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Anti-Ackermann Steering System of Formula Student car

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Abstract - *In present study, as a basic step for modeling and building a steering system is shown. The aim is to design a steering system for a formula student vehicle with the desired steering ratio, zero play. The design process is to be done in DS Solidworks and finite element analysis using Ansys. There are various parts in the FSAE car steering with need to be designed considering the various impact forces and stresses, like the rack and pinion and the steering shafts, which are majorly caused due to the longitudinal and lateral accelerations which act on the driver as well as the car and in turn the driver must apply a force much greater than that to control the vehicle. In formula student vehicles, weight, simplicity, and accuracy of systems have prime importance. As a conclusion the use of high strength engineering materials in the steering system of a formula student vehicle will make the system lighter and more efficient than traditional one.*

***Key Words*: Rack, Pinion, Steering, Anti-Ackerman, Steering Arm**

1.INTRODUCTION

The function of the steering system is to provide directional control to the vehicle. For this, a rack and pinion gear are used, which converts the rotational motion coming from the steering wheel to linear motion going towards the wheels, hence turning them. There are many types of steering mechanisms which include worm gear type, ball type etc. but rack and pinion gear is the one which has relatively a smaller number of parts and has less error. Thus, this paper focuses on the design of a steering system consisting of rack and pinion for a Formula Student vehicle to run in Formula Bharat event, which is the Indian version of the Formula Student event organized globally by various countries and hosted in India by Curiosum Tech Private Limited, and the SAE-SUPRA event, by the Society of Automotive Engineers (SAE), United States of America, hosted in India by Maruti Suzuki India Private Limited. Both the events require undergraduate students to design, fabricate, validate, and run a race car prototype and then run it through a series of static and dynamic events.

There are many types of steering geometries which are used in various types vehicles like the Ackermann geometry, Anti-Ackermann geometry and the parallel steering type geometry. Each of these geometries offer a different type of advantage and hence based on the operating conditions, the geometry must be finalized.

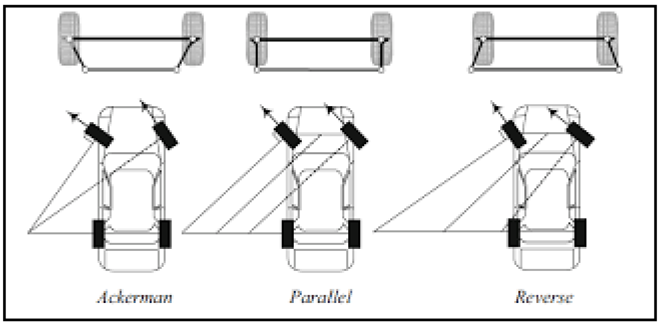

Fig.1. Types of steering geometries. From left, Ackermann geometry, Parallel geometry and Reverse aka. Anti-Ackermann geometry

2.TARGET

1. To design a steering system which is fully compliant with Formula Bharat Rules Booklet 2021(FBRB-2021).[1]
2. To achieve a turning radius of 3200mm.
3. To design the steering system in such a way that the tires are optimized to their full capacity

3.DESIGN WORKFLOW:

The design procedure of the steering system starts from the tires. The force required to turn the tires gives out the force which will be transmitted through the wheel assembly, the knuckle, the steering arms, the tie rod and to the rack.

4.BASICS ABOUT THE STEERING GEOMETRY:

The steering geometry currently used in the car is the 6-point anti-Ackerman Geometry. In this type of geometry, the outer wheel turns at more angle than inner wheel and is more efficient for allowing the outer wheel to steer a tighter radius. In using Anti-Ackerman steering we hope to be able to influence the slip angle on the outer tire to our advantage. There will be a range of slip angles where the outer tire will be producing near maximum grip. So, we have a degree of flexibility in how much Ackerman angle we use.

The geometry shown in Fig. 2 is the first geometry that is drawn when we decide on starting with the geometry. The top horizontal line here represents the front track width (1240mm here) and the bottom line represents the rear track. The perpendicular distance between the two tracks is the wheelbase (1600mm here). To get the turning angles, two lines are drawn from the edges of the track width to intersect the rear track. The turning radius is set (3200mm here) on the outer wheel and the turning angles for the inner and outer wheels are determined i.e., 45.755º and 30.526º, respectively.

