Design of Steering System for a Formulae Student Car for On-Road Racing

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Abstract

In this exertion efforts have been made to summarize the Steering system of a student formula vehicle is discussed in detail along with calculations. Detail analysis of the components are also attested in this report which gives the behavior of various components under certain working conditions. In practical view the fabrication of these systems in the vehicle is done with low cost by following certain rules and regulations. Objective is to structure a directing framework for the 2017 FSAE vehicle to address all the deficiencies of past plans, yet hold the ease, manufacturability, and low weight. Here efforts are concentrated to choose such a rack and pinion arrangement which will produce a greater lateral displacement of the wheels with the same turn of the steering wheel. As the focused has been made on the FSAE cars, the aim can be achieved by varying the length of the steering column which is adequate for these kinds of cars. The other start point is that the rack and the pinion must be the same modulus and the same material. A suitable material for these elements is SAE 1045 steel which is easy to mechanize.

1. Introduction

Security is a prevalent issue today; hence, a lot of research concerns wellbeing issues. Wellbeing in autos can be isolated into two classifications, inactive and dynamic security. Latent wellbeing alludes to capacities that help relieve the seriousness of mishaps when, for example, safety belts, airbag and so forth. Dynamic wellbeing highlights allude to capacities that help the driver to keep away from a mishap, for example, electronically monitored slowing mechanisms, footing control [1], and dynamic yaw control. Dynamic controlling, which is depicted by P. Kohn [2, 3]. At the point when actualized in the vehicle, the framework doesn't influence dynamic security however could be utilized for dynamic yaw control. Research concerning dynamic yaw control using the force controlling framework has been done by J Ackermann et al., [4–5]. Cutting edge dynamic security may likewise include the controlling framework in directing the driver out of a wellbeing basic circumstance, for example, Lane Keeping Aid, LKA. LKA frameworks help the driver keep the horizontal situation of the vehicle, along these lines decreasing the hazard for street flight mishaps; this can be contrasted with the ACC framework, which is a longitudinal control. The LKA framework has been explored by various specialists and with various incitation. Franke et al. empower the framework by adding an adjustment to the driver's info controlling, [6]; though Pohl and Ekmark added a directing torque to the guiding wheel, in this way empowering a haptic correspondence with the driver, [7]. Before the presence of the vehicle the utilization of the rack and pinion was constrained uniquely to little vehicles in light of the fact that the controlling demonstrated excessively overwhelming and the upgrades were not adequately reasonable so it was important to make a ton of turns with the directing wheel so as to manage the wheels on the attractive direction. Nowadays this problem has been solved with the power-assisted steering. Actually, the rack and pinion are very used by the fact that is cheap and the assembly is simply allowing the incorporation system that help on the driving as the power assisted steering.

But in our case of a formula S.A.E. Regularly the rack and pinion are picked for the most guiding boxes and for the specific since it is light, essentially and modest. It additionally gives a great deal of data to the pilot about the track due to have not helper systems that limit the sentiment of the pilot. Rack and pinion guiding rigging being minimized and light bundle with kinematically stiffer qualities is generally utilized on traveler vehicle autos. Agreeable execution of the guiding framework is dictated by an acknowledgment test, which checks the composite mistake in the rigging. The acknowledgment test checks the amassed gearbox for its torque qualities as opposed to checking the individual parts. The torque required by the pinion to pivot is the 'Free Pinion Torque' (FPT). Rules for formulae SAE: The steering system has been designed on the rules laid down by the Formulae SAE 2017. For the rules and regulations laid down by the FSAE.

2. Description of the work done

2.1 Calculations

Table 1. Specification of Vehicle and Steering Design

Quantity	Value
Wheelbase	62.5 inches
Trackwidth	52 inches
C.G from ground	14 inches
Inner steer angle { δi }	36
King-pin inclination	5.5^{o}
CASTOR (v)	2^{o}
Camber	-2.6^{o}
Gross vehicle weight	330kg's
Weight distribution	40:60 = 132:198
Steering wheel radius	4.5 inches
Gradient	40^{o}

a. Ackermann expression

cotδo−cotδi = Track width/Wheel Base

- δi - inner steer angle
- δo – outer steer angle
- O=Turning Centre
- cotδo−cot 36^{o} = 52/62.5(inches) =1.3208/1.58 (meters)
- **δo = 24.32^{o}**

