Design and Manufacturing of Steering System of Formula Student Car

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Abstract- *Formula Bharat is a student engineering competition held in India.The competition consists of several tasks to be performed by the vehicle which includes design evaluation, acceleration test, Autocross, Skid-pad, Endurance, Fuel Economy, Cost Presentation and Business Plan Presentation. The steering is mainly designed for hairpin turns in race course track, which is the sharpest turn of 3.5m. the objective of research was to minimize steering effort and turning radius of the vehicle. These objectives were achieved by getting optimum angles of tire from Ackerman geometry in consideration of rulebook. The crucial part in designing of steering system is force analysis and calculations. Turning radius, gear ratio, steering torque, are completely dependent on lateral force. The role of suspension ICR geometry and various angles like caster camber KPI helps you understand dynamics of vehicle. CAD and CAE software's like Solid Works, ANSYS, Lotus-shark are some supportive design tools to design and simulate virtually. The sitting build, test and race with their formula style racing car, position and steering wheel position must be designed and positioned according to ergonomics of driver.*

Keywords- Rack and Pinion, Steering, Ackerman, Calculations of steering system, analysis of steering system, components.

I. INTRODUCTION

In this paper we have presented the design of a steering system for formula student race car. In order to design a good race car, the conditions to which the car will be exposed has to be taken into account. At the same time the steering system is able to withstand the external requirements, the design of it has to be compliant with the rules and the design of other parts of the car.

II. OBJECTIVES OF DESIGNING

1. Steering system can have maximum of 4.5m turning radius.
2. Minimize steering ratio.
3. Decrease steering effort.
4. Generate force feedback.
5. Generate high lateral G for comfortable handling of vehicle in cornering without losing grip.

III. SET-UP OF STEERING SYSTEM

- **LOCATION OF RACK AND PINION**:

The first step is to think where to place the rack-pinion and the advantages of locating it above or under the driver's legs. The conclusion was that locating it over the driver's legs would increase the height of the center of gravity, besides it would be more difficult to fulfill 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 location of the rack and pinion.

- **CONNECTION OF THE STEERING WHEEL TOTHE RACK AND PINION**:

In order to make a decision about this, it is necessary to have some dimensions of the car and calculate which would be the angle of steering wheel. This is important because ergonomics is a fundamental part of the car and not taking it into account would be a big mistake.

ALTERNATIVES FOR COMPONENTS:

	Option 1	Option 2	Option 3
Location of the rack-pinion	Over the legs	Under the legs	
Connection steering wheel to rack and pinion	Direct	Universal joints	Gearbox
Shaft section	Round	Square	Hexagonal
Shaft material	Steel	Aluminum	Carbon fiber
Rack-pinion	Commercial	Homemade	

IV. LITERATURE REVIEW

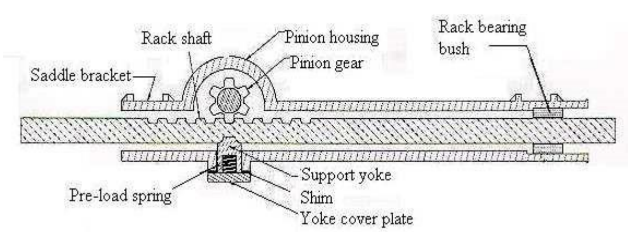
MANUAL STEERING SYSTEM:

a) The manual steering system incorporates:

1. Steering wheel and column.
2. A manual gear box and pitman arm or a rack and pinion assembly.
3. Linkages, steering knuckles and ball joints.
4. Wheel spindle assemblies.

b) Rack and Pinion:

A typical rack and pinion steering gear assembly consists of a pinion shaft and bearing assembly, rack gear, gear housing, two tie rod assemblies, an adjuster assembly, dust boots and boot clamps, and grommet mountings and bolts. When the steering wheel is turned, this manual movement is relayed to the steering shaft and shaft joint, and then to the pinion shaft. Since the pinion teeth mesh with the teeth on the rack gear, the rotary motion is changed to transverse movement of the rack gear. The tie rods and tie rod end then transmit this movement to the steering knuckles and wheels.



