

Design & Analysis of Steering System for a Formula Student Car

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ABSTRACT: Formula student India is a student based design and fabrication project of a single seater formula styled car. The aim is to design a simple yet effective steering system that reduces driver effort and also adapts to the track conditions offered such that the system does not fail causing harm to both the car and also the driver. A mathematical model is set up followed by geometrical validation and modelling of the entire steering system using CAD software SolidWorks and CATIA V5. Since like any other mechanical system stress is generated in the system, Static analysis is performed using ANSYS Workbench to check the static stress distribution. To ensure the dynamic response of the steering system Multi Body Dynamics (MBD) analysis is performed using Altair Hyperworks.

KEYWORDS: Formula student, steering system, Geometric validation, Multi body dynamics.

I. INTRODUCTION

Formula Student India is a student based design and fabrication project of a single seater formula styled car. The steering system is involved in the control of the car and hence the steering is to be in such a way that the effort by the driver is to be less and should help in easy manoeuvrability. The steering system of this car is a rack and pinion based steering mechanism that converts the rotational motion generated at the steering wheel into a linear motion at the end of the rack. The design is based on the rule book for SAE SUPRA 2017 and FORMULA BHARAT 2018, according to which drive by wire forbidden and hence we have selected a simple rack and pinion system with no additional electrical or hydraulic help. Though the tracks used for such events are flat in nature, one must account for the natural kinematic behaviour of the steering system and hence it is essential to not only factor the static stress but also the dynamic aspects of the steering system.

II. DESIGN PROCEDURE

The steering system is a front wheel based steering unit, as it is in most formula student cars, The design involves formation of a mathematical and geometrical model followed by CAD and FEA procedures. The approach in designing said system involves the following steps,

- I. Identification of the vehicle requirements
- II. Geometrical set up
- III. Geometric validation
- IV. Design of mechanism
- V. Modelling and Analysis by CAD and FEA respectively

I. IDENTIFICATION OF VEHICLE REQUIREMENTS:

The requirements are in accordance with the standard rulebook of SAE SUPRA and FORMULA BHARAT, but are also made sure to be satisfactory to the driver comfort and also to ensure safety to the driver.

The following parameters are set up,

- Wheel Track (Front) = 1150 mm
- Wheel Track (Back) = 1100 mm

- Wheel Base = 1550 mm

II. GEOMETRICAL SET UP:

The basic set up of an entire steering system is based on the Ackerman geometrical calculations. The Ackerman geometry (Fig 1.) is a graphical representation of the steering system.

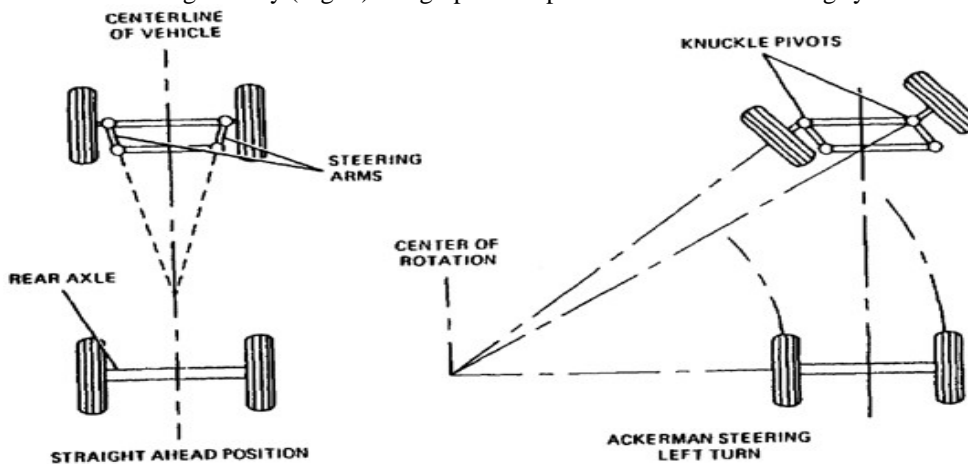


Fig 1. Ackerman steering

It is to be seen that steering and suspension go hand in hand as the entirety of the effort in the suspension hampers the steering too in terms of roll, bump, jounce etc. Here comes the input set up of another set of parameters, the steering angles.

- Caster = +3.5°
- Camber = -1.5°
- Kingpin inclination = 3°

III. GEOMETRIC VALIDATION

A more analytical approach is applied to the validation of the results by using geometric relations, Ackerman (Fig 2.) calculations are performed to find the turn radius as well as the turning angles of the car i.e, the lock angles of the car.