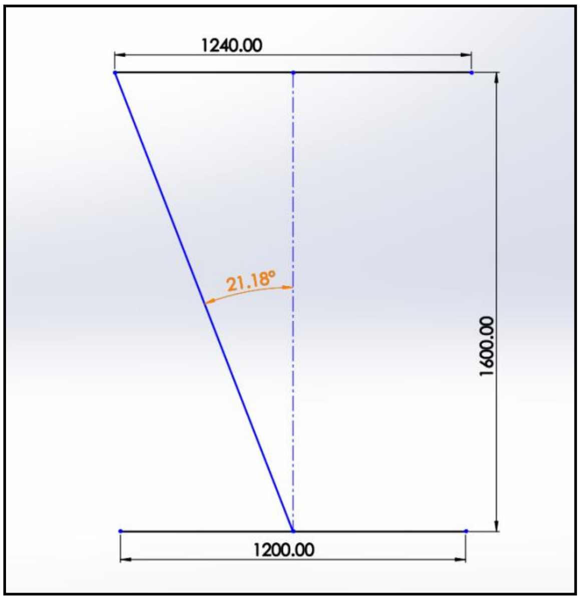

Fig.2. Initial constrain geometry

The next task is to determine the Ackerman angle. This has to now include the optimization of Ackermann angle which we found by taking wheelbase and track width into consideration. To get the Ackermann angle, a line has been drawn from the endpoint of the track width up to the mid-point of the rear wheelbase. The angle between the perpendicular bisector and the line drawn from the endpoint of the track width is the Ackermann angle which we have to achieve via numerous calculations. Hence, we got the Ackermann angle.

Using this value of Ackerman angle further two geometries were developed which are required for the designing of the steering system.
Starting with the theoretical calculation of inner and outer wheel angles which are used as a base for drawing the first geometry.
Inner Wheel angle = Wheelbase / (Turning Radius - (Track Width/2)) = 30.82^0
Outer Wheel angle = Wheelbase / (Turning Radius + (Track Width/2)) =46.34^0

Above formulas were used to calculate the theoretical values of inner and outer wheel angles. These angles obtained were used as initial values to generate the first geometry of turning radius shown in Fig.3.

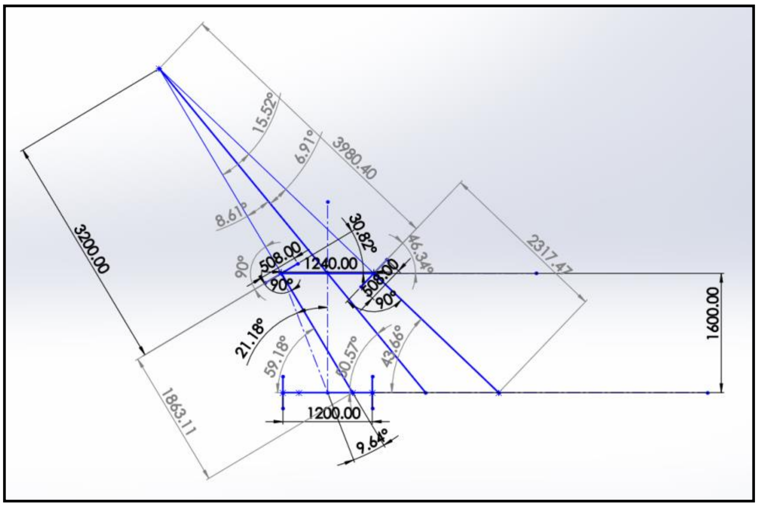

Fig.3. First turning radius geometry with wheel angles

In this geometry, the wheels are rotated up to their maximum positions. These wheel rotations angles were taken from the theoretical calculation done above. After these perpendicular lines were drawn to the centerlines of wheels up to the rear axle. Further these lines were extended on the opposite side until they intersect. The geometry was created using the values given in Table 1. The top horizontal line represents the front track width (1240mm), bottom horizontal line represents rear track width(1200mm) and the perpendicular distance between the track width and the front axle is the wheelbase (1600 mm).

Table 1

Turning radius	3200
Track width rear	1240
Track width front	1200
Wheel base	1600
Outer angle	30.82
Inner angle	46.34

And the Second geometry, shown in Figure 4 shows the anti-Ackermann geometry. The geometry is created using values provided in Table 2. This geometry is drawn to optimize the dimensions of tie rod, steering arm, rack offset so that to achieve the required wheel turn angles from first geometry and obtain the rack travel for the same, which is further required for the designing of rack and pinion.