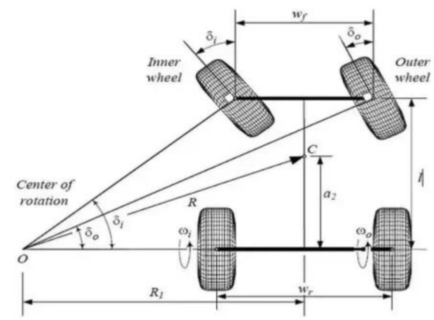
V. GEOMETRIC PARAMETERS

ACKERMAN CONDITIONS:

Ackerman steering geometry is used to change the dynamic toe setting, by increasing front wheel toe out as the car is turned into the corner. Racers are interested because of the potential to influence the handling of the car on corner entry and mid corner.

The typical steering system, in a road or race car, has tie-rod linkages and steering arms that form an approximate parallelogram, which skews to one side as the wheels turn. If the steering arms are parallel, then both wheels are steered to the same angle. If the steering arms are angled, as shown in the figure below, this is known as Ackerman geometry. The inside wheel is steered to a greater angle then the outside wheel, allowing the inside wheel to steer a tighter radius. The steering arm angles as drawn show 100% Ackerman.

When a car goes around a corner, it turns around a point along the line of its rear axle, which means the two front wheels will have to turn through slightly different angles so that they are also guiding the vehicle round this point, and not fighting the turn by scrubbing. Ackerman geometry results when the steering is done behind the front axle and the steering arms point toward the center of the rear axle as seen on Figure below.



CAMBER ANGLE:

Camber angle is regarded as the inclination of the wheel plane to the vertical Negative camber inclines the top of the tire toward the centerline of the vehicle as seen in and positive camber inclines the top of the tire away from the centerline.

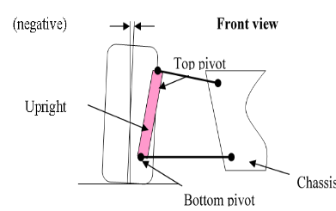


Fig 2.9: Camber angle

A small amount of negative camber of up to 1.5 degrees it is recommended to induce camber thrust. However, changes in camber should be kept at minimum during chassis roll in order to reduce the loss of camber thrust and the change in wheel track load distribution during cornering.

Rate of camber change:

The rate of camber change is the change of camber angle per unit vertical displacement of the wheel center relative to sprung mass.

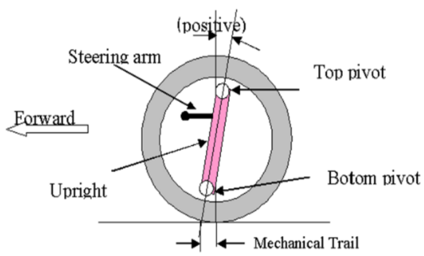
CASTER ANGLE:

Caster angle is the angle in side elevation between the steering axis and the vertical. It isconsidered positive when the steering axis isinclined rearward (in the upright direction) andnegative when the steering axis is inclined forward. Caster angle can be visualized by below figure. Positive caster induces a self-correcting force that provides straight line stability but increases steering effort.

Casterranges from approximately 2 degrees in racing vehicles up to 7 degrees.

Rate of caster change:

The rate of caster change is regarded as thechange in caster angle per unit vertical displacement of the wheel center relative to the sprung mass.



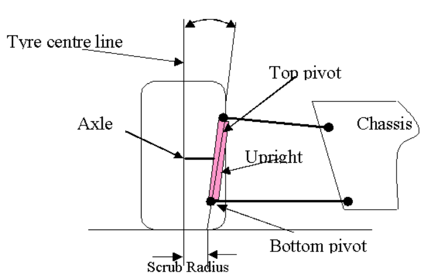
KINGPIN INCLINATION:

The angle in front elevation between the steering axis and the vertical is regarded as kingpin inclination. It is also known as steering axis inclination (SAI) and can be seen in.

It is used to reduce the distance measured at the ground between steering axis and tyre'scentre of pressure in order to reduce the torque about the steering axis during forward motion. A right kingpin inclination will reduce the steering effort and will provide the driver with a good 'road feel".

Kingpin offset measured at the ground is the horizontal distance in front elevation between the point where the steering axis intersects the ground and the center of tire contact.

