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Study and Analysis of Formula Style Vehicle

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***Abstract*— Formula Student (FS) is the proven educational motorsport competition in the world. Universities around the world are included in this challenge to design and build a single-seater racing car to compete in static and dynamic disciplines that demonstrate their knowledge and test the performance of the car. Specifically this work aims at the proper design of all respective geometries of the vehicle with proper analysis as supporting documents for the study. This work aims at producing design of components such that those can be manufactured in a feasible manner. The objective is to design a formula student vehicle with careful study and analysis of its various assemblies and related sub-assemblies, that will finally produce a vehicle with most possible efficiency by following proper engineering practices. This design will be such that it is a light weight design with a very efficient suspension & steering geometry. The design and analysis part will be done using software namely SolidWorks, Autodesk Fusion 360, Lotus, Ansys Workbench. This study will be such that, with the information in this report a student formula team can easily manufacture a vehicle.**

Keywords— Formula 4, Student Formula, Design, Analysis

I. INTRODUCTION

The objective of this work is to design a formula student vehicle with careful study and analysis of its various assemblies and related sub-assemblies, that will finally produce a vehicle with most possible efficiency by following proper engineering practices. This design will be such that it will satisfy below mentioned points.

- A light weight design.
- Very efficient suspension & steering geometry.
- Aesthetically pleasing design.

This study will be such that, with the information in this report a student formula team can easily manufacture a vehicle. To aid us design a formula student car, we have decided to follow the rules and regulations as in the SAE Supra 2022 Rulebook.

II. SELECTION OF INITIAL DESIGN CONSTRAINTS

We finalized the key specifications of the vehicle such as wheelbase and track-width considering the track constraints and SAE rules.

- Wheelbase : 62 Inches
- Front Track Width : 46 inches
- Rear Track Width : 45 inches
- Wheels and tyres : 10 inch alloy wheel with 16*7.5 inch intermediate tyres.

The rear track width of the vehicle is fixed smaller than the front track width in order to achieve better cornering stability for the driver. In the SAE Supra competition, a dynamic test called auto cross is conducted where the hairpin turns will have a minimum of 9 m (29.5 feet) outside diameter (of the turn).

From the available track tyres from the market we chose to use smaller tyres to obtain good acceleration values, as top speed is not a major requirement in SUPRA when compared with acceleration needs.

III. SUSPENSION GEOMETRY

The center of gravity (CG) position of the vehicle was found with the help of wheelbase, track width and with the desired weight of 290 kgs.With the help of CG position the front and rear roll center heights are determined which are the major components for developing the suspension geometry. With the help of CG position and Roll center heights the front and rear suspension geometry are designed considering the weight ratio of the vehicle as follows.

A. *Front Suspension Geometry*

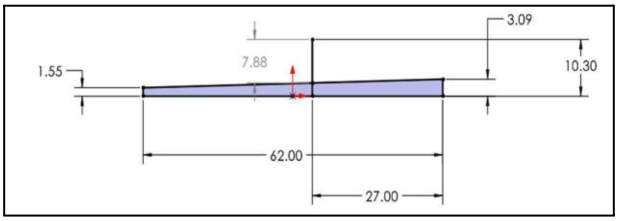

Figure 1 Roll center height of front and rear

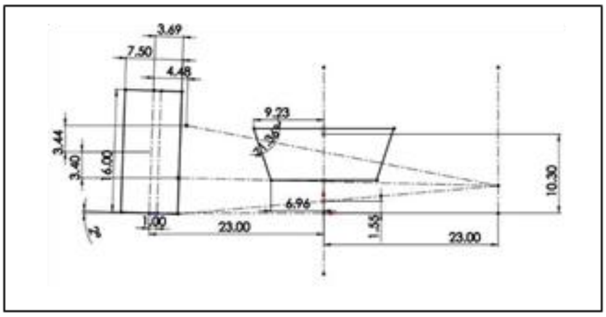

Figure 2 2D Geometry of front suspension setup

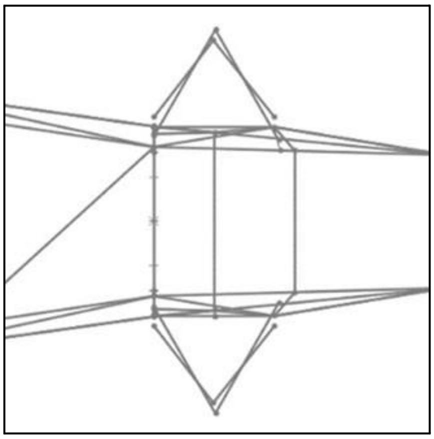

Figure 3 3D Geometry top view of front suspension setup

B. *Rear Suspension Geometry*

- Camber : -2degree
- Caster : +2 degrees
- Toe : 0 degrees
- Type : Push rod suspension system
- Dampers : DNM RCP 2s