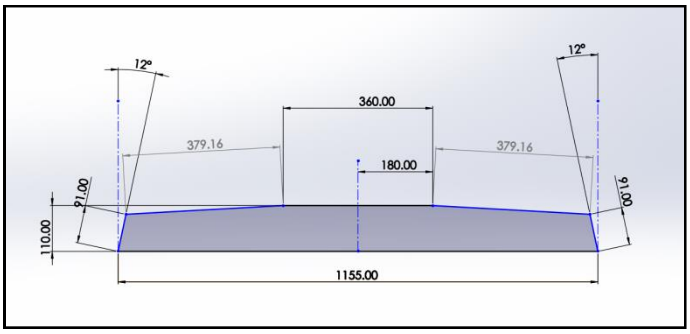

Fig.4. Anti-Ackermann geometry with all dimensions

Table 2 Dimensions from Fig. 4

Rack length	360
Rack offset	110
Wheelbase	1600
Tie rod	379.16
Steering arm	91
Ackermann angle	12

SIDEVIEW:

As we see the Fig. 5 represents the side view of the steering system. The whole steering system is 480mm horizontally. Starting from the initial position the steering wheel is at 181.65 mm from the roll bar which is located above the driver's legs, in proximity. And it is 10.98 mm below the roll bar and 274.9mm above the floor. The outer diameter of the near oval shaped steering wheel is 210mm. It is attached to the quick release actuator which is fixed to the upper shaft of the steering column that is 173 mm long and has 12mm diameter. The upper shaft attached is now holder by a pedestal bearing which is 40mm long diametrically. It is placed at 93.70mm from the steering wheel. The mount bearing mount which holds the bearing is 107.28 mm long and is welded to the roll bar or front hoop. The upper shaft is connected to a universal joint at a distance of 64mm from the bearing. The universal joint is further attached to the shaft of the steering column which is measured 350mm from center of the universal joint till the pinion casing. It has a diameter of 12mm. The angle measured between the upper shaft and the steering column shaft is 135°. The distance between the initial position of the steering column shaft and the floor is 300mm. The steering shaft is then attached to the pinion connector rod which is pressed fit. The pinion connector rod is inserted into the pinion casing which is at the angle of 35° from the floor. The pinion casing is 89mm long. The pinion casing is attached to the tertiary member of the chassis by a Mild Steel plate which is 4mm thick and length of 48mm.

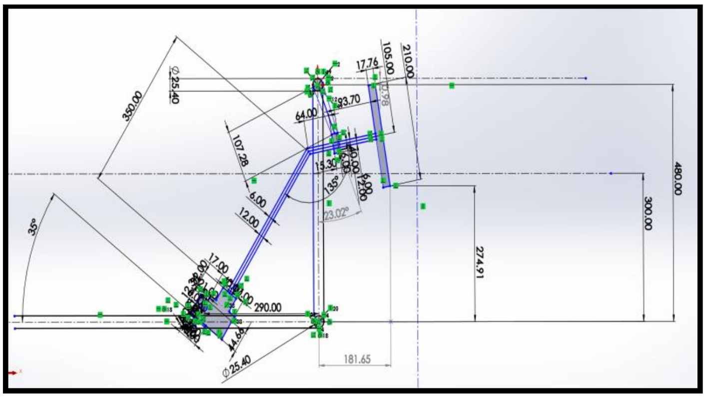

Fig. 5. Side view of the steering geometry

5. PARTS OF THE STEERING SYSTEM:

a. Steering Wheel
b. Bearings
c. Universal (UV) Joint
d. Rack and Pinion Gear (RPG)
e. Tie Rods
f. Steering Arm

a. STEERING WHEEL

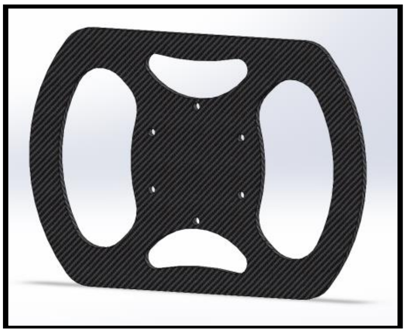

Fig. 6 Formula Student carbon fiber steering wheel

The steering wheel used in the car is a customized Carbon Fiber based wheel. It is elliptical in shape keeping in mind the FBRB-2021. The steering wheel houses the automatic gear shifting mechanism and attaches the quick release bearing is attached to it.

The placement of the steering wheel is determined by the driver based upon his ergonomic factors and keeping in mind the rules.