TURNING RADIUS:

$$\begin{aligned} \text{Turning Radius} &= (\text{Track Width}/2) + (\text{Wheel Base}/\text{Average steer rate}) \\ &= (1150/2) + (1550/\sin 30^\circ) \\ \mathbf{R} &= \mathbf{2875 \text{ mm} = 2.875 \text{ m.}} \end{aligned}$$

OUTER ANGLE: (ϕ)

In triangle ABC, angle C = 90 deg.
 $\tan \phi = (1550 / 2875 + 575) = 1550 / 3450$
 $\phi = 24.19^\circ$

$$\begin{aligned} \text{Length of AB} &= \sqrt{AC^2 + BC^2} \\ &= \sqrt{3450^2 + 1550^2} \end{aligned}$$

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AB = 3782mm. (Outer turning radius)

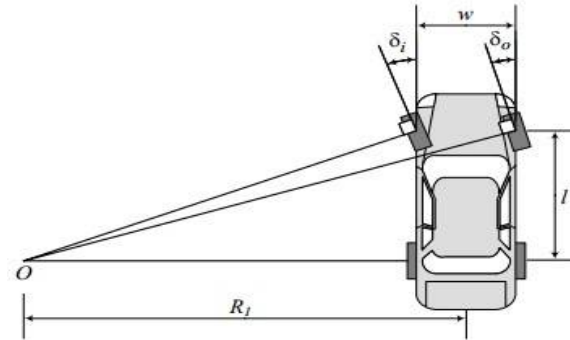


Fig2. Ackerman geometry

INNER ANGLE: Θ

In triangle ADE, angle E = 90 deg.

$$\tan \Theta = (1550 / 2875 - 575)$$

$$\Theta = 33.97^\circ$$

$$\text{Length of DA} = \sqrt{(AE^2 + DE^2)}$$

$$\text{DA} = 2773.5\text{mm (Inner turning radius).}$$

ACKERMANN % (and) ACKERMANN ANGLE (α):

$$\tan \alpha = c / (2b)$$

Where, c - distance between steering arm on left & right uprights

b - wheelbase

$$\alpha = \tan^{-1}(1000 / (2 * 1550))$$

$$\alpha = 17.75^\circ$$

Steering arm length = 55 mm (assumption)

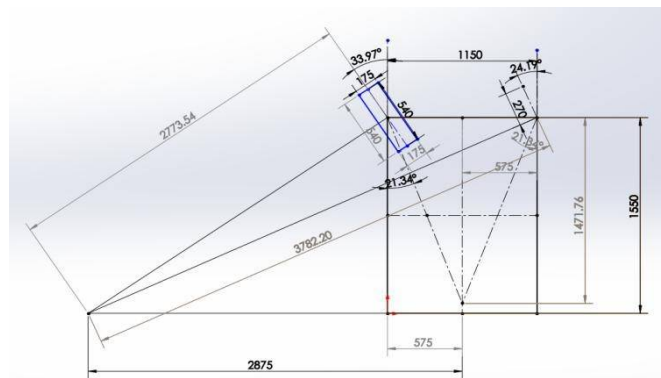


Fig3. Graphical representation

STEERING RATIO:

The steering ratio is the ratio of how much the wheel turns to the amount of travel generated in the rack. Approximating maximum turn to be of 30 degrees and steering wheel movement to be 180 degrees the steering ratio can be calculated as

$$\text{S.R} = 180/30 \Rightarrow \text{S.R} = 6:1$$

IV.DESIGN OF MECHANISM

As mentioned before a simple rack and pinion mechanism is selected for the steering because of its simplicity and reduced space occupancy. Selection of gear tooth profile, as per BIS standards a 20° full depth tooth profile is selected because of its property of generating less interference and a reduced risk of undercutting.