Kingpin offset it is also known as scrub radius. It is positive when the center of tire contact is outboard of the steering axis intersection point on the ground. Kingpin offset is usually measured at static conditions (zero-degree camber). The kingpin offset at the wheel center is the horizontal distance in front elevation from the wheel center to the steering axis.



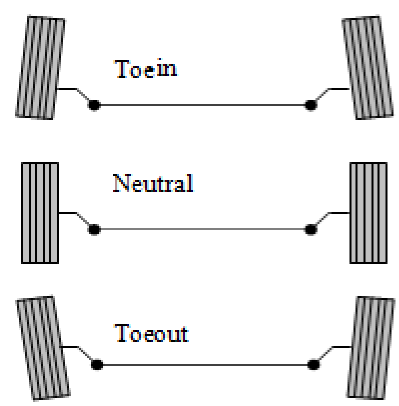
STATIC TOE ANGLE:

Static toe angle is measured in degrees and is the angle between a longitudinal axis of the vehicleand the line of intersection of the wheel plane and the road surfaces. The wheel is "toed-in" if the forward position of the wheel is turned toward a central longitudinal axis of the vehicle, and "towed-out" if turned away.

STATIC TOE:

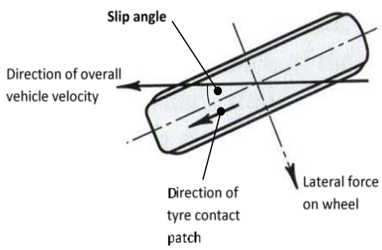
Static toe-in or toe-out of a pair of wheels is measured in millimeters and represents the difference in the transverse distance between the wheel planes taken at the extreme rear and front points of the tire treads. When the distance at the rear is greater, the wheel is "toed-in" by this amount; and where smaller, the wheels are "toed-out" as illustrated in.

It is necessary to set the static toe such way to prevent the tires to become toe out during maximum bump and roll in order to prevent the outboard tire to steer the vehicle to the outside of the turn when cornering. Toe-in produces a constant lateral force inward toward the vehicle centerline during forward motion that will enhance the straight-line stability.



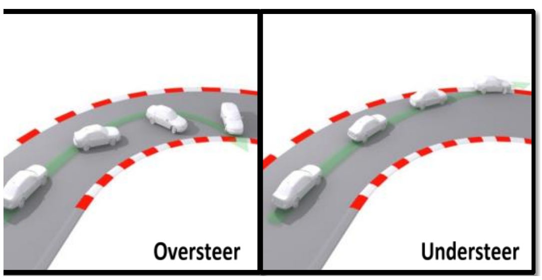
SLIP ANGLE:

Slip angle is the angle made by the direction of the tire contact patch with the direction of overall velocity of vehicle. This principle is best demonstrated by observing below which also highlights the lateral forces imposed on the wheel as it corners.



UNDERSTEER AND OVERSTEER:

When the front wheel slip angles are smaller than the rear ones and for understeer, when the front wheel slip angles were larger than the rear. oversteer is where the car loses grip at the rear wheels causing the car to turn more than expected by the driver whereas understeer induces the opposite of this behavior. These concepts can be pictured above figure. It should also be mentioned that when the slip angles for the front and rear wheels are equal, then the vehicle is said to be neutral steering.



JACKING

The tire reaction forces generated when the vehicle is accelerated during cornering are transmitted to the vehicle through the suspension links. In suspension that place the roll center above the ground, the upward tire reaction force generated by the outside tire is greater than the downward tire reaction force generated by the inside tire. Summing these forces, the resultant will be positive upward acting through its roll center. This upward jacking force lifts or "Jacks" the sprung mass upward when cornering.

BUMP-STEER:

If the vehicle experience bumps on the track, the wheels may have the tendency to steer themselves without the driver doing any changes to the steering wheel. This is undesirable and known as bump-steer. The wheels will change between toe out and toe in as the suspension compress and decompress during the bump. The steering wheel must be moved constantly to keep the vehicle in a constant turn. The wheel will also tend to toe out in a sharp turn as some of the weight is distributed to the outer wheel and hence makes the suspension on the outer wheel to compress. Bump-steer will also cause increase tire wear.