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b. BEARING

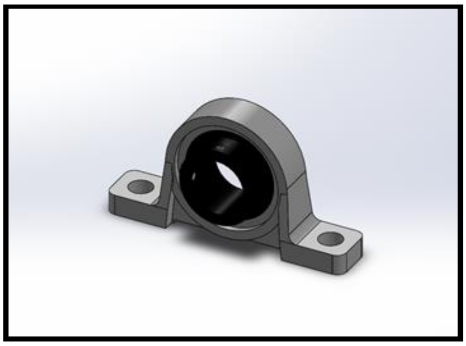

Fig. 7 Ball Bearing with casing

The bearing used in the car is a deep groove ball bearing. It is an open bearing. The open bearings have an open side where the balls are visible and are not sealed or shield. The specifications are as follows 17mm * 35mm * 10mm. The material used for the bearing is steel, brass and nylon cage.

c. *UNIVERSAL (UV) JOINT*

It is designed and selected as per the car requirements for the steering system to connect the steering wheel to the steering column. It has low side thrust on bearings and large angular displacements are possible in the steering column. Since a double universal joint is used in the vehicle and design, appearance of a disadvantage of the motion being not transmitted exactly as the input by the driver is eliminated.

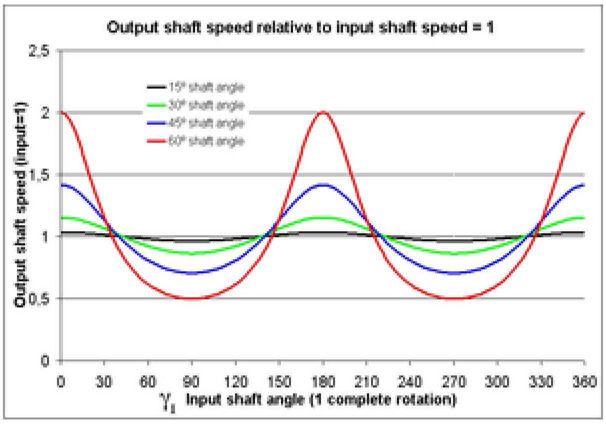

Fig. 8 Graph showing variation of the input speed to the output speed in case of a single universal joint which is a disadvantage
[Source: Wikipedia]

Hence, it was decided to use a custom made double universal joint.

Fig. 9 Double cardan/ Universal Joint

Table 3 Comparison between single and double universal joint

SINGLE UV JOINT	DOUBLE UV JOINT
It will not act as a constant velocity joint	It will act as constant velocity joint
Assembly will weigh less due to it.	Assembly will weigh slightly more.
Does not provide added length	It provides some added length.

d. RACK AND PINION GEAR (RPG)

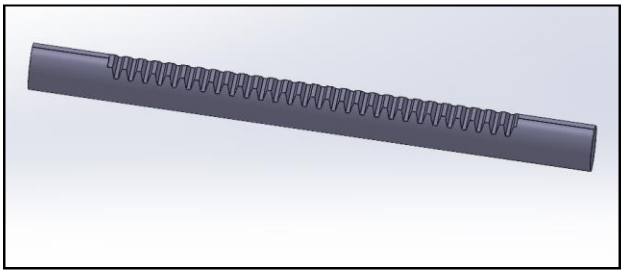

Fig. 10 Rack

The rack and pinion gear are the most crucial part of the steering geometry. The RPG must be the strongest part of the whole steering system since almost all the forces combine at the interface of the teeth. The teeth are involute in shape.

Rack:

The rack is made of solid steel billet of 16mm diameter and length 190mm on which teeth are milled.

Pinion:

The pinion gear is manufactured of solid steel billet of 70mm diameter and length 40 mm. It has a hole of 40mm diameter with a key slot.

