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DESIGN PROCEDURE AND OPTIMIZATION OF STEERING SYSTEM FOR FORMULA STUDENT CAR

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ABSTRACT

Steering system is one of the major part of any Automotive design. It is responsible for transferring drivers input to the wheels and gives feedback from road to driver. To design a proper steering system for a formula SAE vehicle, several design parameters needed to be determined. The most important design intent for this system was to provide the driver with a system that did not require excessive steering input force or excessive steering wheel rotation, proper wheel feedback, and adequate wheel steering angle to allow the driver to navigate the tightest corners on the autocross course.
This report gives clear idea about how the steering geometry has to be decided for formula student car by using CAD Software Solidworks, and how to select from different possible alternatives. Also, designing of steering components like rack and pinion, shafts and Steering wheel is explained in the report.
The overall report can be divided into two objectives. The first is to design the proper steering geometry for the car. The second objective is to design the steering system components, so that the steering effort decreases without compensating the feedback which is obtained from the road.
Designing and optimizing of steering system and its components is done considering the rules provided by Formula Bharat 2020, ergonomics, drivers safety, Components Manufacturing, Assembly and performance. The CAD file is entirely developed in Solidworks 2018-19. Static force analysis on components is also performed in Solidworks 2018-19.

KEYWORDS- FSAE Steering System, Ackerman Steering Geometry, Steering Components, Spiral Bevel gears, Simulations, DFMA principle.

1. INTRODUCTION

Formula Student: The Challenge
Team Ojaswat is a formula student racing team consisting of students, from the Charotar University of Science &Technology. Each year the team designs, builds, tests, and eventually races their car against other university teams from all over the world in the Formula Student competition.

The students are to assume that a producing firm has engaged them to supply a prototype car for evaluation. The intended sales market is the nonprofessional weekend auto crosser sprint race and the firm is planning to produce 1,000 cars per year at a cost below 10 lakhs.

The car must be low in cost, easy to take care of, and reliable, with high performance in terms of its acceleration, braking, and handling qualities. Watched closely by industry specialists who volunteer their time each team will go through the following rigorous testing process of their car: Static events: Design, Cost, and Presentation Judging– Technical and Safety Scrutineering – Tilt Test to prevent cars from rolling over – Brake and Noise Test. Dynamic Events: Skid Pad – Acceleration – Sprint/qualification – Endurance and Fuel Economy – Autocross.

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Importance of Steering System

The primary purpose of the steering mechanism is to permit the driving force to guide the vehicle. The most common steering arrangements to show front wheels employing a hand operated wheel which is positioned ahead of the driving force via the steering column which can contain universal (which can also be a neighborhood of the collapsible steering column design), to permit it to deviate somewhat from a line .other arrangements are sometimes found on different vehicles, for example, a tiller or rear wheel steering, tracked vehicles like bulldozers and tanks usually employ differential steering mechanism. A basic steering mechanism has 5 main parts: Steering box, Linkages, Knuckle or upright, Column Shaft & wheel. When the driving force turns the wheel, a shaft from the steering column turns a gear. The gear moves tie rods that hook up with the front wheels. The tie rods move the front wheels to show the vehicle right or left.

Problem Definition

A typical formula student steering system consists designing several parts: Steering Geometry, Steering gear box, Steering column, Steering wheel, etc. So far Team Ojaswat has designed the rack and pinion based steering system with Ackermann Steering geometry. In the design, team is using bevel gears instead of universal joint. As transmitting component from steering wheel to pinion is gear, there is problem of free play in pair of bevel gears. That induces free play in steering system. Also, the assembly of the system in the car was very difficult.

The main challenge for our team was to design and optimize new steering system, in which the free play in rack and pinion which occurs due to poor manufacturing and assembly has to be reduced. Also, the play in pair of bevel gears has to be reduced. And the most important was to design the Steering system according to principle of DFMA.

After preparing several CAD models and performing several analysis and simulation all the above problems were solved by making important changes in design. The new optimized design was lighter in weight and more efficient.

Design Constraints

Considering Formula Bharat 2020 rule book which is affiliated with FSG (Formula student Germany) following were main constraints considering Steering system design.

Figure:

T 2.6	**Steering**
T 2.6.1	Steering systems using cables or belts for actuation are prohibited.
T 2.6.2	The steering wheel must directly mechanically actuate the front wheels.
T 2.6.3	The steering system must have positive steering stops that prevent the steering linkages from locking up. The stops must be placed on the rack and must prevent the tires and rims from contacting any other parts. ~~Steering actuation must be possible during standstill.~~
T 2.6.4	Allowable steering system free play is limited to a total of 7° measured at the steering wheel.
T 2.6.5	The steering wheel must be attached to the column with a quick disconnect. The driver must be able to operate the quick disconnect while in the normal driving position with gloves on.
T 2.6.6	The steering wheel must be no more than 250 mm rearward of the front hoop. This distance is measured horizontally, on the vehicle centerline, from the rear surface of the front hoop to the forward most surface of the steering wheel with the steering in any position.
T 2.6.7	The steering wheel must have a continuous perimeter that is near circular or near oval. The outer perimeter profile may have some straight sections, but no concave sections.
T 2.6.8	In any angular position, the top of the steering wheel must be no higher than the top-most surface of the front hoop.
T 2.6.9	The steering rack must be mechanically attached to the chassis.
T 2.6.10	Joints between all components attaching the steering wheel to the steering rack must be mechanical and visible at technical inspection. Bonded joints without a mechanical backup are not permitted. The mechanical backup must be designed to solely uphold the functionality of the steering system.

Steering Design Constraints

Figure:

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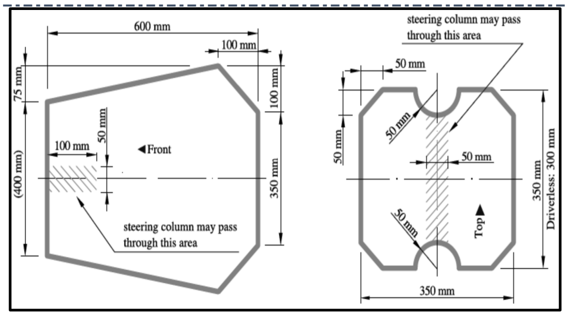


Cockpit Templates

2. DESIGN AND DEVELOPMENT

Geometry Selection

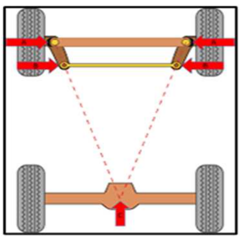
The first step was to define which type of geometry should be used for the car.

1) *Ackermann:* In Ackermann geometry, during cornering the inner tire turns more than the outer tire with the same input steering angle.
2) *Anti-Ackermann:* In Anti Ackermann geometry, during cornering the outer tire turns more than the outer tire with the same input steering angle.
3) *Parallel steering:* In parallel steer both the wheels will turn with the same angle with the same input steering angle.