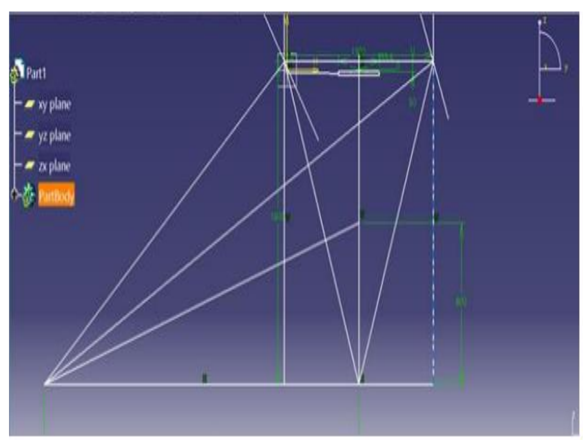
CALCULATIONS

ACKERMAN CALCULATION:

$(1/\tan \theta o) – (1/\tan \theta i) = B/L$
Where,
θo = turn angle of the wheel on the outside of the turn
θi = turn angle of the wheel on the inside of the turn
B = distance between center of pivot
L = wheel base
b = distance from rear axle to center of mass
From the general equation we can calculate the turn angle of the wheel on the outside of the turn for a given inside wheel angle as follows:

Steering Angle Calculation:

L = 1565mm
B = 1220.7 mm
$\theta i = 36°$
$(1/\tan \theta o) – (1/\tan 37.304) = 1300/1600$
So, $\theta o = 25.17$
R=Turning Radius= 2.82 m



RACK TRAVEL:

In order to achieve the true Ackermann Condition iterations were carried out. The varied parameters and their effects on Steering geometry are given below:

In order to achieve the steering ratio in Between 4 to as to get the ICR of front Wheels exactly intersect on the rear axle axis. Ackermann angle must be varied to satisfy the Ackermann condition. Now we can calculate angle in

whichSteering wheel required to turn to turn wheelby inner angle from steady position knowing
the rack travel.

RACK TRAVEL	ANGLE (steering wheel rotation)
2*Π*r	360°
40	Θ

Where
r = radius of pinion.
Cross multiplying and solving we get,
θ=160 degree

Now knowing the angle in which steering wheel required to turn in order to turn wheel by inner angle from steady position;

PARAMETERS	OUTPUT
STEERING RATIO	4.289:1
TURNING RADIUS	2812.66mm
RACK TRAVEL	40mm

We have calculated steering ratio as follows,
Steering ratio= θ /inner angle
=160/37.304

Steering Ratio=4.289 degree

In order to satisfy the all above parameters but to avoid bump steer, **length of tie rod**must be varied by iteration we achieved following outputs:

FORCE CALCULATION

FORCE REQUIRED TO TURN WHEELS:

The frictional force caused by the contact between the ground and tire is transmitted from the tire, through the steering arm, to the tie rod, all the way to the rack and pinion and finally to the steering wheel where the driver has to overcome this frictional force so as to make the wheels turn. The force that is transmitted is destructive to the mechanical components and can cause failure. To prevent failure, the force is calculated and the components are designed to withstand such forces.

Forces on the wheel
F_r = friction force
mg = weight
F_L = lateral force applied from the steering wheel
N = normal reaction

Adding up the horizontal forces:

$$\Sigma F_x = 0 \quad F_L - F_r = 0$$
$$F_L = F_r \qquad (1)$$

Adding up the vertical forces:

$$\Sigma F_y = 0 \quad N - mg = 0$$
$$N = mg \qquad (2)$$

Summing up the moments about the centre of
the wheel:

$$\Sigma My = 0$$
$$(F_L \times R_s) - (2 \times F_r \times r) = 0 \qquad (3)$$

Since this project is limited to the steering system and the other parts of the car are not designed, the weight of the car is assumed. A typical Formula 1 student race car usually weighs 300 Kgs and the driver can weighs about 80 Kgs. Therefore, the total mass is

(300 + 80) Kgs =380 Kgs.

To calculate the weight distribution on each tyre, the weight ratio for front to back was taken to be 50:50. That means that the front tyres takes only 50% of the total weight of the car. Therefore the mass on the front tyres is:

380 × 50/100= 190 Kgs.