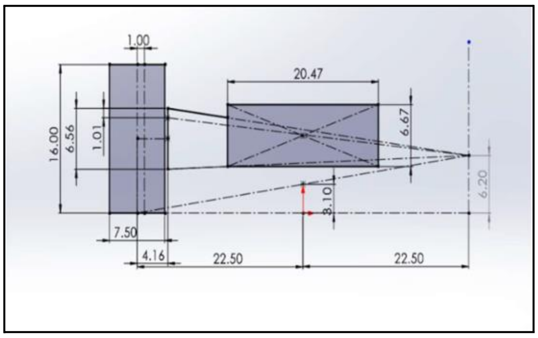

Figure 4 2D Geometry of rear suspension setup

All these values' positions in the cartesian plane were noted down and the chassis building process and the design of elements of the suspension system was done. All these values were confirmed by simulation in Lotus software. The following are the results of the software.

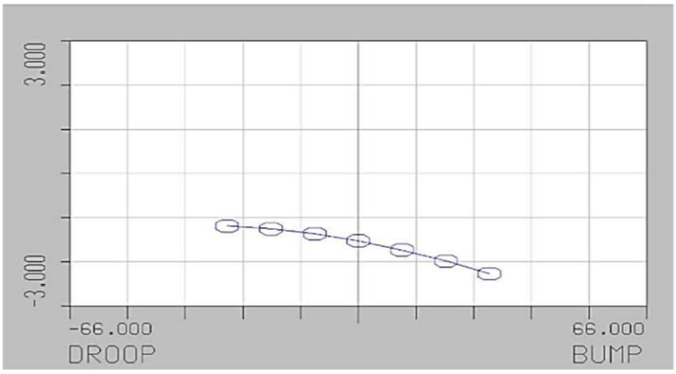

Figure 5 Camber Graph

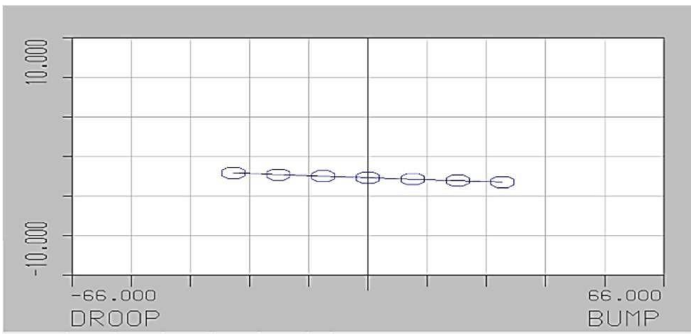

Figure 6 Castor Graph

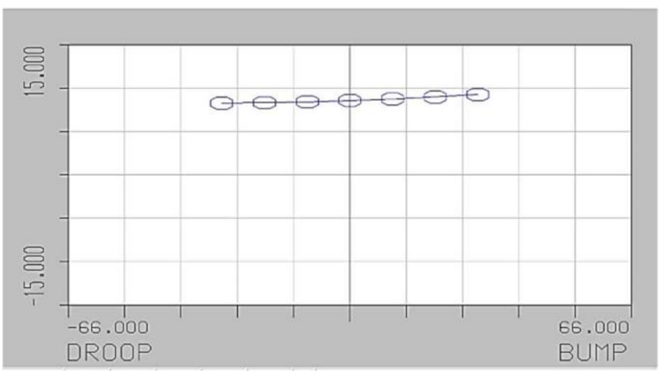

Figure 7 KPI Graph

IV. CHASSIS

We finalized to use AISI 1018 for our roll cage member considering the following conditions.

- AISI 1018 has good weldability
- Its availability in the market compared to other alternatives.
- There is no heat treatment requirement
- The cost of AISI 1018 is affordable than the counter materials

Criteria	Minimum Rulebook Requirement	AISI 1018 (as per AZO Materials)	AISI 4130 (as per AZO Materials)
Young's Modulus (E)	200 GPa	205 GPa	205 GPa
Yield Strength (Sy)	305 MPa	370 MPa	435 MPa
Ultimate Strength (Su)	365 MPa	440 MPa	670 MPa