The decision for the steering geometry was on the basis of speed of the car during cornering. In most of the Formula student car the average speed of the car is less than 80 Km/h. Also, parallel Steering geometry can be used but due to the design of the Autocross and Endurance track which includes sharp cornering, the cornering Speed further reduces to 30km/h. For which if parallel steering Geometry is used then the wear of the tire would be more, without any extra advantage for the car. In road vehicles, the rate at which, each wheel turn with respect to each other determines how the car will turn at both low and high lateral accelerations. This concept can be understood easiest in the low lateral acceleration case, for which weight transfer and tire slip is negligible. In this case, a pure Ackermann steering configuration would be ideal, since the inside wheel will turn with a smaller radius than the outside wheel.

This geometry can be easily defined if the wheelbase, kingpin locations, and moment arm lengths are known. A straight line must be made at neutral steer to the centre of the rear wheelbase from each of the kingpins at the specified height. The steering pickup location will lie on that line at a point which is the distance of the moment arm length from the kingpin. This is considered as a 100% Ackermann geometry. Following research from the Milliken Book (CITATION), the important correlation between Ackermann steering geometry and handling at different lateral accelerations was made. At low lateral accelerations, an Ackermann geometry system would be ideal, yet at higher accelerations which a race car would see, a parallel steer or even reverse Ackermann system would be preferable.

Figure:



Ackermann Geometry

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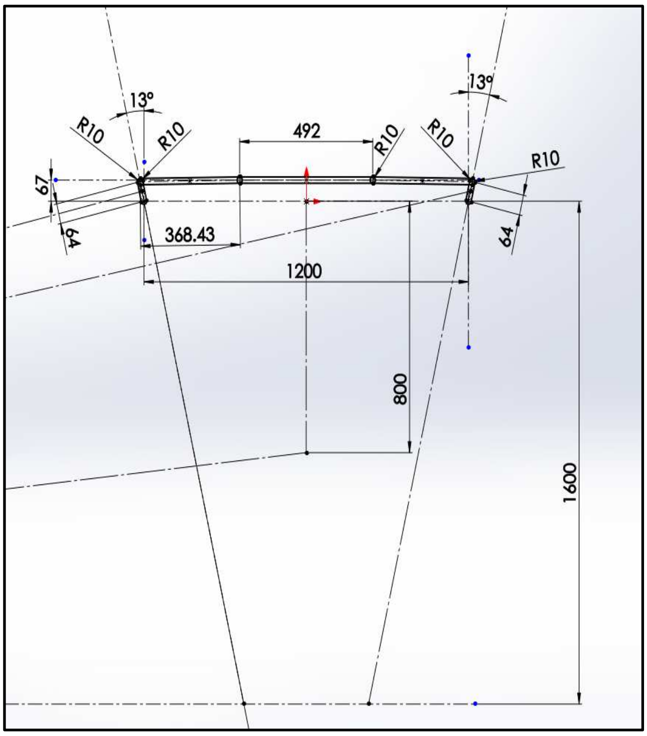
Figure:



Steering Geometry Sketch

Other than this information, there was no quantifiable perfect geometry since each system had drawbacks. A parallel or reverse Ackermann system would be difficult to steer at low speeds, such as during a corner with slow speed or moving the vehicle through the pits. Reverse Ackermann geometry is also difficult to achieve without longitudinal translation of the steering rack since the pickup points for the steering arms are on the outboard side of the kingpins. While this is possible to package at low steering angles, at higher steering angles collision becomes a major issue between the tie rod, upright, and wheel.

For instance, on a normal road vehicle, the steering wheel may rotate approximately 1000 degrees from lock to lock. In this scenario, the driver must reposition their hands if the needed steering angle exceeds 180 degrees from neutral. In order to eliminate time lost during repositioning and increase safety, the maximum steering angle from neutral was set to 120 degrees to safely and comfortably allow the driver to navigate the hairpin corners of the autocross track. If the system was truly optimized for only steering force, the steering angle would be increased along with the moment arm length to allow for a lower input force. If the input angle in this case is set to the maximum angle, the steering force will be minimized. Also, Anti-Ackermann is beneficial if the average cornering speed is more than 100 km/h. which is not possible in the above case. So, from above study it was finally concluded to use Ackermann Steering geometry, as it would help in each of the case explained above.

For different positions of the rack, Ackermann Percentage was calculated by creating the sketch of the geometry. The minimum radius of the inside of the hairpin corner of the autocross track was stated to be 3m.

Due to the lack of professional drivers, the system needed to be able to facilitate an average driver during this course. A minimum turning radius of 3m was set as a goal.
Inner Turning Wheel Angle is 28 degrees.
Outside Turning Wheel Angle is 25.04 degrees.
Turning radius of inner wheel is 2.475 meters.

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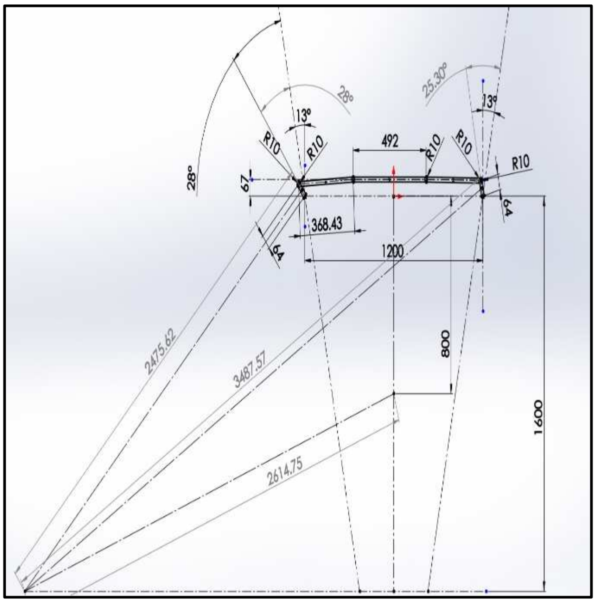
Turning radius of outer wheel is 3.4 meters.
This above value where finalized from the Sketch shown below to use for the geometry of steering. As it satisfies the minimum turning requirement for the Autocross Event.