The mass exerted on one tyres will be half of the 114 Kgs which is 95 Kgs.

Hence the weight will be:

95× 9.81 =931.95 N
F_r may be found using the followingformula:
$F_r = \mu \times N$ Where: μ = friction coefficient
N = mg

The friction coefficient will be of a higher value in order to establish a safety coefficient. So we take μ = 1.

Now we calculate the friction force:

$F_r = \mu \times N$
= 1 × 931.95N
= 931.95N

And from equation (1) Fr = F_L
Thus:

F_L = 931.95N

This is the force that the rack has to transmit to the tie rods and these to the steering arms to move the wheel. According to the conditions given, this will be the minimum force required to cause a turn of the wheels. But since the friction coefficient was rather large, we can assume that this is the force that will be applied by the driver during racing.

TORQUE ON PINION:

Now we can calculate the torque on the pinion. To calculate the torque we use the following equation:

T = F × rpinion

In our case we have a pinion with a diameter
Of 28mm so:
T = 931.95 × 28
= 26094.6 Nmm
= 26.09 Nm

This is the amount of torque required on the steering wheel to turn the pinion.

Finally the tangential force needed on the steering wheel by the driver to turn the wheels is calculated as below:
T = F × Rsteering wheel
Where: Rsteering wheel = 114.3 mm
Therefore: F = T / Rsteering wheel
F = 26.09 ÷0.1143 m
F = 228.25 N

This is the highest possible value that can be used to turn the wheels. This is because we took a rather high value of the friction coefficient as a safety factor. The most probable case is that the force to turn the steering wheel will be of a much lower value.

DESIGN OF THE PINION REQUIRED FOR DISPLACEMENT OF RACK:

As total rack travel for 360° rotation of pinion is 90mm.Circumference of pinion is 90mm.

Therefore,
2*π*r=90
r=14mm

For the design of pinion, we use following gear data:

For 20˚full depth involute teeth system.

Input shaft speed =18 rpm.
The pinion is heat treated at 340 BHN.
The material used is 20MnCr5, the Ultimate tensile stress=570N/mm^2.

Service factor (Cs)=maximum torque /Rated torque.
According to the assumption,
maximum torque=7000N-mm.
column=4220.7N-mm.
Cs=7000/4220.7=1.7

Since our pinion is commercially cut gear using cutter with velocity greater than 10m/s,
According to the calculation of
Cv=3/(3+v)

According to the calculation of column the power required to turn the pinion shaft is 8.84watt, if we consider the efficiency of UV joint to be 90%, therefore power is 7.596watt.

Therefore, torque on the column is 4200.7N-mm.
The Lewis form factor =0.308.
F(s)=1.8 ……consider according to material Pdf
For spur gear assume Zp=17 & Zg=28

For Beam Strength:

Sb=m*b*6b*Y
Assume b=10mm,6b=570/3=190N/mm^2
Therefore

Sb=m*10m*190*0.308=585.2m^2 N……..(1)

For Wear Strength:

Sw=b*Q*dp*K
Q=2Zg/(Zp+Zg) =1.24
K=0.16(340/100) ^2 =1.84
dp=m*Zp=17m

Therefore

Sw=10m*1.24*17m*1.84=389.7m^2 N……(2)

Sb>Sw, criteria for design is wear strength
Sw=Peff *f(s)…Estimation of module using f(s)

According to Barth's equation for initial stage of breaking of gear tooth
Peff=Cs*Pt/Cv
For spur gear v=5m/s…….std assumption
Cv=3/(3+v)=3/(3+5)=0.375
Pt is a tangential force =2Mt/dp=(2*4220.7)/17m

Pt=496.55/m
Therefore Peff=(1.7*496.55)/(0.375*m)=2251.04/m……..(3)
Sw=Peff*f(s)
389.7m^2=(2251.04*1.8)/m
m=2

Gear Dimension :

m=2
b=10m=10*2=20
dp=m*Zp=2*17=34mm
By hunting of tooth concept,
Zpinion=17…. (theoretical value)
But, Zpinion=14…... (practical value)
Length of rack=12 inch.

