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Design and Analysis of a Formula Vehicles

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Abstract: This paper provides an overall idea of design and development of suspension, steering, and frame of a FORMULA VEHICLE. The suspension system is designed on LOTUS SHARK software (ADAMS can also be used), it is developed keeping in mind the wheel travel required at the time of cornering and the weight of the vehicle.The design and analysis of the frame are performed on SOLIDWORKS and ANSYS. Some key points for the design of the frame are the strength of the material required and the weight of the vehicle. Steering system taken for the vehicle is RACK and PINION system as it is best suited for the design and manufacturing purpose.

Keywords: formula vehicle, Lotus Shark Software, frame design and analysis, suspension design, Rack and Pinion

I. INTRODUCTION

Frame, Steering, and suspension are three of the most important systems in a vehicle manufacturing. If the steering or suspension system is not designed properly then the vehicle may face major problems like toppling and rolling etc For designing these systems accurately, some computer-aided designing and analysis are required for which software like Solid Works, Lotus Shark, ANSYS etc. are being used. The Frame is of TUBULAR structure made with AISI 1018 steel pipes with a circular cross-section. For the base frame 2mm thickness pipes are used and for the upper body, 1.5mm thickness is sufficient. Double Wishbone Suspension systems are used on both front and rear side majorly because it gives freedom of camber angle variance to the manufacturer. On the front suspension, the spring and damper are mounted on the lower wishbone while it Is on the upper wishbone at the rear side to provide ambient space for the rear axle to be mounted. Rack-n-Pinion steering system with 100% Ackerman Geometry is being designed as it is best suited for the vehicle under the prevailing conditions of weight and manufacturing allowance to be considered.

II. CHASSIS DESIGN AND ANALYSIS

The frame is the basic unit to which various components are attached and body is bolted to the frame. Frame design included the design and modeling of a perfect frame satisfying the dimensions mentioned in rulebook ISIE HVC-2018. CAD modeling of the tubular frame was done in Solidworks and Catia simultaneously and its stability and stiffness were assured.

A. Functions Of Frame

1) To support the chassis components and the body
2) To withstand static and dynamic loads without undue deflection or deformation.

B. Frame Cross-Sections

During movement of a vehicle over normal road surfaces, the chassis frame, is subjected to both bending and torsional distortion as discussed in the previous section. Under such running conditions, the various chassis-member cross-section shapes, which find application, include.

1) Solid round or rectangular cross-sections,
2) Enclosed thin-wall hollow round or rectangular box-sections
3) Hollow sections are designed to produce more resistance to bending than solid section pipes.
4) They have less weight compared to that of members of solid cross sections
5) Therefore, Hollow sections are employed for fabrication of our vehicle.

C. Frame Type(Ladder Frame)

The ladder frame is one of the simplest and oldest of all designs. It consists of two symmetrical beams, rails, or channels running the length of the vehicle, and several transverse cross-members connecting them. This design offers good beam resistance because of its continuous rails from front to rear, but poor resistance to torsion or warping if simple, perpendicular cross-members are used

D. Frame Material

The choice of frame material is a first and most important factor for automotive design. The most important criteria that a material should meet are lightweight, economic, effectiveness, safety, recyclability and life cycle. Considering the above-given parameters we have chosen to use AISI1018 as the desired material for our chassis construction.

Advantages of AISI 1018

1) It provides high surface hardness and a soft core to parts that include worms, dogs, pins, liners, machinery parts, special bolts, ratchets, chain pins, oil tool slips, tie rods, anchor pins, studs etc.
2) It is used to improve drilling, machining, threading and punching processes
3) It is used to prevent cracking in severe bends.
4) AISI 1018 mild/low carbon steel can be instantly welded by all the conventional welding processes.