Table 1 Material Properties Comparison

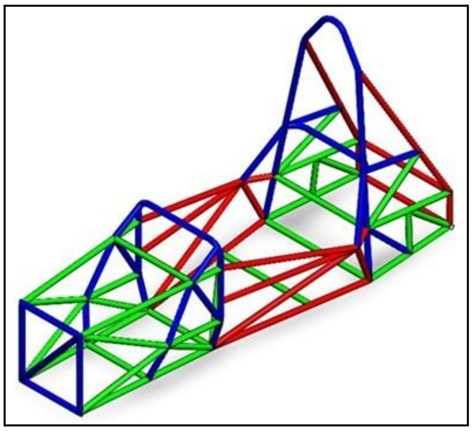

Figure 8 Chassis with members with different cross sections highlighted as per table 2 below

Colour	Dimensions
🟢	1" x 0.049"
🔴	1" x 0.065"
🔵	1" x 0.095"

Table 2 Cross section of pipes used notated with colour

V. STEERING SYSTEM

Anti-Ackerman geometry has been used which aids in reducing the slip angle of the inner wheel & providing more grip for vertical loads. Our Anti-Ackerman geometry is as follows: & the following values were obtained from this geometry.

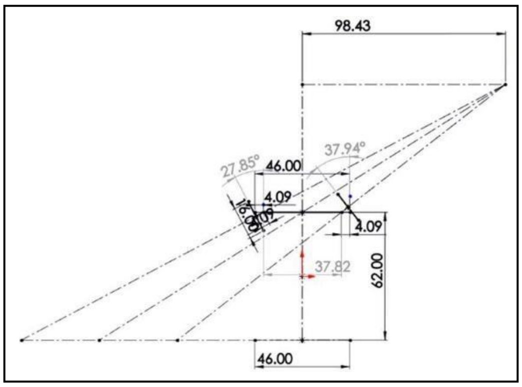

Figure 9 Anti-Ackerman Geometry

- Outer wheel turning angle for 2.5m turning radius Φ=27.85.
- Outer wheel turning angle for 2.5m turning radius θ=37.94.
- From the above obtained angles the steering ratio is 2.735 with wheel total angle as assumption of 240°.

A. *Length of Rack*

One of the objectives of steering design is to reduce the roll and bump steer. In consideration of bump steer the eye-to-eye length of rack is found. From the softwares (Solidworks and Lotus) the rack travel is found as 110 mm.

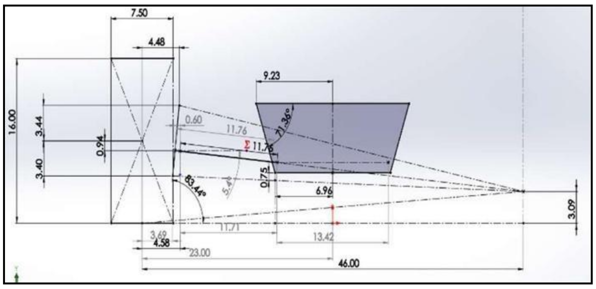

Figure 10 Rack Length

Considering the obtained rack travel value, the following values of rack and pinion were obtained.

S. No	Specifications	Formula	Values
1	Pinion Diameter	R=π*dpi*η.	27 mm
2	No of teeth on pinion	M= dpi / tpi	18
3	No of teeth on rack	R = 3.14*dr	12

Table 3 Rack and Pinion Specifications

Steering wheel torque is found to be 7.7481Nm

If the steering wheel diameter is considered as 10 inches, then the steering effort is calculated as 67N.

At static condition the force required to rotate the steering is 67N. which is the maximum force required to rotate the steering wheel.

Ackerman value = tan-1{wheel base / [wheel base/tan Φ] - front track width} = 37.37

Ackerman % = (angle turned by inner wheel / Ackerman) 100 = 69%

B. *Steering Assembly Mount*

A simple and cost-efficient steering setup housing has been designed and made for smooth transmission of rotational motion of steering-to-steering column.

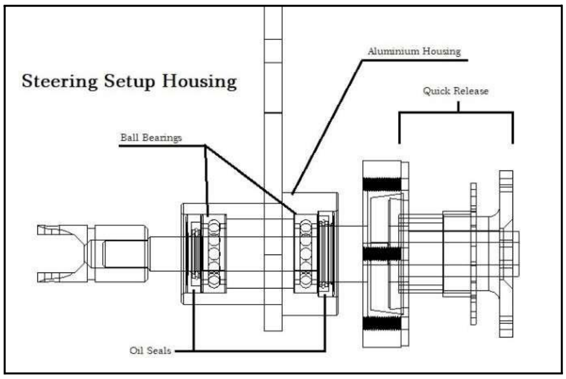

Figure 11 Wireframe Representation of Steering Mount Assembly

VI. POWERTRAIN

According to rulebook constraints and to achieve high performance, KTM Duke 390 engine was chosen which provides better performance within the price category of the competitive vehicles.