DESIGN OF STEERING COLUMN:

We had design the column according to driver ergonomics, as driver has to sit in the for long duration, so it should be comfortable and it should follow rules as discussed earlier. From literature review we got some data for steering column.

1) Distance between steering wheel and driver chest should be 30 to 45 cm.
2) Steering column should withstand atleast a 660N force without failing.

Steering Column Stresses:

The steering column is located just after the steering wheel and is used to transmit the force from the steering wheel to the pinion. It undergoes a torsional force and thus a shear stress due to the torsion. Thus, the column had to be designed with adequate material to design against the shear stress.

We can calculate if the material for the steering column is adequate for the torque that is transmitted by the force on the steering wheel.

The steering column support torsion
efforts :
τmax=T*r/J

Where

τmax=Shear stress
T=Torque in steering column
r= radius of steering wheel
J= Inertia of solid column

Therefore;

J=(π/32) *D^4
J=1.570*10^-8m^4
τmax=26.09*0.018/1.570*10^-8m^4
τmax=29.896*10^6Mpa
Design of tie-rod:

From ICR geometry we got tie rod length from rack end to steering arm as 441.62mm, that is eye to eye length of tie rod is 410mm.

We have used 4130 chromoly hollow pipe for tie rods.
Length of tie rod=410mm
Diameter of tie rod=16mmThickness of tie rod=2mm

UPRIGHT DESIGN OF CASTER AND KPI:

We have simulated the effect of caster and KPI on wheel toe in LOTUS SHARK SUSPENSION ANALYSIS software with suspension team of our project. The values are given below

STATIC VALUES:

CAMBER ANGLE (deg):0.00
TOE ANGLE (SAE) (+ve TOE IN)
(deg): 0.00
TOE ANGLE (PLANE OF WHEEL)
(deg): 0.00
CASTOR ANGLE (deg):2.87
CASTOR TRAIL (HUB TRAIL) (mm):
0.26
CASTOR OFFSET (mm):12.31
KINGPIN ANGLE (deg): 2.89
KINGPIN OFFSET (AT WHEEL)
(mm):48.91
KINGPIN OFFSET (AT GROUND)
(mm): 36.22
MECHANICAL TRAIL (mm): 12.30
ROLL CENTRE HEIGHT (mm):198.58

STEERING PARAMETERS:

Parameters	Future Steering System
Steering Rack Location	101.1mm vertically from ground plane,60mm behind front axles (below driver's legs towards the knees)
Steering Rack length (left tie rod connection to right tie rod connection)	355.6mm
Steering arm 's tie rod connection location	60mm behind the front axle, 60mm from KPI to centre rear axle line.
Static Ackerman	100%
Steering arm length	60mm
Steering ratio	4:2:1
C-factor	90mm
Steering wheel diameter	228.6mm
Max steer angle of inside wheel	37.3
Max steer angle of outside wheel	25.2
Caster angle	3 degrees
KPI	3 degrees

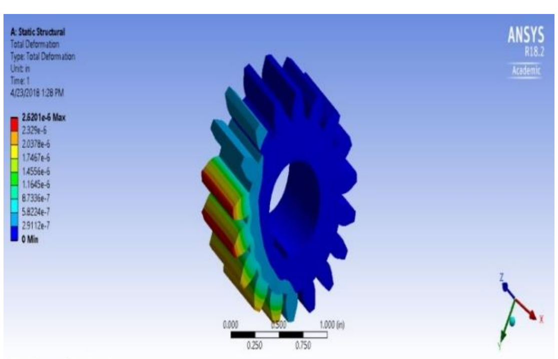
ANALYSIS OFCOMPONENTS

ANALYSIS OF PINION:

For analysis following steps are carried out The material for pinion is selected as 20MnCr5 with yield stress of 570 N/mm^2.

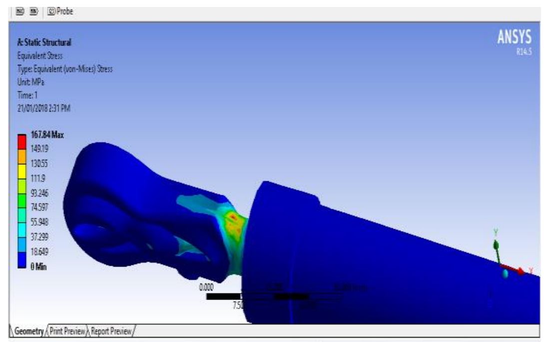
The model is drawn in CATIA software &file saved in IGES format which is then imported in ANSYS software for analysis. The fine meshing is done to have results. Load applied of 2340.8 N on tooth of pinion that is tangential force. The constraint were applied to the steering of pinion.