Chemical Composition

Element	Content
Carbon, C	0.14 - 0.20 %
Iron, Fe	98.81 - 99.26 % (as remainder)
Manganese, Mn	0.60 - 0.90 %
Phosphorous, P	≤ 0.040 %
Sulfur, S	≤ 0.050 %

Physical Properties

Physical Properties	Metric	Imperial
Density	7.87 g/cc	0.284 lb/in^3

Mechanical Properties

Mechanical Properties	Metric	Imperial
Hardness, Brinell	126	126
Hardness, Knoop (Converted from Brinell hardness)	145	145
Hardness, Rockwell B (Converted from Brinell hardness)	71	71
Hardness, Vickers (Converted from Brinell hardness)	131	131
Tensile Strength, Ultimate	440 MPa	63800 psi
Tensile Strength, Yield	370 MPa	53700 psi
Elongation at Break (In 50 mm)	15.0 %	15.0 %
Reduction of Area	40.0 %	40.0 %
Modulus of Elasticity (Typical for steel)	205 GPa	29700 ksi
Bulk Modulus (Typical for steel)	140 GPa	20300 ksi
Poissons Ratio (Typical For Steel)	0.290	0.290
Machinability (Based on AISI 1212 steel. as 100% machinability)	70 %	70 %
Shear Modulus (Typical for steel)	80.0 GPa	11600 ksi

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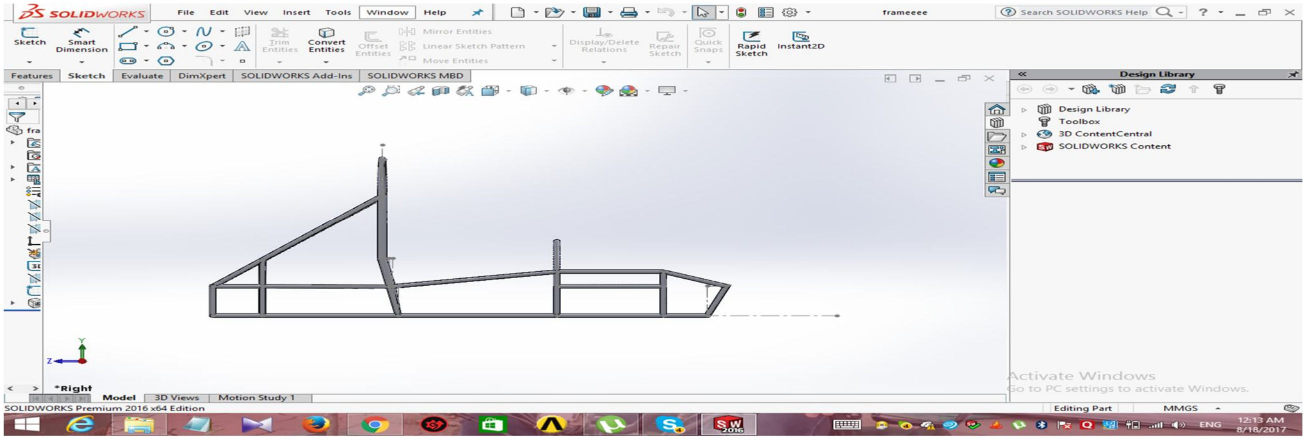
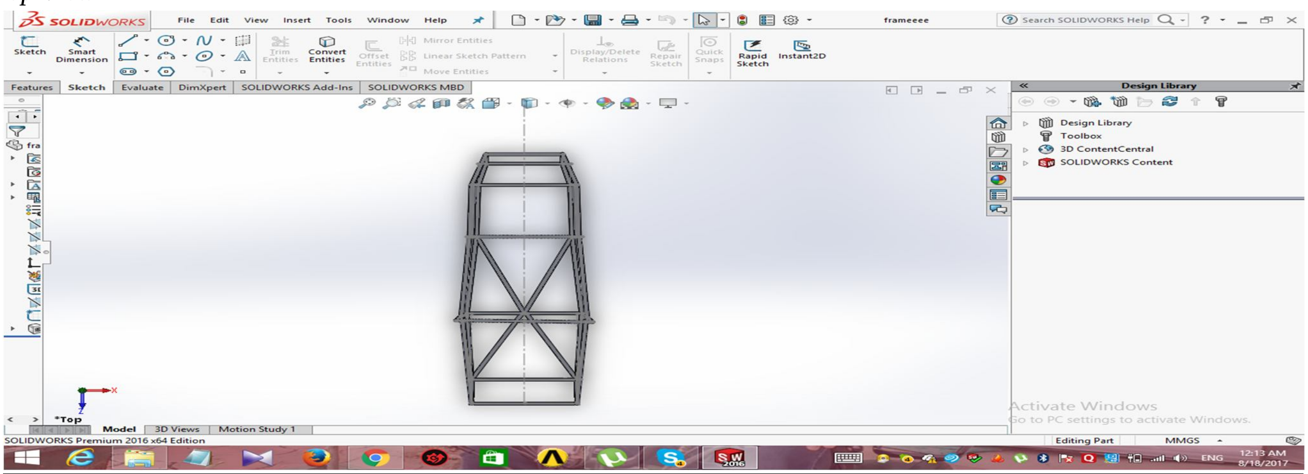
E. Cad Model Of Frame