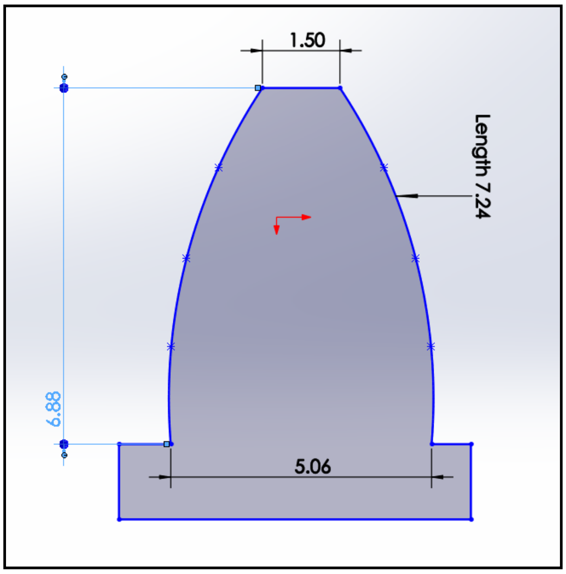

Fig.11 Sketch of Pinion tooth on DS Solidworks

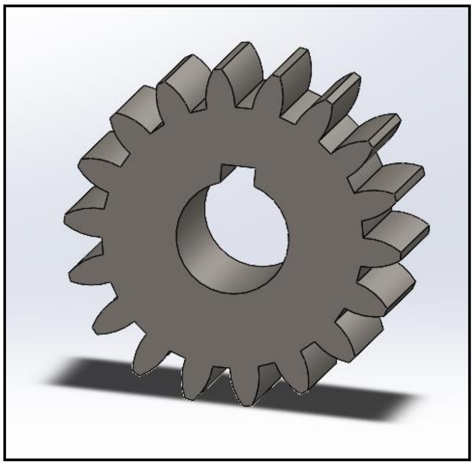

Fig. 12 Model of Pinion on DS Solidworks

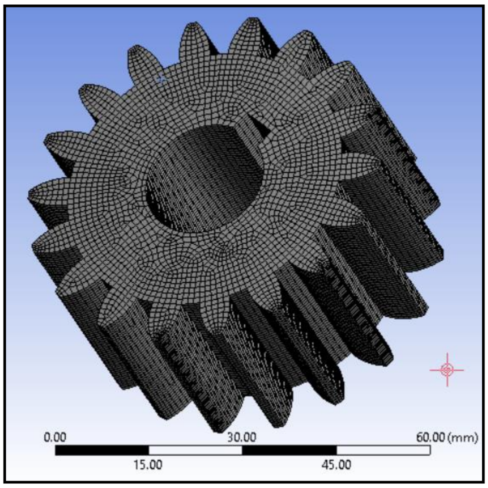

Fig. 13 Model imported in Ansys and Meshing of 1mm element size done

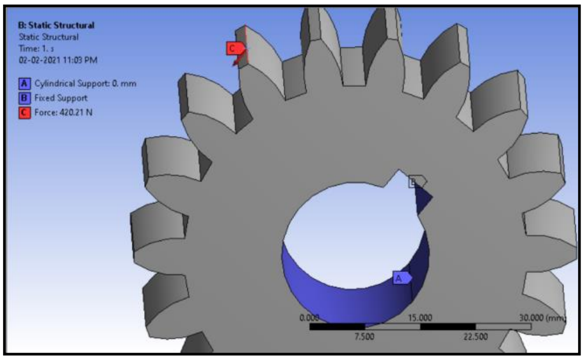

Fig. 14 Supports and Forces demonstrated in Ansys

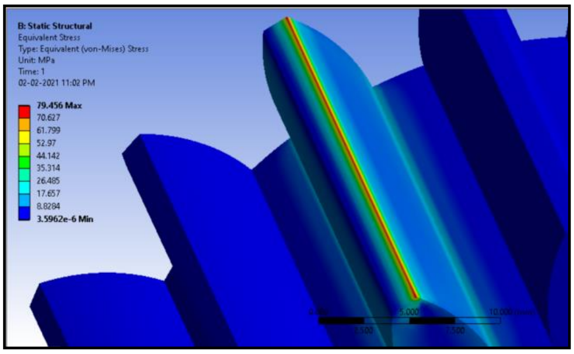

Fig. 15 Ansys result showing a maximum stress of 79.456 MPa on the edge of the tooth

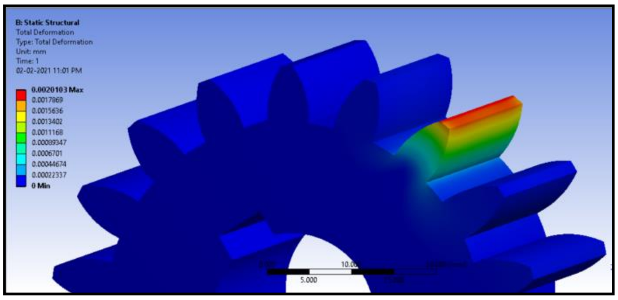

Fig. 16 Ansys result Showing the total deformation of 2.0103 µm

Since the pinion is safe from any failure hence the rack is also safe.[4]

All calculations done with reference to Bhandari[4].