Tables:

Table 1. Ackermann Percentage Calculation

SR. NO.	wheelbase	Track front	A inside	A outside	Tan outside	Ackerman value	Ackerman angle	Ack %	Offset from wheel centreline	Tie Rod length
1	1600	1200	10	9.65	0.168424	0.192775	10.91136	91.6475	0	368.4
2	1600	1200	20	18.61	0.324805	0.429412	23.23929	86.0611	0	368.4
3	1600	1200	30	26.91	0.469668	0.725077	35.94501	83.4608	0	368.4
4	1600	1200	40	34.48	0.601789	1.096839	47.64423	83.9555	0	368.4

Figure:



3-meter radius cornering

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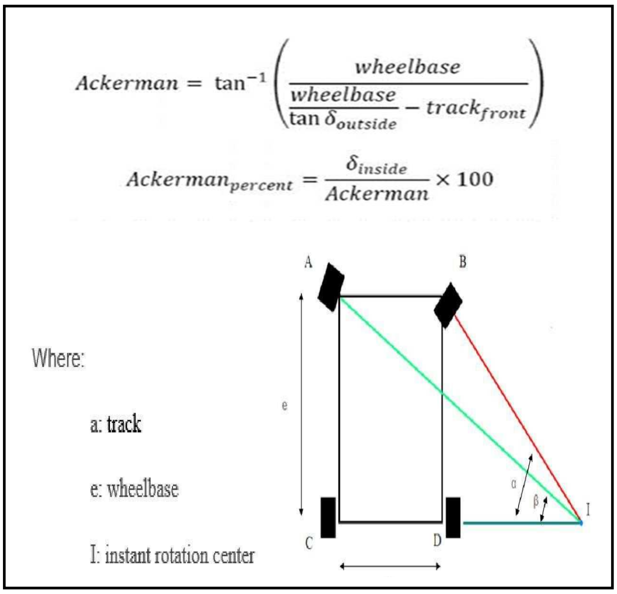
Figure:



Ackermann Percentage Formula

Analysis of Alternatives

Before being able to run, the car has to pass a technical inspection where the judges check that the car fulfil all the rules. Besides the rules just listed, there are two templates used to measure some minimum dimensions in the cockpit, from the steering wheel to the pedals. Templates are shown in Figure (Cockpit Templates). So, the next step is to decide the location of rack such that rule of templates is easily satisfied.

These are the different possible configurations of the steering system shown below:

Tables:

Table 2. Different possible steering configurations

	Option 1	**Option 2**	**Option 3**
Location of Rack and Pinion.	Over the legs	Under the legs	
Connection Steering Wheel to Rack and Pinion.	Direct	Universal joints	Bevel Gears
Shaft Section	Round	Square	Hexagonal
Shaft Material	Steel	Aluminium	Carbon fibre
Rack pinion	Commercial	Custom	

Position of Rack

The next Step was to decide the position of steering arm which would be connected with Rack with the help of Tie Rods, to transfer the motion from Rack to the wheel. In a vehicle, a wheel using a standard upright design with a rack and pinion, the wheel assembly can be steered by attaching the steering rack to the upright in one of the four quadrants of the upright. Since a low centre of gravity is a driving factor in this system design, the upper quadrants were not considered since the steering arm, tie rods, and steering rack would have to be located at an elevated position, therefore raising the centre of gravity. The first step is to think where to place the rack-pinion and the advantages of locating it over or under the driver's legs.

The conclusion was that, locating it over the driver's legs would increase the height of the centre of gravity, besides it would be more difficult to fulfil with the rule of the second template. So, the decision is to place the rack-pinion under the driver's legs attached to the chassis.

Reason for Selecting Bevel gears.
The distance from the centerline of the steering wheel to the centerline of the steering rack is over 300mm. The configuration using universal joint is not necessarily a strong construction due to the very short upper and intermediate shafts. Each shaft will need to be supported to avoid idle degrees of freedom which could cause the system to lock up. The bearings for the shafts will need to be placed very close to each other, and only radial load ball bearings would be able to be used due to the lack of room for shaft collars to transmit axial load to a roller bearing or spherical roller bearing. The other major problem this initial configuration exhibits is the high angle between each shaft, which would have to be increased to meet the packaging requirements to 40 degrees for each u-joint. A normal u-joint becomes very inefficient at transmitting torque at higher angles. U-joints also do not transmit rotation at a constant speed with respect to the input speed, causing a non-linear steering speed. This can be mitigated by properly phasing two u-joints, but due to the large angle, a geared system will be better to use. ("Ackermann Steering Geometry.").

After analysis of the steering side view geometry was done, it was determined that the longitudinal distance could be reduced through the use of a right-angle gearbox, thus eliminating the intermediate shaft, all u-joints, and lengthen the upper shaft. This increases support area for the upper shaft and allows for better packaging of the lower section of the steering shaft. In order to allow the system to transmit torque at an angle of 90 degrees, bevel gears were selected. Due to the total pinion angle of the steering rack being comfortably in the acceptable steering angle range, a reduction in bevel gears wasn't needed.

During testing of several bevel and miter gear sets, it was found that straight cut bevel and miter gears did not have a smooth mesh and had an unacceptable amount of backlash. To remedy this case, spiral bevel gears were selected to produce a smooth, constant mesh with minimal backlash, which is completed by always having a region (i.e., is line Contact) of the teeth in mesh at one time, unlike straight cut gears where only small sections (i.e., point contact) are in constant mesh at one time.

Figure:

Bevel gears

The position of the Rack and Pinion was finalized according to different Ackermann percentage calculated. Also, the medium for transferring force from Steering wheel to Pinion was decided from different aspects. Pair of Spiral bevel gears were selected to transfer motion from steering wheel to Pinion, which transmits the force perpendicularly which eliminates use of Universal joint.

Material Selection.
Material Selected For different components were selected on the basis of different parameters like:
Strength, Durability, Weight, Cost and Availability.
Different Materials used for different component were: -
1) S.S. (Pinion, Rack, Shaft).

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2) Aluminum (Rack Housing, Bevel Housing, Rack Support, Clevis).
3) Hardened M.S. (Spiral Bevel gears).
4) Wood. (Steering Wheel)
5) Brass (Bushings for Rack Housing and Rack Support.)

3. CALCULATIONS

Pinion and Rack

The next step is to design the gear i.e. rack and pinion on the basis of normal force acting on the wheel. To overcome this force, we have to calculate the moment to be transferred through the pinion to the wheel. we have to calculate the total force that the driver has to put to steer the wheel. We will study the highest value of the force to turn the wheels. This force appears when the car is stationary i.e. static force calculation, and it starts the movement of the wheel. Weight of the car = 320kg. Now, considering the weight distribution as 45% on front side and as engine is assembled at rear portion, so 55% weight at the rear portion.

For this vehicle, we are going to design the front-wheel steering system, so for that we are considering 45% weight of the body and further calculations are based on that.
Weight at front side = (320*45)/100 = 144 Kg
Now weight on each front wheel= 144/2 = 72 Kg
Due to this weight, the generated force is in vertical (upward) direction and is given by,
= 72*9.81
= 706.32 N (Normal Reaction)
Now, by assuming that uniform pressure is acting on tire.
which is -
P (pressure) = 706.32/contact patch area
Contact patch area is taken as area of circle with radius =88.9 mm. Here 180 mm is considered as width of the wheel.
Now, according to uniform pressure theory, moment required to turn the tire is-
Mt= (2*(co-efficient of friction) *weight*radius)/3
= (2*1*706.32*88.9)/3
(taking co-efficient of friction=1)
We get, Mt= 41861.232 N mm
$M \geq 1.26 * \sqrt[3]{((Mt))/(Y*\sigma b*\Phi m*Z)}$
Mt= 41861.232.
Y= 0.3472 (From Databook.)
σb= Sut/3 = 530/3 = 176.67 (For SS Material).
Φm = b/m= normally taken as Constant. = 10.
Z= 21 (Assumption).
M≥1.866

To decrease the module, the ratio b/m is increased to 15. The new Calculated module was M=1.67. So, the nearest module value which was possible to manufacture was 1.75 or 1.5. For ease of Manufacturing, the selected module value is 1.5 for the gear and performed Simulation on the gear to validate the calculations.
Material Used for Manufacturing of Pinion is S.S.

