Tenth Edition,” ISBN 978-0-07-339820-4
9) Hui, L. Z., “Design, Analysis and experimental verification of tubular chassis for FSAE application”