- The aerodynamic drag force (Fa)= 162.69 N
- The Rolling resistance force (Fr) = 56.898N
- The Gradient resistance force (Fg) = 1202.30N

From these values the Total Tractive Force is found to be (Fa+Fr+Fg) = 1421.89N

With the found total tractive force
Torque at wheels = TTF x resistance factor x wheel radius
= 1421.89 x 1.15 x 0.2032 = 332.26Nm

The primary drive ratio = 30:80 = 2.66

With the found primary drive ratio the individual ratio, overall gear ratio, torque and acceleration for individual gears are calculated as follows

Gear	Torque (Nm)	RPM	Acceleration (m/s2)
1	824.6	111.41	13.99
2	575.4	85.94	9.76
3	440.3	146.97	7.47
4	345.27	226.23	6.01
5	296.45	324.08	5.03
6	260.4	420.16	4.41

Table 4 Individual Gear Outputs

Top Speed (S):

S = [(2. p) / (Cd.A.ϱ)] ^1/3

Max. power from engine (p) = 43hp (32065.1watt)

S = 136.656 Km/h

Torque at drive sprocket (Mf):

Mf = max torque x primary reduction ratio x first gear ratio = 261.7972Nm

Torque at driven sprocket (Mr):

Mr = Mf * final drive ratio = 871.78Nm

A. Drive Sprocket Analysis

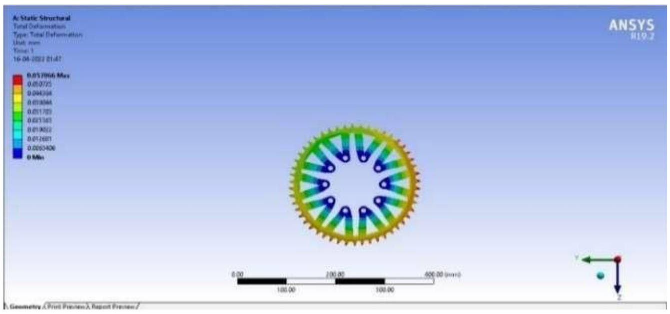

Figure 12 Driven Sprocket Analysis

B. Tractive Effort

Force = (τ x ε x η) / wheel radius

ε - gear reduction ratio

η - Transmission efficiency= 96% (Our assumption)

F1 = 3913.07N

F = F1 - Fa - Fr

F= 3693.482N

C. Differential Housing

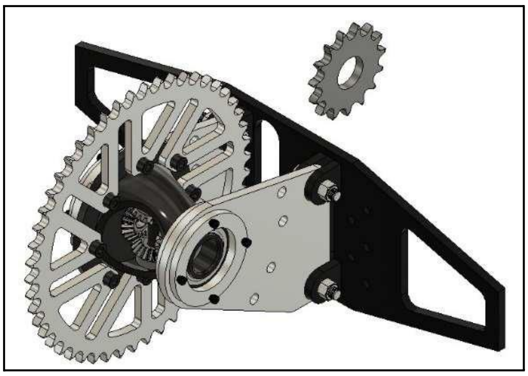

Figure 13 Design of Differential Housing

Above picture depicts the custom design of the light weight and efficient differential housing that integrates the differential to the engine and the chassis.

VII. RESULTS & DISCUSSION

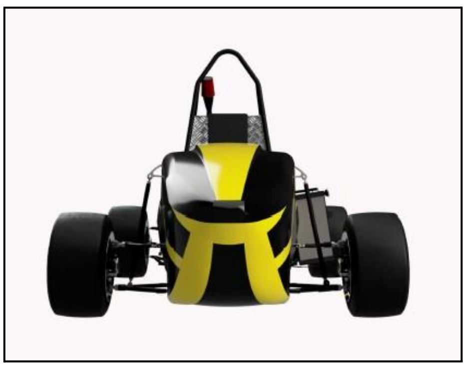

Figure 13 Vehicle Front View

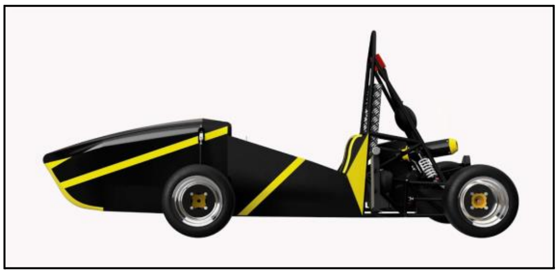

Figure 14 Vehicle Side View

Figure 15 Vehicle Isometric View

VIII. CONCLUSION

The design of this car has been done with a target to extract the best performance from the various components with consideration of driver safety to the fullest.

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