Table 4 Calculations for Rack and Pinion Tooth

Name	Notation	Value	Unit
Pitch Diameter	d'	54	mm
No. of teeth on Pinion	Z_p	18	nos
Circular Pitch	p	9.4248	mm
Diametral Pitch	P	0.3333333	mm^{-1}
Module	m	3	3
Face Width	b	30	mm
Crowning		0.012	mm
Pressure Angle	α	20	deg
Addendum	h_a	3	mm
dedendum	h_f	3.75	mm
clearance	c	0.75	mm
Working Depth	h_k	6	mm
Tooth Thickness	S	4.7124	mm
Tooth Space		4.7124	mm
No. of Teeth on rack	Z_r	28	
Addendum circle diameter	d_a	60	mm
Dedendum circle diameter	df	46.5	mm

Table 5 Force calculations on teeth of rack and pinion

Name	Notation	Value	Unit
Permissible Bending Stress	$\sigma_b = S_{ut}/3$	53.33	N/mm^2
Lewis Form Factor	Y	0.308	
Tangential Force/ Beam Strength	P_t/ Sb	1478.4	N
Transmitted Torque	M_t	39916.8	N-mm
Radial Force	P_r	3307.41874	N
Resultant Force	P_n	3622.8007	N
Service Factor	Cs	1	
Velocity	V	9	m/s
Velocity Factor	Cv	0.25	
Effective Load	P_{eff}	5913.6	N
Deformation Factor	C	11400	
Error	E	0.001	mm
Dynamic Load	P_d	1485.1354	N
Tolerance	φ	4.837	mm

e. Tie Rods:

Tie rod lengths are the most flexible part of the steering geometry which can be increased or decreased upto 5mm accuracy. The tie rod is initially made of 360mm length. And billets of 10 mm each are attached to it. And rod ends with threaded length of 20mm are attached to it to help in flexible movement.