Table1. Gear parameters

| Parameter | value |
|--------------------|---------|
| Pressure angle (φ) | 20° |
| Addendum (ha) | m |
| Dedendum (hd) | 1.25m |
| Clearance (c) | 0.25m |
| Working depth | 2m |
| Whole depth | 2.25m |
| Tooth thickness | 1.5708m |

Where m is the module of the gear

$Z_1= 20$ Teeth (Assuming) Standard module is 1.5mm
 $b=10 \times m=15\text{mm}$

Pitch circle diameter (PCD):

$$d_1 = m \times Z_1$$

$$d_1= 1.5 \times 20 =30$$

$$d_1= 30 \text{ mm}$$

$$V=\pi \times d_1 \times N_1 / 60 = 0.0209 \times m$$

$$=0.0209 \times 1.5=0.032 \text{ m/s}$$

$$h_a=1.5\text{mm}$$

$$h_d=1.25 \times 1.5 =1.875\text{mm}$$

Tooth thickness, $1.5708 \times 1.5=2.3562 \text{ mm}$

Fillet $=0.4 \times 1.5 =0.6 \text{ mm}$

Let us consider the material to be Al 7075 T6

Table2. Material properties

| | |
|--|--------------------|
| Hardness: 1. BHN 2.Knoop 3.Rockwell (A) | 150 191 53.5 |
| Ultimate tensile strength (S_{ut}) | 541 Mpa |
| Yield strength (S_{yt}) | 468 Mpa |
| Modulus of elasticity (E) | 71.7 Gpa |
| Poisson's ratio | 0.33 |
| Melting point temperature | 477 0 |

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Rack and pinion is designed so that aligning torque is generated during cornering without failure and countering pitting failure

BEAM STRENGTH:

Beam strength of gear tooth is the maximum tangential load that gear tooth can take without tooth damage.

Assumptions in analysis of beam strength

- The full load acts at the tip of a single tooth
- The effect of radial force is neglected
- Stress concentration is neglected
- Frictional force due to sliding of teeth are neglected

Analytical calculations

a. bending endurance strength of pinion (σ_{bp}): $(\sigma_{bp}) = (S_{ut})/3 = 541/3 = 180.33$

b. bending endurance strength of gear (σ_{bg}): $(\sigma_{bg}) = (S_{ut})/3 = 541/3 = 180.33$

c. Lewis form factor (Y):

$$1. Y_p = 0.484 - (2.87 / Z_p) = 0.484 - (2.87/20)$$

$$Y_p = 0.3151$$

$$2. Y_g = 0.484 - (2.87 / Z_g) = 0.484 - (2.87/29)$$

$$Y_g = 0.3850$$

$$\sigma_{bp} * Y_p < \sigma_{bg} * Y_g$$

Since the pinion is weaker than gear, we design the pinion for bending.

Assuming $b = 10 * m$,

The beam strength is given by,

$$P_b = \sigma_{bp} * b * m * y$$

$$P_b = 180.33 * 10 * m * m * 0.4172$$

$$P_b = 752.336 \text{ m}^2 \text{ N}$$

WEAR STRENGTH:

The rack and pinion mechanism is also subjected to a certain amount of wear mainly due to its material properties, especially its hardness. Failure due to pitting occurs when the stress between the two meshing gears exceeds the surface endurance limit.

$$P_w = b * Q * dp * k$$

$$Q = \text{Ratio factor for external gear pair} = [(2 * Z_g) / (Z_g + Z_p)]$$

Here $Z_g = 29$, $Z_p = 17$ So that $Q = 1.260$

$$k = [\sigma_c^2 \cdot \cos\phi \cdot \sin\phi \cdot (1/\epsilon_1 + 1/\epsilon_2)] / 1.4$$

$$k = \{[(0.27) \cdot (9.81) \cdot (\text{BHN})]^2 \cdot \cos(20) \cdot \sin(20) \cdot (1/71700)\} / 1.4$$

$$k = 0.45 \cdot (\text{BHN}/100)^2$$

$$k = 0.45 \cdot (150/100)^2 \quad k = 1.0125$$

Hence wear strength can be calculated as follow;

$$P_w = 255.15 m^2 N$$

V.CAD MODELING AND FEA PROCEDURE:

3D CAD Modelling is performed using Solid Works and CATIA V5R20,

The Steering assembly consists of the following components in order,

- Steering column
- Steering rack
- Pinion gear
- Steering Wheel
- Universal Joints

In addition to these are the Tie rods, Steering uprights and Wheel hubs.