Material is selected for reducing the wear and tear during application of the rack and Pinion.
Specifications of Pinion: -
1) Module- 1.5
2) Teeth- 21
3) Material- S.S.
4) Face width- 20mm.
5) Shaft Diameter- 10mm.

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Specifications of Rack: -

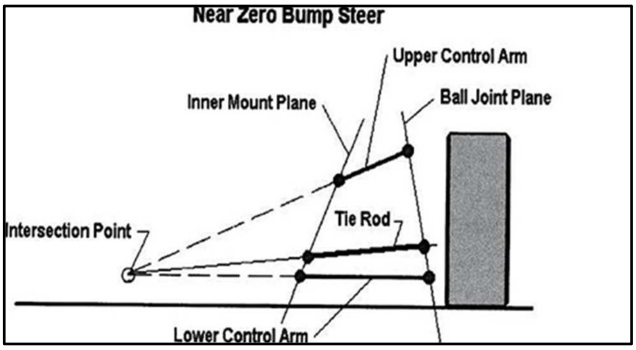
1) Module 1.5
2) Diameter 20
3) Material S.S
4) Teeth Length 120mm.
5) Total length 420mm
6) Rack travel per Revolution 98.96mm

Consideration for Rack Length

For selecting rack length, the Phenomenon of bump steer was considered and for selecting teeth length on rack the sketch shown in Figure steering geometry sketch was considered.

Bump steer or roll steer is that the term which is used for the tendency of the wheel of a car to steer itself because it moves because of the suspension stroke. It's mostly measured in degrees of steer per meter of upwards motion or degrees per foot.

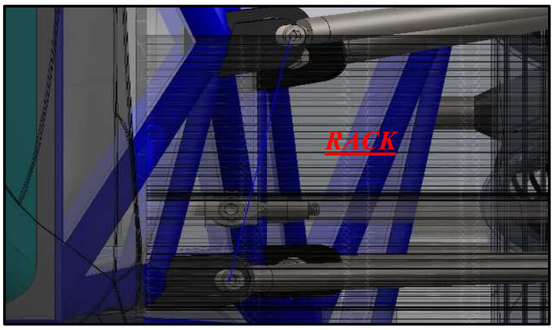
Figure:



Condition for zero bump steer

By adding this parameter, the drivability of the vehicle was increased. The condition for reducing the Bump Steer Phenomenon is as shown in figure

Figure:



Actual design

Calculation of Shaft Diameter

By using the maximum steering effort, the diameter of steering column can be calculated and as a result we can also finalize the material type and grade which can withstand the torque transmitted by steering effort.

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$$\tau_{max} = \frac{T*r}{J}$$

Where:
T: torque in the steering column
R: ratio column
J: Inertia for hollow columns and given by $= \frac{\pi}{32} d^4$

Here T(torque) = 41861.232 N mm=Mt
τ_{max} = 160 M Pa
By taking the **factor of safety = 2.5**
Therefore,
τmax = 160/2.5 = 64 M Pa
As per the formula
T= 3.14 / 16 * (d^3) * τ
41861.232= [3.14 * (d^3) * 64] / 16
Hence, **d= 14.93 mm**
Therefore, the steering shaft of diameter is taken as 15 mm.

Calculation of Steering Wheel Diameter
Weight of the car = 320kg
Now, considering the weight distribution as 45% on front side and as engine is assembled at rear portion, so 55% weight at the rear portion.
For this vehicle, as the front-wheel steering system is to be designed, so for that we are considering 45% weight of the body and further calculations based on that.
Weight at front side = (320*45)/100 = 144 Kg
Now weight on each wheel= 144/2 = 72 Kg
Due to this weight, the generated force is in vertical (upward) direction and is given by,
= 72*9.81
= 706.32 N (Normal Reaction)
Torque to be Produced at pinion, T = F*Radius of pinion.
= 706.32*16
= 11301.12 Nmm.
So that from the torque and assuming the applied force by driver on steering wheel the radius of steering wheel is calculated.
F = Applied force on steering wheel by driver.
= 100N (Assumed)
R= Radius of steering wheel.
T = F*R
11301.12 = 100*R
R= 113.0112 mm.

Therefore, the final Diameter for steering wheel is taken as 240 mm. Certain simulations on steering wheel were done to test its strength during its operation. Force taken for Simulation was 250N. Material selected for the steering wheel was Teak wood. This material was selected to reduce the weight and improve the design of the steering wheel aesthetically.

Problems and Its solution
1) *Problem: -*
As the Cross section of the Rack was Circular, its linear motion was Smooth. But it was having some rotational Motion of near about 6 degrees.

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Due to which the play was increasing in the system and during the riding of the vehicle the stability of the vehicle was decreasing.

Due to this driver's feedback was that, he cannot handle the car smoothly and the steering wheel constantly moves on its own as due to increase in play.

Solution: -
For this the solution was found that a slot should be made on rack such that this horizontal works as guide way for the rack. This technique reduced the rotational motion without disturbing the sliding motion of the rack.

2) Problem: -
The second problem was that, it was difficult to remove assembly once it was assembled perfectly.
Solution: -
For which we made a removable plate assembly as shown in figure.

Figure:

Removable plate

Due to which if any problem occurs and if changes in bevel gear assembly is to be made, then directly changes can be done in the bevel gear assembly without taking out the whole steering assembly outside.

3) Also, for reducing extra parts, the positive stops were made in clevis and Rack Support which reduces complexity during assembly.