Fig. 17 Tie rod

f. STEERING ARM

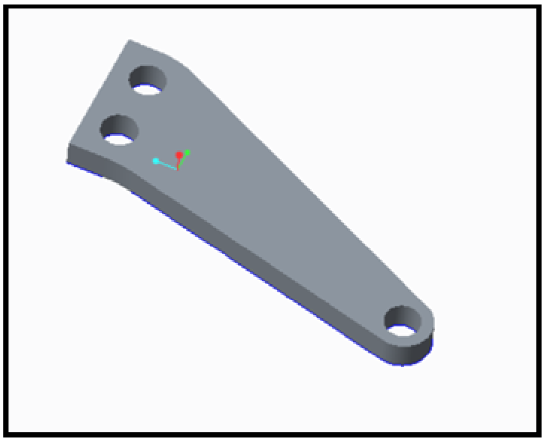

Fig. 18 Steering Arm CAD on DS Solidworks

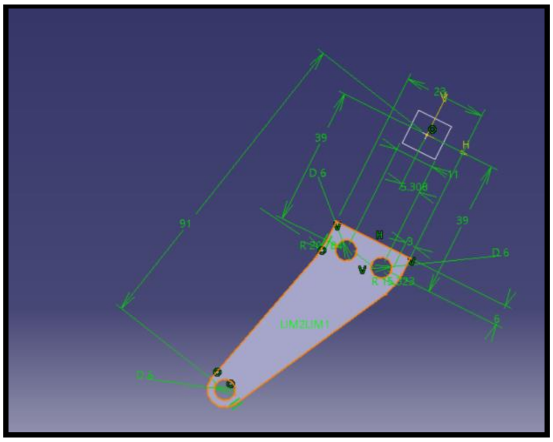

Fig. 19 Dimensions of the steering arm on DS Catia

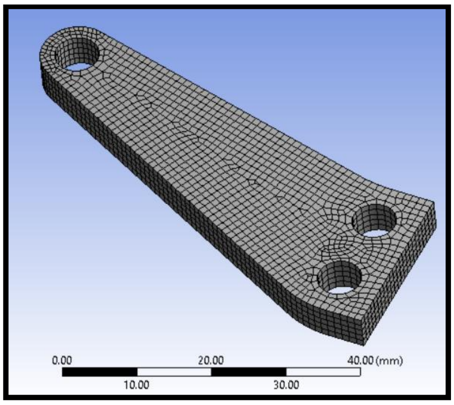

Fig. 20 Mesh on the model on Ansys

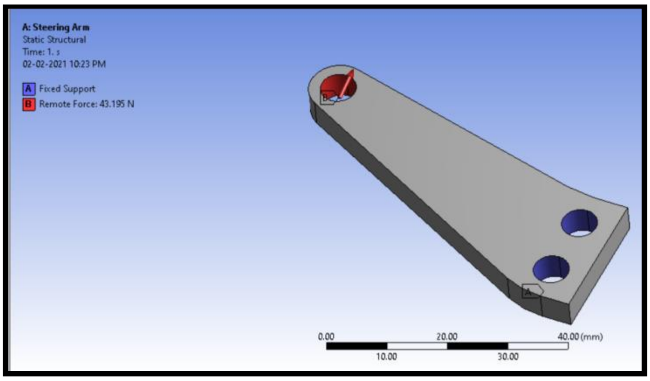

Fig. 21 Force of 43.195N and supports applied

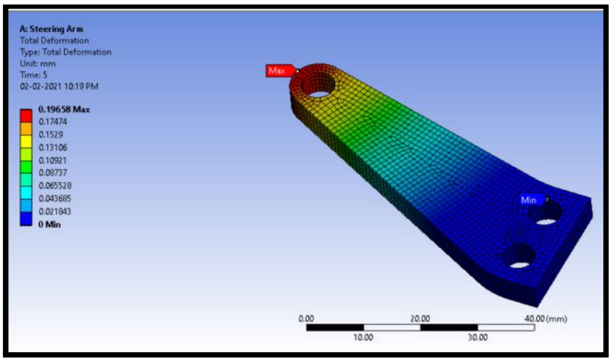

Fig. 22 Maximum deformation of 0.19658 mm found out.

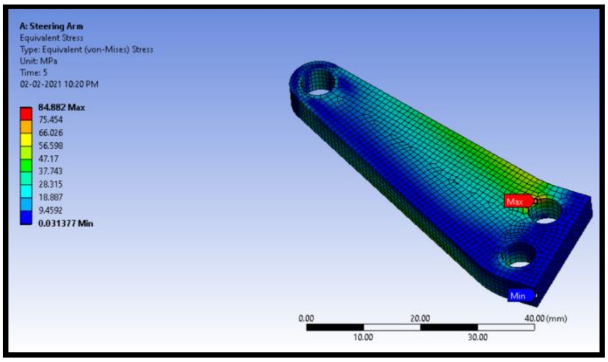

Fig. 23 Stress Analysis resulted in tensile stress of 84.882 MPa

The steering arm is the primary connecting part from the knuckle to the tie rod which forms a part of the basic 4-point steering geometry. The steering arm takes into consideration the position of the Ackermann angle and the percentage Ackermann of the steering system. In the current steering arms, the arms are inclined at an angle of 12 degrees with one end connecting to the specially designed knuckle and the other to the tie rods.

6.DESIGN AND CALCULATIONS

Wheelbase(b)=1600mm

Rear Track Width(a)=1240mm

Front Track Width(c)=1200mm

Turning radius(R)=3200mm

Steer Angle

ϴ = arctan(R/L)=Wheelbase/Radius of turn

= arctan(1600/3200)

$=26.56^0$

Steering Force Calculations:

The basic concept is that torque required to turn the wheel should be more than resisting torque by friction

The friction force at the tire is = μ*Reaction at each tire

= 0.7*9.81*58.15

= 399.91605N

As we know that steer happens about the kingpin axis of the car

Input torque from the ground (on one wheel) = force of friction x perpendicular distance from contact patch to kingpin axis we can take it approximately to scrub radius

The scrub radius is the distance in front view between the kingpin axis and the center of the contact patch of the wheel, where both would theoretically touch the road

Input torque from the ground (on one wheel) = frictional force* perpendicular distance from contact patch to kingpin axis

Here scrub radius is approximately equal to the perpendicular distance = 47.81mm

Therefore, Torque due to frictional force = 0.04781*399.91 =19.1196Nm

Torque due to lateral push = force on tie rod (f_t) x 0.091m

19.1196 = Ft*0.091

So, Ft i.e., force on tie rod = 210.1065N

The total force on rack = 210.1065*2 = 420.213N

Radius of the pinion = 30mm = 0.030m

Torque on pinion = 420.213N*0.030m = 12.60639Nm

The radius of our steering wheel = 105mm = 0.105m

Force applied by the driver to steer the wheels = torque on steering wheel / radius of the steering wheel

=12.606 / 0.105 = 120.060857N

Nature of Steering (Oversteer or Understeer)

Cornering force = M*v^2/R

Fyf = Rear axle lateral force

Fyr = front axle lateral force

Fy = Fyf + Fyr

Calculation at 3.5m radius and 10 m/s:

FY = 182*10*10/3.5

= 5200 N

Sprung mass of vehicle = 160kg

Sprung mass at front axle and rear axle

M_r = 160*b/wheelbase = 160*866.67/1600 = 86.667 kg.