Fig 4. Rack and pinion assembly as seen in Solid Works

ANALYSIS OF RACK AND PINION

STATIC ANALYSIS:

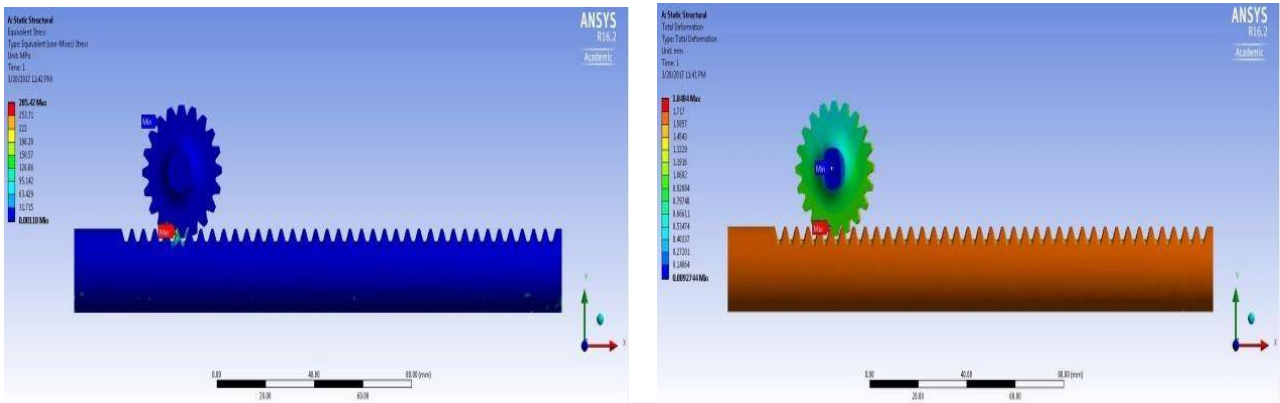


Fig 5. Static analysis results for the rack and pinion assembly

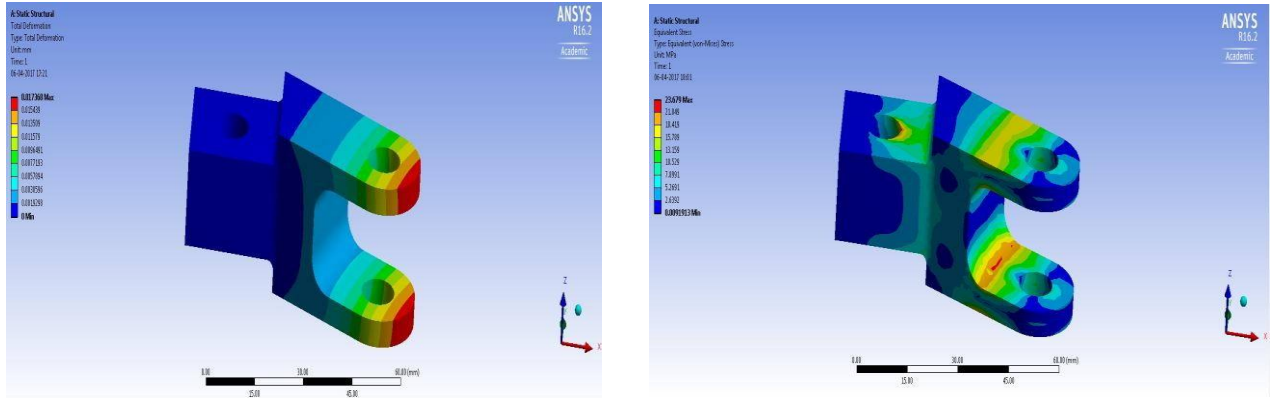


Fig 6. Static analysis of steering arm

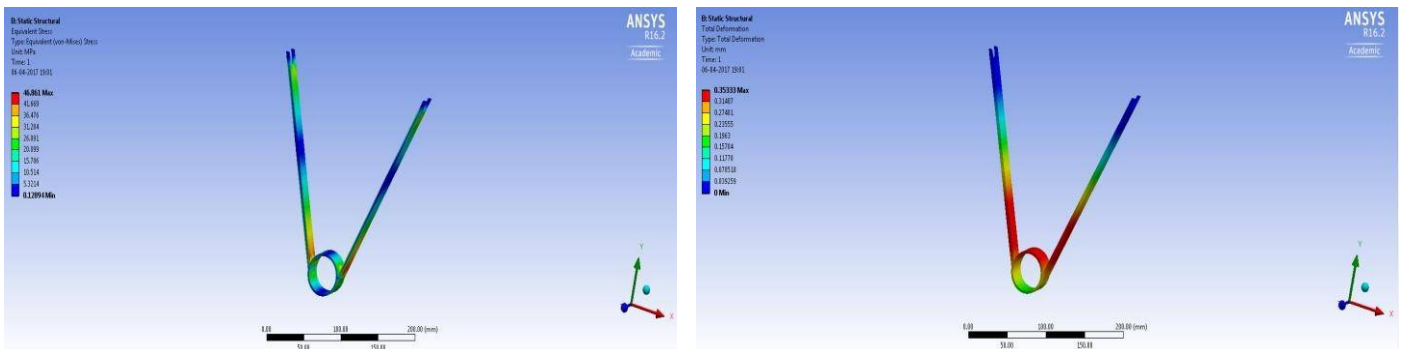


Fig 7. Static analysis of steering column

Table3. FEA results

| Component | Von Mises stress (N/mm ²) | Deformation (mm) | Factor of safety |
|-----------------------|---------------------------------------|------------------|------------------|
| Rack and pinion | 285.42 | 1.848 | 2 |
| Steering arm | 23.679 | 0.017 | 3 |
| Steering column mount | 46.861 | 0.353 | 5 |

The stress distribution and total deformation is generated in the rack and pinion, steering arm, steering column mount in order to check for its safety.

DYNAMIC ANALYSIS:

The dynamic analysis is performed using Altair Hyperworks, and plots are generated with respect to time, as steering systems are not only subjected to static stress but also exhibit a dynamic behaviour.