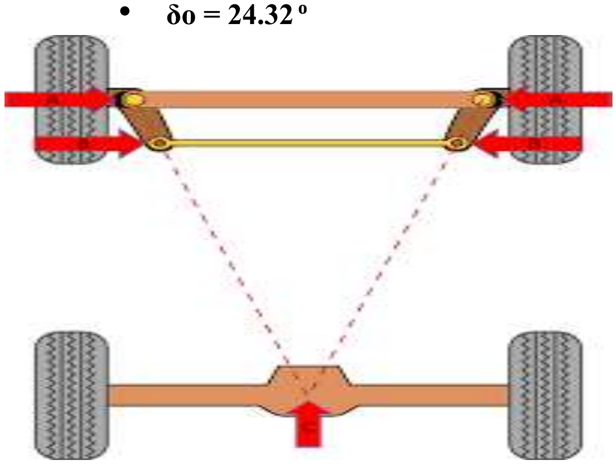


Fig.1 Ackermann Arrangement

b. **Ackermann**

$$= \arcTan\left[\frac{Wheelbase}{\frac{Wheelbase}{Tan\delta o} - Trackfront}\right]$$

=35.986

Ackermann %= (δi/Ackermann) X 100

=36/35.986 X 100

=100.05%

c. **Average Steer Angle**

δmean = (δi + δo)/2

=30.16^{o}

d. Turning Radius

T.R = (T.W/2) + (W.B/sinδ_{mean})

=(1.32/2+1.58/sin 30.16^{o})

=0.6604 + 3.14 = 3.80 meters

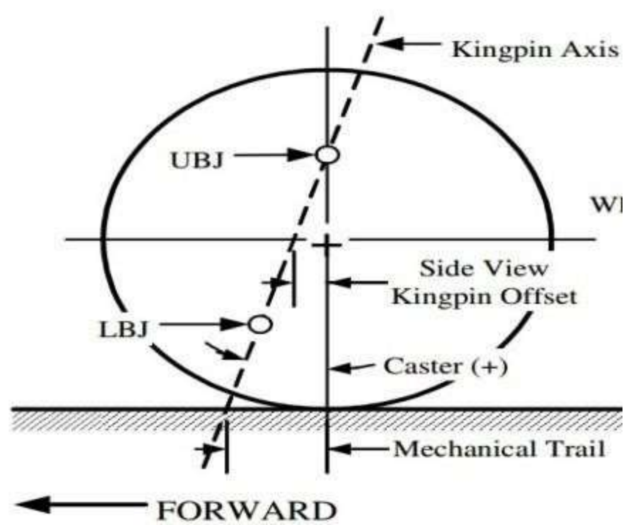


Fig. 2. Schematic diagram showing schematic trail.

e. Mechanical Trail

M_T = Tan v x r

= Tan 2^{o} x 0.254

= 0.0349 x 0.254

= 0.009 meters

f. Diagrammatically calculated as 12 mm = 0.012 m

g. Load transfer during braking (downhill)

W_f = W/L[cos θ x c-sin θ x h+(a_x x h/g)]

Where,

- W_f = Weight Transfer
- G = Gravity
- Wheel base (l)=62.5inches
- Height of centre of gravity (h) = 0.355 meters
- Deceleration (a_x) = 1g m/s^2
- Gradient angle (θ) = 20°
- Total weight (w) = 330 kg
- Distance of C.G. from back(C) = 0.632 meters
- Weight left = Weight right = 170/2 =85 kg.
- W_f= 170 kg

h. Vertical Force

$F_v = W_{FR}$ x g

= 85 x 9.81

= 833.85 N

i. Slip Angle (α) = 1 degree

Cornering stiffness = C_S=0.16/1 °

j. Lateral forces

$F_y=C_s x F_v x α$

=0.16x833.85x1

= 133.416 N

k. Moment due to lateral forces

M_L= (F_{YL}+F_{YR}) x r x Tan [v (caster angle)]