M_f = 160*a/wheelbase = 160*733.33/1600 = 73.333kg

At fixed radius of 3.5m and different velocities values of F_{yf} and F_{yr} are as follows:

At v = 0m/s, F_{yf} and F_{yr} both are 0 since v = 0 m/s

At v = 1.6m/s, F_{yf} = 53.637 N and F_{yr} = 63.3907N

At v = 2.78 m/s, F_{yf} = 161.9276 N and F_{yr} = 191.3706 N

At v = 4.16m/s, F_{yf} = 362.5919 N and F_{yr} = 428.5213N

At v = 5.55 m/s, F_{yf} = 645.3828 N and F_{yr} = 762.7315 N

At v = 6.99 m/s, F_{yf} = 1023.7308N and F_{yr} = 1209.8738N

$F_{yf\,avg}$= (53.637+161.9276+362.5919+645.3828+1,023.7308) /5

= 449.4540N

F_{yravg}= (63.3907+191.3706+428.5213+762.7315+1,209.8738) /5 = 531.1775N

M_r = (Cg to rear distance* sprung mass)/ wheelbase = 866.67*160/1600 = 86.667

M_f = (Cg to front distance* sprung mass)/ wheelbase = 733.33*160/1600

= 73.333

Steering Ratio:

The steering ratio is the ratio of how much the steering wheel turns in degrees to how much the wheel turns in degrees. Approximating maximum turn to be of 76.042 degrees and steering wheel movement to be 180 degrees the steering ratio can be calculated by two methods:

- **Method-1**

Steering ratio = total angle of steering wheel / total steering angle

= 180 / 76.042

= 2.368:1

Position of rack: Rack Height: - 137mm from Ground

Steering arm length from KPI to Pivot Point = 91mm

Ackerman Angle = tan (KPI to KPI distance/ (2 * wheelbase)) = 19.846414deg

Ackerman Percentage=75.44%

Rack travel= 60 mm

- **Method-2**

Inside and outside Steering angle

R=turning radius of wheel= 3200mm

a= track width=1240mm

b=wheelbase =1600mm

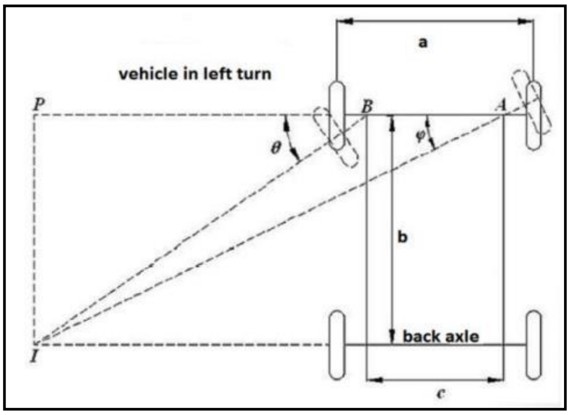

Fig. 24 Basic Dimensions with notations used for calculations

c=KPI to KPI distance= 1155mm R= b/sinΦ + (a-c)/2

Φ = 30.446deg

Now cotΦ - cotθ= c/b

Here, we get θ = 45.596deg

Hence, total steering angle = 30.446deg + 45.956deg= 76.042deg

The total angle turned by steering wheel=180deg

We know, Steering ratio = total angle of steering wheel / total steering angle = 180 / 76.042 = 2.368:1

From both the methods the steering ratio is same and hence is verified.

7.Conclusions

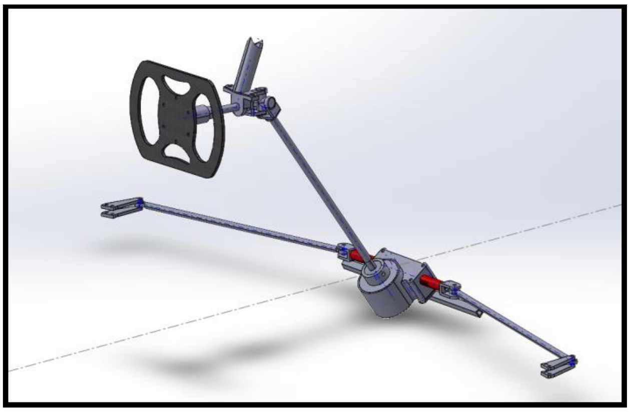

Fig. 25 Completed CAD Model of the steering system

In this work, a numerical study and experimental validation were presented for building a steering system for FSAE car.

The designed Anti-Ackerman Steering system is a reliable and configurable system.

The designed system gave us greater stability while taking sharp turns which further aided the outer wheel to stay in contact with the ground even at high speeds.

The use of advanced materials and manufacturing processes helped to build a lightweight system that is within the budget limit.

The use of Anti-Ackerman system provides more stability during cornering at high speed increasing tractive force on the outer wheel so as the tire remains in contact with the road.

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