4. SIMULATIONS

Pinion
Study Properties
Analysis type Static
Mesh Solver type FFEPlus
Selected Material: AISI 1020
Model type: Linear Elastic Isotropic
Yield strength: 3.51571e+08 N/m^2
Tensile strength: 4.20507e+08 N/m^2
Elastic modulus: 2e+11 N/m^2
Poisson's ratio: 0.29
Mass density: 7900 kg/m^3
Shear modulus: 7.7e+10 N/m^2
Fixture Details
Entities: 4 face(s)

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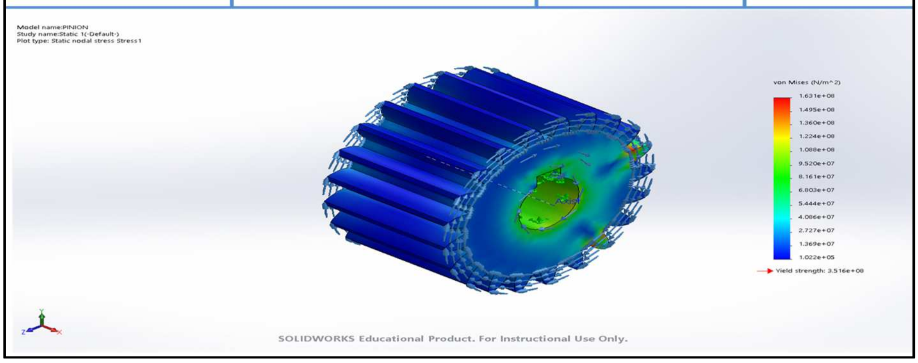
Type: Fixed Geometry
<u>Load Details</u>
Entities: 4 face(s)
Reference: Axis1
Type: Apply torque
Value: 41.861 N.m
Mesh type Solid Mesh
Mesher Used: Standard mesh
Jacobian points 4 Points
Element Size 2.37288 mm
Tolerance 0.118644 mm
Mesh Quality Plot High
<u>Mesh information - Details</u>
Total Nodes 18064
Total Elements 11220
Maximum Aspect Ratio 4.8856

Figure:

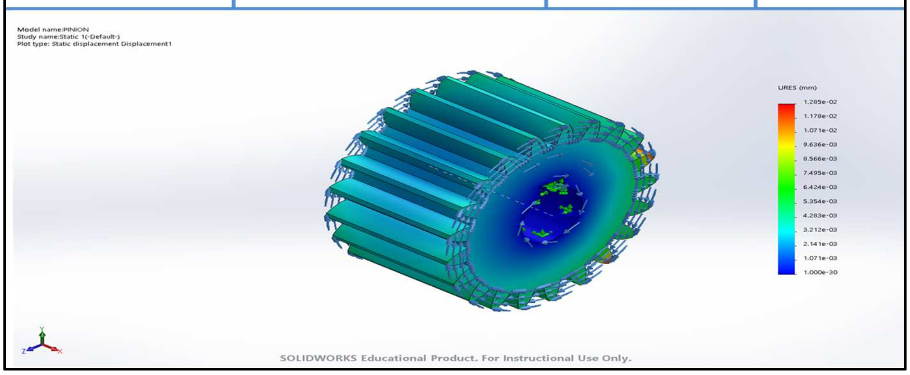
Name	Type	Min	Max
Stress1	VON: von Mises Stress	1.022e+05 N/m^2 Node: 14551	1.631e+08 N/m^2 Node: 16983



Pinion Stress

Figure:

Name	Type	Min	Max
Displacement1	URES: Resultant Displacement	0.000e+00 mm Node: 1	1.285e-02 mm Node: 132



Pinion Displacement

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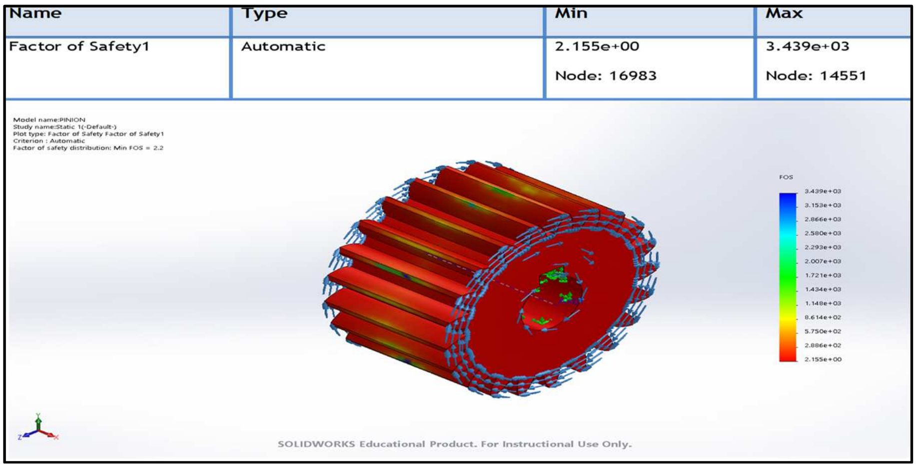
Figure:



Name	Type	Min	Max
Factor of Safety1	Automatic	2.155e+00 Node: 16983	3.439e+03 Node: 14551

Pinion F.O.S

Rack

Study Properties
Analysis type- Static
Solver type- FFEPlus
Selected Material: AISI 1020
Model type: Linear Elastic Isotropic
Default failure criterion: Unknown
Yield strength: 3.51571e+08 N/m^2
Tensile strength: 4.20507e+08 N/m^2
Elastic modulus: 2e+11 N/m^2
Poisson's ratio: 0.29
Mass density: 7900 kg/m^3
Shear modulus: 7.7e+10 N/m^2
Fixture Details
Entities: 2 face(s)
Type: Fixed Geometry
Load Details
Entities: 1 face(s)
Type: Apply normal force
Value: 1500 N
Mesh type Solid Mesh
Mesher Used: Standard mesh
Jacobian points 4 Points
Element Size 4.50491 mm
Tolerance 0.225246 mm
Mesh Quality Plot High
Mesh information - Details
Total Nodes 16944
Total Elements 10454
Maximum Aspect Ratio 6.9321

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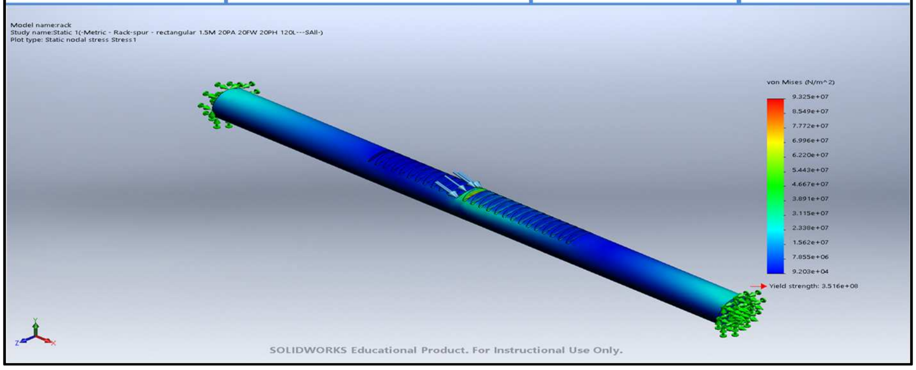
Figure:

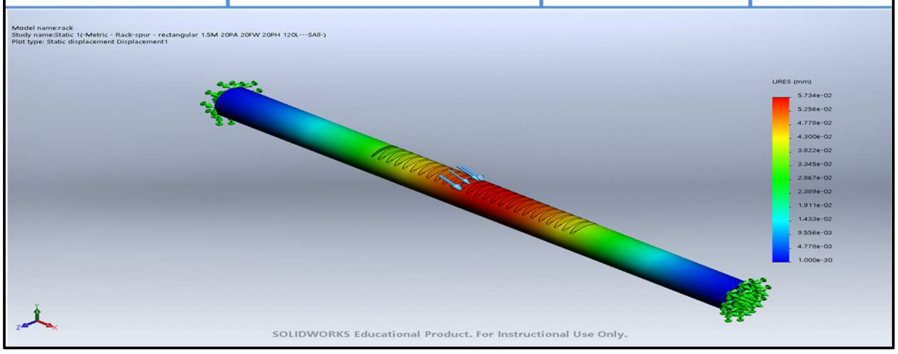
Name	Type	Min	Max
Stress1	VON: von Mises Stress	9.203e+04 N/m^2 Node: 13304	9.325e+07 N/m^2 Node: 8400



Rack Stress

Figure:

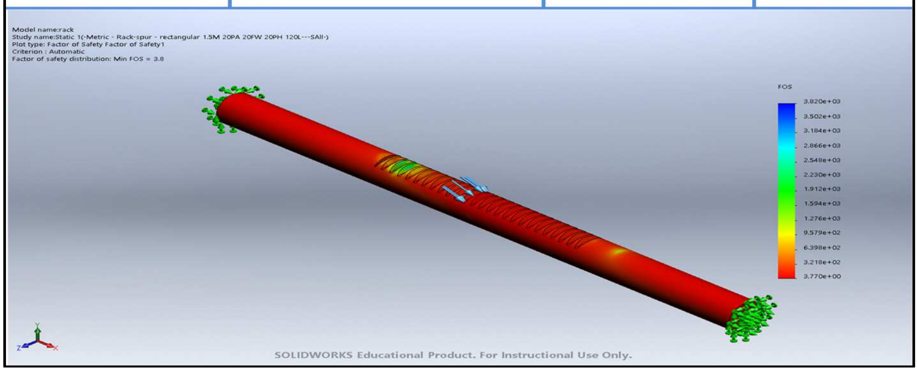
Name	Type	Min	Max
Displacement1	URES: Resultant Displacement	0.000e+00 mm Node: 8	5.734e-02 mm Node: 11923



Rack Displacement

Figure:

Name	Type	Min	Max
Factor of Safety1	Automatic	3.770e+00 Node: 8400	3.820e+03 Node: 13304



Rack F.O.S

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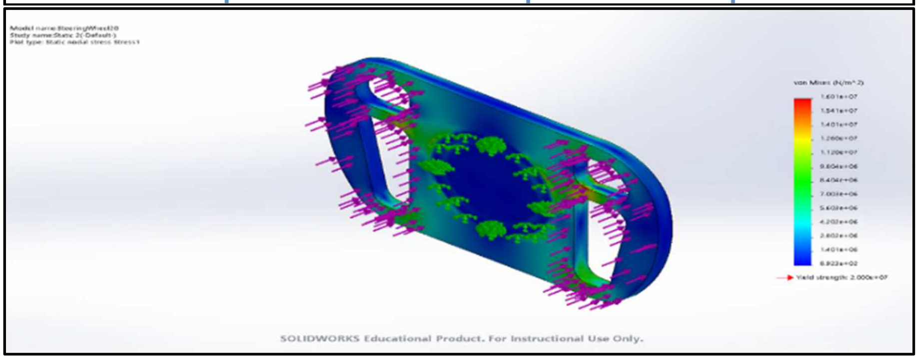
Steering Wheel

Force perpendicular to the axis of steering wheel.

Study Properties
Analysis type Static
Solver type FFEPlus
Name: Teak wood
Model type: Linear Elastic Isotropic
Yield strength: 2e+07 N/m^2
Elastic modulus: 3e+09 N/m^2
Poisson's ratio: 0.29
Mass density: 159.99 kg/m^3
Shear modulus: 3e+08 N/m^2
Fixture Details
Entities: 4 face(s)
Type: Fixed Geometry
Load Details
Entities: 2 face(s)
Type: Apply normal force
Value: 250 N
Mesh type Solid Mesh
Mesher Used: Standard mesh
Jacobian points 4 Points
Element Size 4.42638 mm
Tolerance 0.221319 mm
Mesh Quality Plot High
Mesh information - Details
Total Nodes 38667
Total Elements 23608
Maximum Aspect Ratio 8.4088

Figure:

Name	Type	Min	Max
Stress1	VON: von Mises Stress	8.923e+02 N/m^2 Node: 38482	1.681e+07 N/m^2 Node: 30990



Steering Wheel Stress

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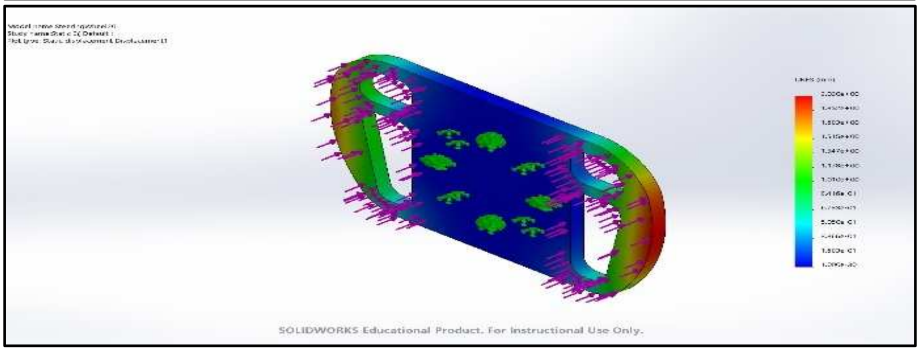
Figure:

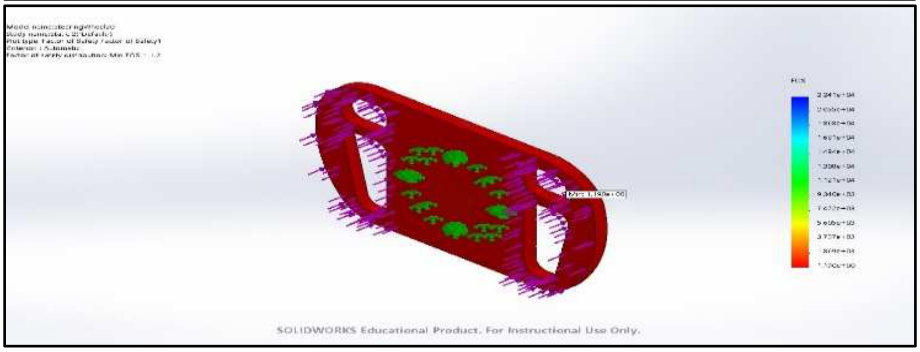
Name	Type	Min	Max
Displacement1	URES: Resultant Displacement	0.000e+00 mm Node: 1	2.020e+00 mm Node: 27107



Steering wheel Displacement

Figure:

Name	Type	Min	Max
Factor of Safety1	Automatic	1.190e+00 Node: 30990	2.241e+04 Node: 38482



Steering Wheel F.O.S

Force parallel to axis of steering Wheel.