= (133.416+133.416) x 10.5 x 0.0254 x Tan 2°

= 2.485 N-m

l. Moment due to vertical force

M_V= (F_{VL}+F_{VR}) x d x sin $\gimel$ x Sinδ

= (833.85+833.85) x 0.012 x sin 5.5 ° x sin (30.16 °) = 0.9636 N-m

m. Moment due to aligning torque

M_{AT}= (M_{VL}+M_{VR}) x cos√($\gimel^2+v^2$)

= (0.9636+0.9636) x cos√(5.5^2+2^2)

= 1.917 N-m

n. Total steering torque

$M_{AT}+M_V+M_L$ = 5.3656 N-m

3. Rack and pinion calculations

Table 2. Parameters for Rack and Pinion Calculation

QUANTITY	VALUE
Rack Travel	5”
Rack Shaft Length	15”
Module	1.5 mm
Teeth on Pinion	18
Teeth on Rack	29
Diameter On Pinion	27 mm
Turns Lock to Lock	1.6 turns

- Steering Ratio = 288:36 = 8:1

$$\text{Ackermann arm length} = \frac{\text{Racktravel}/2}{\text{Inner steer angle (rad)}}$$

$= \frac{5"/2}{0.6108}$

= 4 "

a. Steering Wheel Torque

The calculations for steering wheel toque are as follows,

Total King Pin Torque = 26.22 N-m

b. **Torque on Pinion gear** = Torque x Gear Ratio

= 26.22 x 0.627

= 16.3 N-m.

c. **Torque on steering Wheel** = 16.3 N-m

d. **Human Effort** = 120 N

e. **Steering wheel radius** = $\frac{\text{Torque on steering Wheel}}{\text{Human Effort}}$

$= \frac{16.3}{120}$

= 0.13 m = 5.1 inch

f. **Steering wheel diameter** = 10 inches.

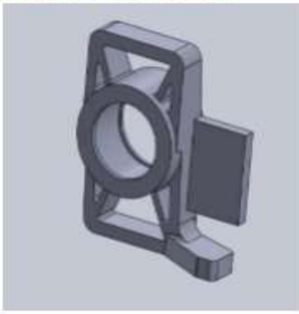
4. FEA (Finite Element Analysis)

4.1 Knuckle / Upright

Controlling Knuckle is one of the basic segments of vehicle which joins suspension, directing framework, wheel center point and brake to the body. It experiences fluctuating burdens exposed to various conditions, while not influencing vehicle directing execution and other wanted vehicle attributes. Taking into consideration static and dynamic load conditions, structural analysis and modal analysis were performed. Finite element model was developed in ANSYS

Front and Rear upright CAD model

FRONT UPRIGHT –



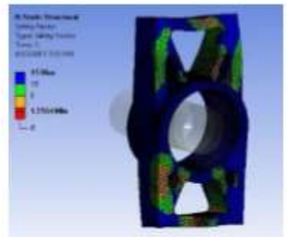
Vertical force = 2500 N

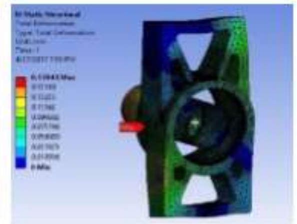
Lateral force = 1500 N

Braking force = 1500 N

Front upright analysis

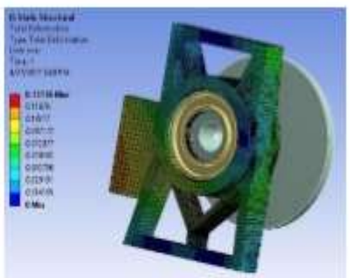
F.O.S minimum: 1.5 **max deform: 0.17mm**