After applying load and material, the total deformation and equivalent stress are plotted as shown in figure. The value of total deformation comes nearly upto 1.0926mm. since the value of deformation is very less. Therefore material will be safe.the factor of safety is 2.1.



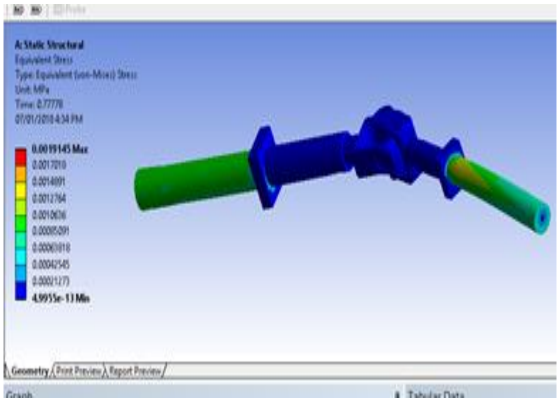
Object Name	Total Deformation	Equivalent Stress
State	Solved	
Scope		
Scoping Method	Geometry Selection	
Geometry	All Bodies	
Definition		
Type	Total Deformation	Equivalent (von-Mises) Stress
By	Time	
Display Time	Last	
Calculate Time History	Yes	
Suppressed	No	
Results		
Minimum	0. in	3.6805e-004 psi
Maximum	2.6201e-006 in	158.47 psi

ANALYSIS OF TIE ROD:



Object Name	Total Deformation		Equivalent Stress
State	Solved		
Scope			
Scoping Method	Geometry Selection		
Geometry	All Bodies		
Definition			
Type	Total Deformation		Equivalent (von-Mises) Stress
By	Time		
Display Time	Last		
Calculate Time History	Yes		
Results			
Minimum	0. in	0. in/in	0. psi
Maximum	9.6959e-003 in	4.6402e-004 in/in	14090 psi

STEERING COLUM ANALYSIS:



RESULTS:

Object Name	Equivalent Stress	Equivalent ElasticStrian	Total Deformation
State	Solved		
Scope			
Scoping Method	Geometry Selection		
Geometry	All Bodies		
Definition			
Type	Total Deformation	Equivalent (von-Mises) Stress	Total Deformation
By	Time		
Display Time	Last		
Calculate Time History	Yes		
Results			
Minimum	7.2454e-011 psi	6.2343e-018 in/in	0. in
Maximum	0.27768 psi	9.3601e-009 in/in	2.0407e-007 in

MATERIAL SELECTION

- **MATERIAL SELECTION FOR RACK AND PINION:**

low carbon steel.

20MnCr5 steel are low alloyed engineering case hardening steel for parts which require core tensile strength of 1000 – 1300 N/mm² and good wearing resistance. It is used in boxes, piston bolts, spindles, camshafts, gears, shafts and other mechanical controlling parts.

Chemical composition% of grade 20MnCr5 steel (1.7147): EN 10084-2008Elements not quoted in this table shall not be intentionally added to the steel without the agreement of the purchaser, other than for the purpose of finishing the heat.
Selected material for tierodand steering column:

Mild steel is used for both tie rod and steering column.

There are various forms of steel in the market, but we have to pick one that is readily available and cheap enough to build the student race car steering system. We found out that mild steel was easily available to us.

Mild steel has the following mechanical properties:

Material Property	Magnitude
Modulus of Elasticity	200GPa
Tensile strength	455MPa
Yield strength(tension)	250MPa
Ductility, percent elongation in 50mm	23
Poisson's ratio	0.29

MATERIAL SELECTION FOR POSITIVE STOPPER:

There are many types of nylon rod, two of the most common rods are nylon 6 & nylon 6.6.The largest application for nylon is for various types of wear components such as gear, cams etc.NYLON is a 30% glass-fiber-reinforced nylon 6 material whose important properties include high tensile and flexural strength, stiffness, excellent heat deflection temperature, and superior abrasion and wear resistance.
Here we have used nylon for two main purpose: -
For positive locking of steering wheel to get desire angle of turning

To reduce wear of sleeves and get efficient rack displacement.