<u>Study Properties</u>
Analysis type Static
Solver type FFEPlus
Name: Teak Wood
Model type: Linear Elastic Isotropic
Yield strength: 2e+07 N/m^2
Elastic modulus: 3e+09 N/m^2
Poisson's ratio: 0.29
Mass density: 159.99 kg/m^3
Shear modulus: 3e+08 N/m^2
<u>Fixture Details</u>

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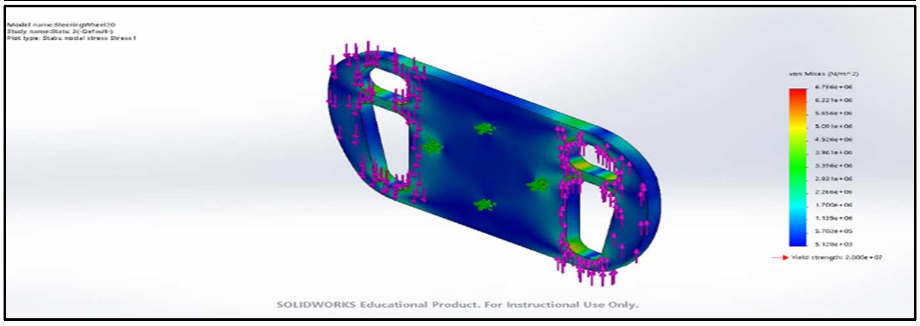
Entities: 4 face(s)
Type: Fixed Geometry
Load Details
Entities: 1 face(s)
Type: Apply force 1
Values: 300 N
Entities: 1 face(s)
Type: Apply force 2
Values: 300 N
Mesh type Solid Mesh
Mesher Used: Standard mesh
Jacobian points 4 Points
Element Size 4.11021 mm
Tolerance 0.20551 mm
Mesh Quality Plot High
Mesh information - Details
Total Nodes 44960
Total Elements 27700
Maximum Aspect Ratio 4.045

Figure:

Name	Type	Min	Max
Stress1	VON: von Mises Stress	5.128e+03 N/m^2 Node: 29573	6.786e+06 N/m^2 Node: 44845



Steering Wheel Stress

Figure:

Name	Type	Min	Max
Displacement1	URES: Resultant Displacement	0.000e+00 mm Node: 1	2.107e-01 mm Node: 1388

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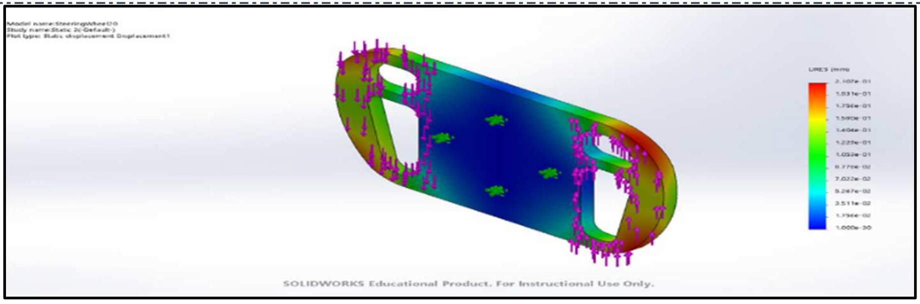
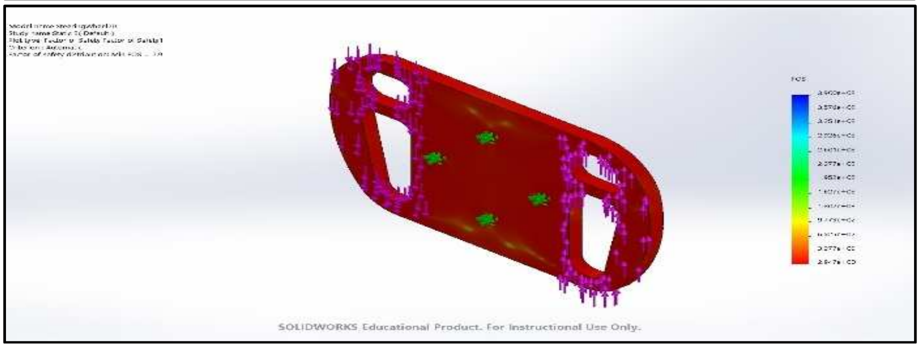


Steering Wheel Displacement

Figure:

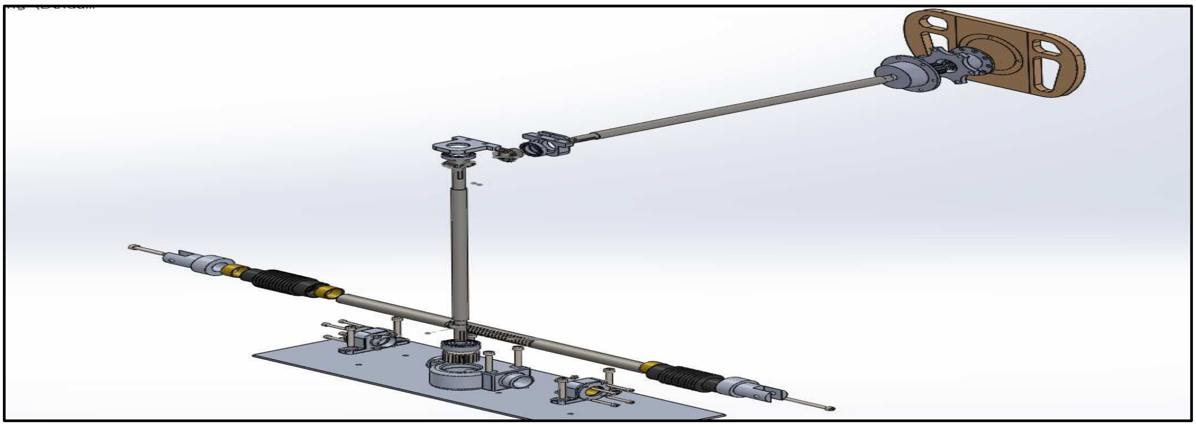
Name	Type	Min	Max
Factor of Safety1	Automatic	2.947e+00 Node: 44845	3.900e+03 Node: 29573



Steering Wheel F.O.S

5. STEERING SYSTEM CAD

Figure:



Complete Steering Assembly

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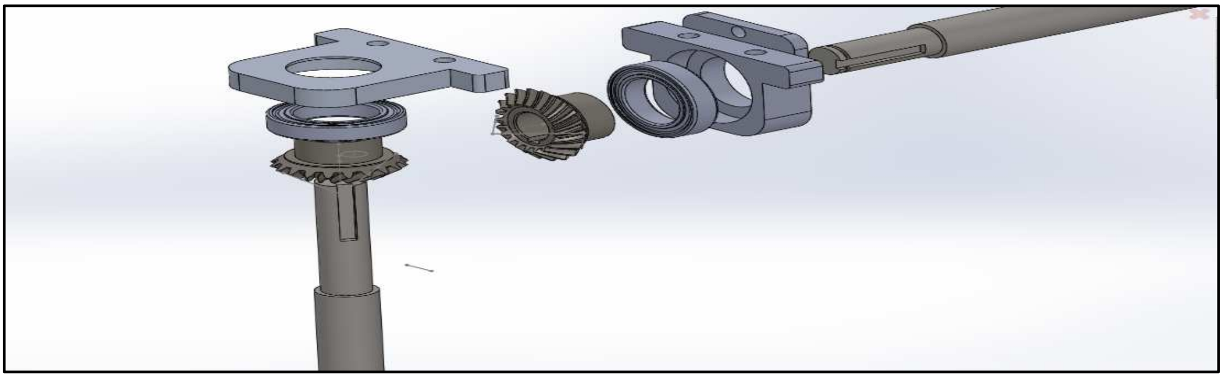
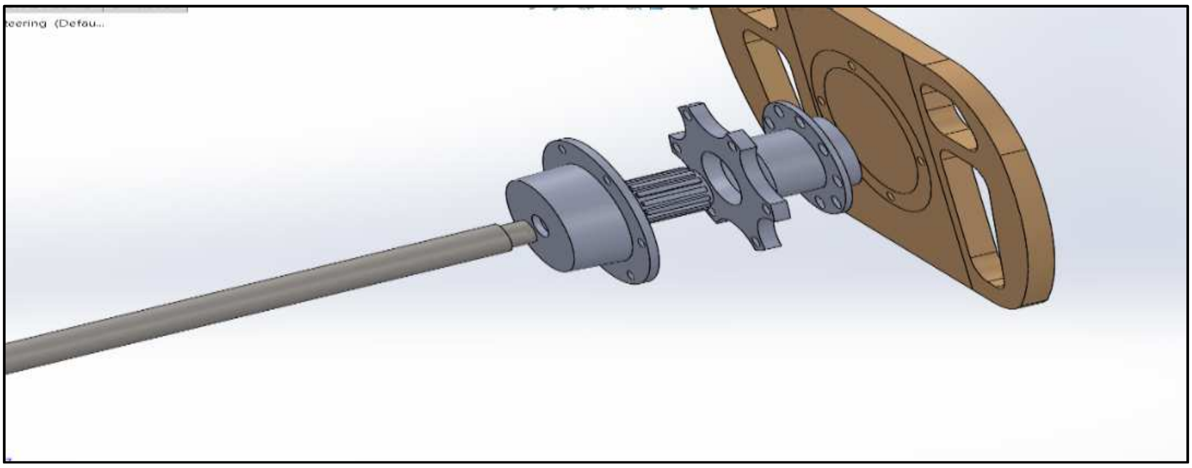
Figure:



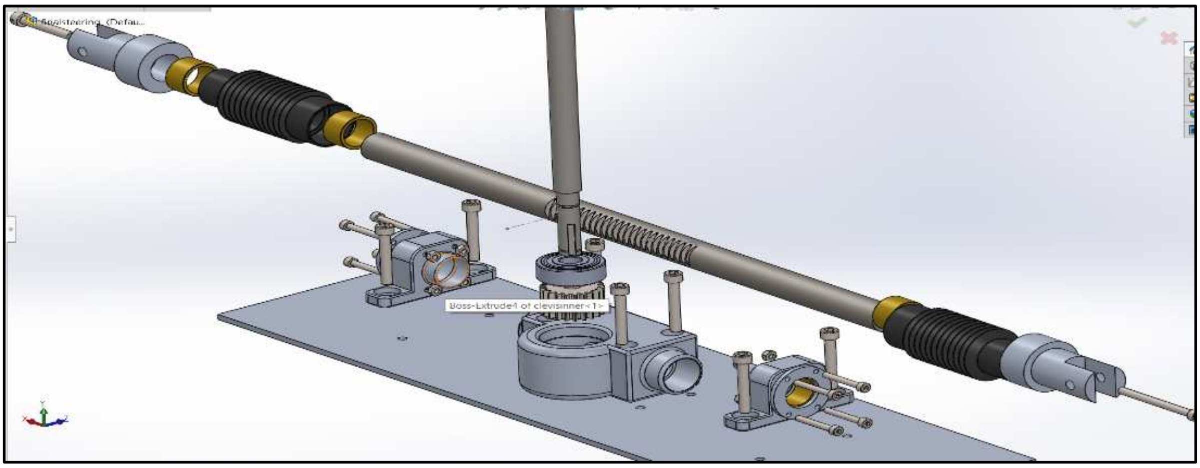
Bevel Assembly

Figure:



Steering Wheel Assembly with shaft

Figure:



Rack and pinion Assembly

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6. CONCLUSION

The new design for a steering system for car 2019-20 Team Ojaswat was to improve drivability with respect to overall performance and responsiveness with better ergonomics, mass reduction and steering capabilities. All these parameters were achieved by the optimized design of steering system. This Steering system was the most optimized, as through several decisions made by the design team during designing process. Which has made the system more reliable, more responsive, much lighter and compact. The optimized design is a rack and pinion configuration with a set of spiral bevel gears connecting the upper and lower columns. An extensive static force analysis was then conducted on the proposed design to determine proper sizes of components and to ensure that they maintain the minimum safety factor, to make the system safer and reliable.

This project has further helped us to learn -

- Formula racing vehicle –Apart from Steering System and vehicle dynamics, the project has helped us boost our knowledge in the areas of Wet & Dry Powertrain, Chassis design, Electrical systems & Aerodynamics.
- Vehicle dynamics –Basis concepts of vehicle Dynamics and tire dynamics.
- CAD software like Solid works & Fusion –A good practice with CAD features like industrial drawings, weldments, sheet metals, surface modelling, and many more.
- CAE software like Ansys and Solid works 3D simulation have been used.
- Manufacturing techniques like welding, profiling, etc. – Within this course of 4 years, great manufacturing skills such as TIG, MIG & Arc welding profiling, cutting, grinding, drilling, and many more were performed and learned.
- Since these FSAE competitions (SAE Supra, FMAE FFS & FIA Formula Bharat) take place on National & International levels, it gives an exposure to interact with great teams, expert judges like Pat Clarke & Claude Rouelle and famous industrialists.

FUTURE SCOPES: The new design for a steering system for car 2019-20 Team Ojaswat was to improve drivability with respect to overall performance and responsiveness with better ergonomics, mass reduction and steering capabilities.

7. ACKNOWLEDGEMENT

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Conflict of Interest: The authors declare that there is no conflict of interest to disclose.

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