Rear upright analysis

Total deformation: 0.13 mm **F.O.S minimum: 1.4**



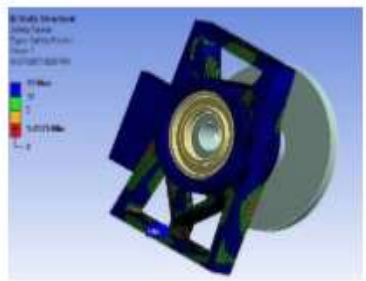
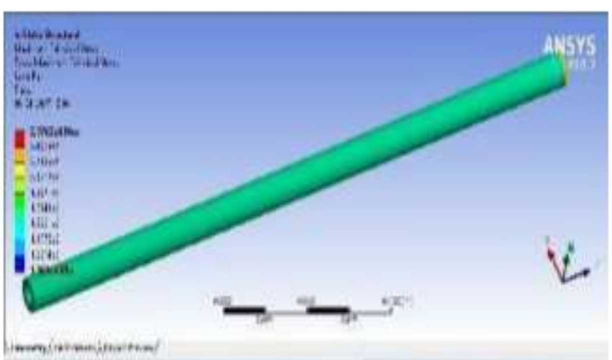


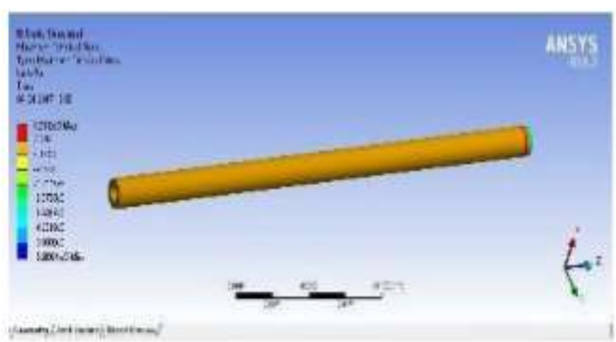
Fig 3. Front and Rear upright analysis

4.2 Tie rod analysis

(a) Tensile Stress



(b) Compressive Stress



(c) Buckling Analysis

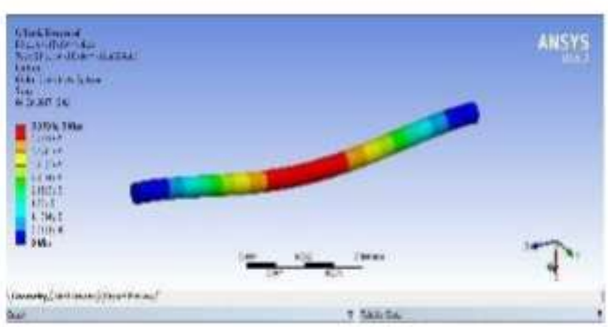


Fig 4. Tie Rod Analysis

Rack and Pinion

The physical rack and pinion steering system is not used in heavy weight vehicles due to high axle loads, although it is simple in design and easy to manufacture, therefore it is commonly used in light weight vehicles. We have designed complete assembly of rack and pinion in SOLIDWORKS 2016, is shown below: Torque at Pinion= 16.30 N-m

Fig 5. Complete assembly of rack and pinion

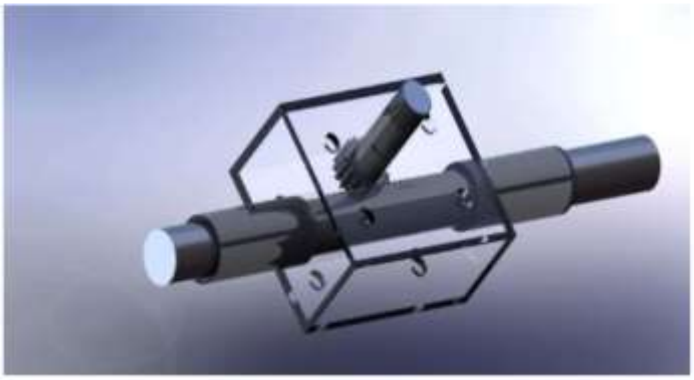


Table 3. Specifications of Rack and Pinion Analysis

Material	EN 08
Max Deformation	0.29 m
Max Total Stress	200 MPa
Min Factor of Safety	2.32

Conclusion

With this little initiation, to increase the output without giving much input, this can prove to be of great help to the FSAE students. This work not only helps in decreasing the steering effort but can also result in weight reduction of the car, which is of prior importance in FSAE. The rack is base mounted to bring down the CG of the rack, pinion, and tie poles. This outcomes in around 3 kg being brought down 400 mm (16 in), contrasted with a top-mounted rack. Thus, the better results can be concluded from this design which provides greater stability and lowering steering time.

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