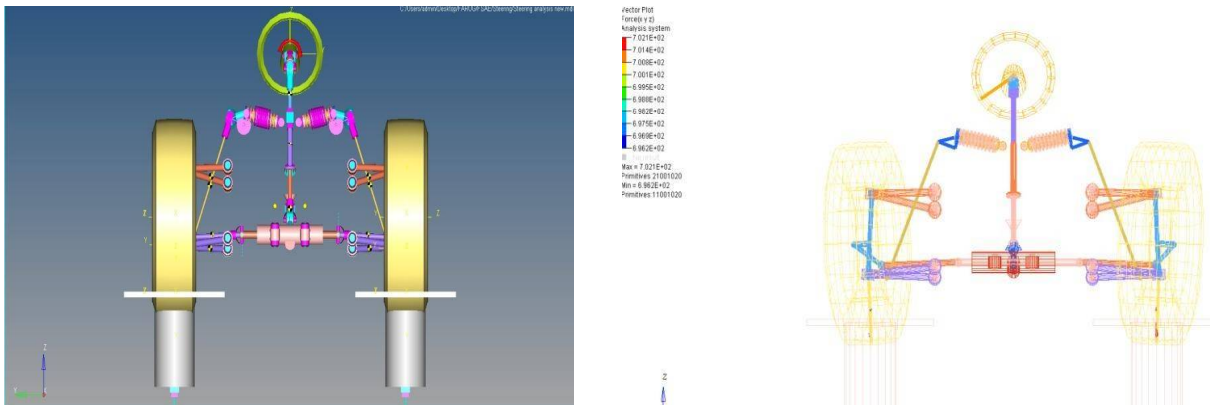
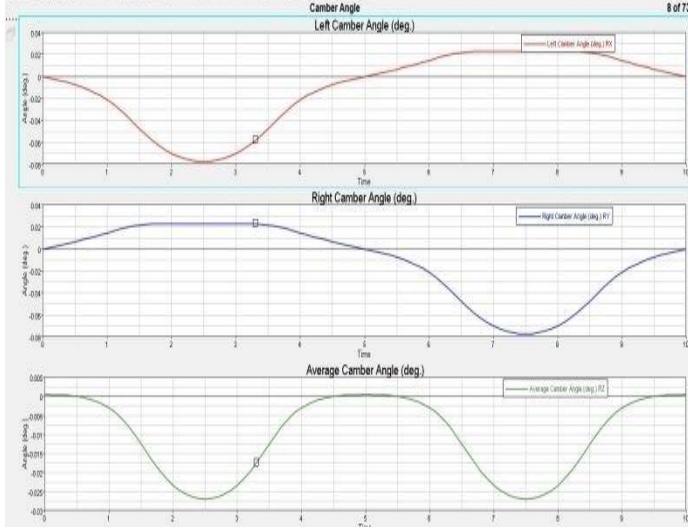
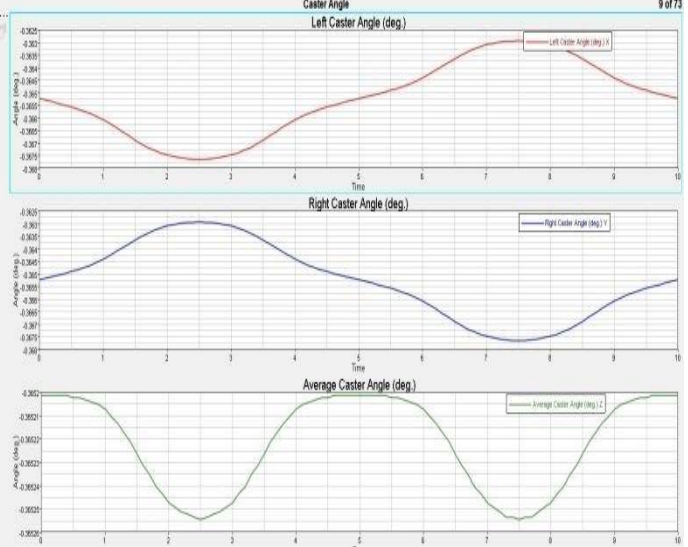
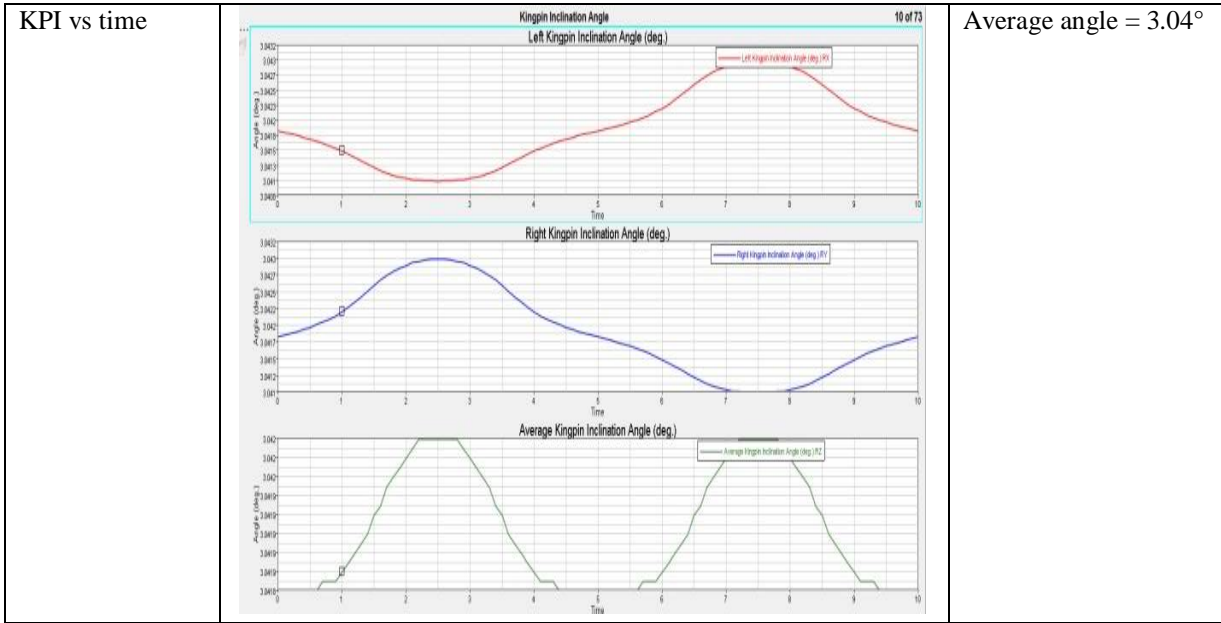


Fig 8. MBD in Altair Hyperworks Motion view

Table 4. Plot results - Multi body dynamics

| Plot | Plot seen in MBD | Inference |
|----------------|--|-------------------------|
| Camber vs time |  | Max deviation = 0.02° |
| Caster vs time |  | Max deviation = 0.3652° |



In addition to the steering angles, there exists the scrub radius that dictates the amount of scrubbing of the wheel on the surface, the amount of scrubbing is directly proportional to the drivers effort, the lesser the scrub radius the lesser the radius, the wheel assembly reveals a scrub radius of **3.55 mm** that is also verified using Motion view in Hyperworks.

VI. CONCLUSION

The observations made in this particular steering system conclude that,

1. The camber change with respect to time is very negligible when compared to other angles.
2. The stress yielded is well under the limit of the yielding point of Al 7075 T6, which means that the design is safe for a much-reduced weight as compared to steel.
3. The steering effort sees minimal change as not much influential change is generated in the steering angles.

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