Nylon offers superior wear resistance, low coefficient of friction & noise abatement etc.

Diameter of nylon =1.2 inch
Length =12 inch
Internal hole diameter =29mm.
Color of nylon=natural
Shape= rod
Weight = 0.49.

FABRICATION

MANUFACTURING OF BUSHES, MOUNTINGS OF RACK AND PINION:

We have manufactured tie rod bushes with following steps:

a) Cutting of solid bar with hacksaw
b) Measuring the Bush length and diameter
c) Facing and turning of bush
d)Measuring final length and diameter
e) 6mm Tapping

b)Manufacturing of Tie Rods:

For manufacturing of tie rod, we have to first cut the defined length of tie rod and then weld the bushes to hollow tie rods.

c) Manufacturing of Column:

While manufacturing column we made prototype of steering column to get perfect lengths and angle of universal joint.

TESTING, EVALUATIONANDMODIFICATION

- **TESTING OF STEERING:**

Serial Number	Dynamic Test	Skid Pad	Auto Cross
1	Speed	20Kmph	20Kmph
2	Turning Radius	3.5	3.3
3	Time	20sec	180sec

- **TESTING OF TURNING RADIUS:**

Serial Number	Speed (Kmph)	Calculated turning radius of outer wheel (meter)	Actual turning radius of outer wheel (meter)
1	20	4.2	4.2
2	30	4	4
3	40	3.6	3.8
4	50	3.3	3.5

COST ESTIMATION

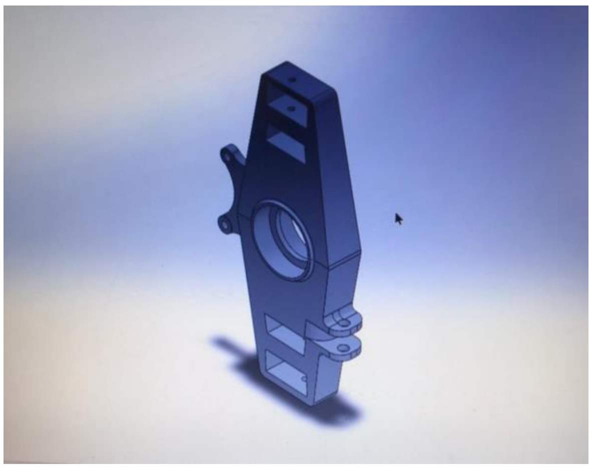
SR.NO	PART LIST	COST (Rupees)
1.	STEERING WHEEL & QUICK RELEASE	8,000
2.	STEERING COLUMN	800
3.	UV JOINT	650
4.	STEERING RACK & PINION	13, 000
5.	TIE ROD	1,200
6.	ROD ENDS	1,800
7.	NYLON BUSHES	150
8.	RUBBER BOOTS	400
9.	NUT &BOILTS	500
10.	RACK MOUNTING MATERIAL	150
11.	COLUMN MOUNTING MATERIAL	200
12	Miscellaneous	2,000
13	Total	28,850

VI. CONCLUSION

1. In this paper of steering system done based on rule book of FB (Formula Bharat) competition organized by mobility consortium engineering in steering system Ackerman steering mechanism is used.
2. Standard procedure is followed to design and manufacturing of steering system. Those components which are OEM part cannot be designed. Proper selections of those components are done systematically. By assembling all designed, manufactured and selected components, it completes our system.
3. Validation is done by different testing on steering system like analysis of rack and pinion and perform different experiments or tests like skid test, autocross, etc. and satisfactory results are obtained. We got our turning radius as 2.8m and steering ratio as 4.2:1. The design is matching with our results. Our design and manufacturing of steering system for student race car is successfully completed

APPENDIX A
STEERING ASSEMBLY IN SOLID WORKS

APPENDIX B
KNUCKLE ARM DESIGN IN SOLID WORKS



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