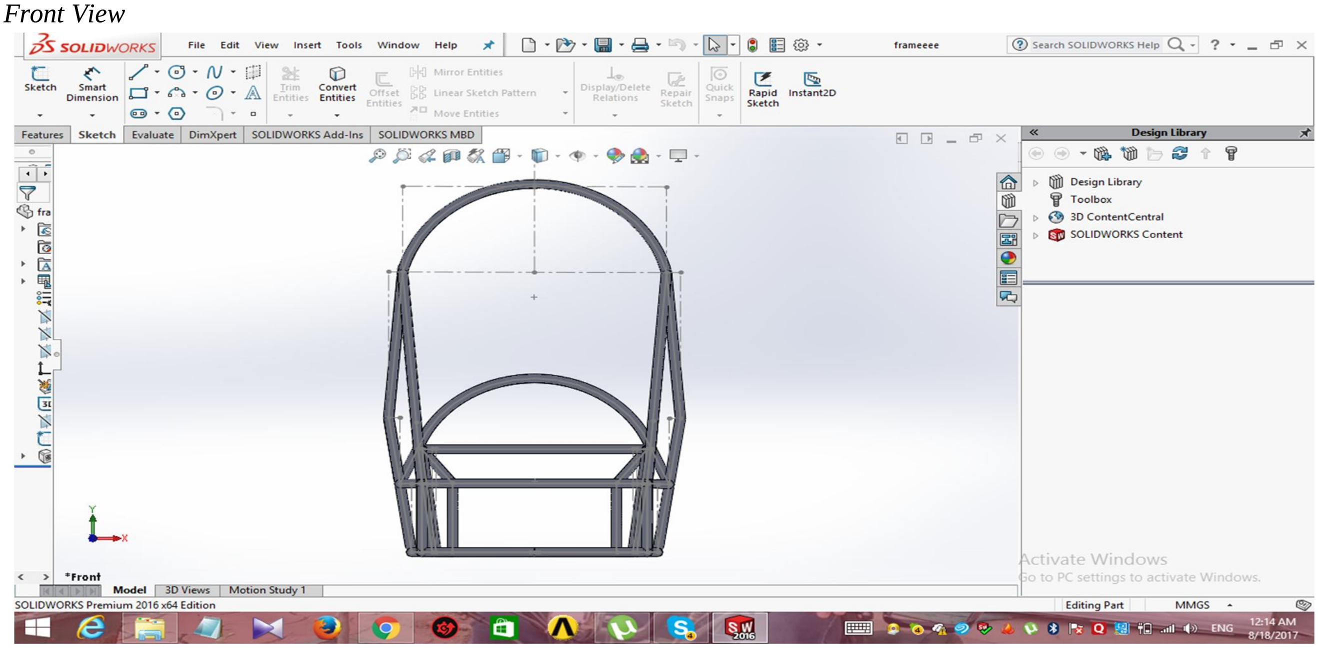
1) Side View



2) Top View



3) Front View



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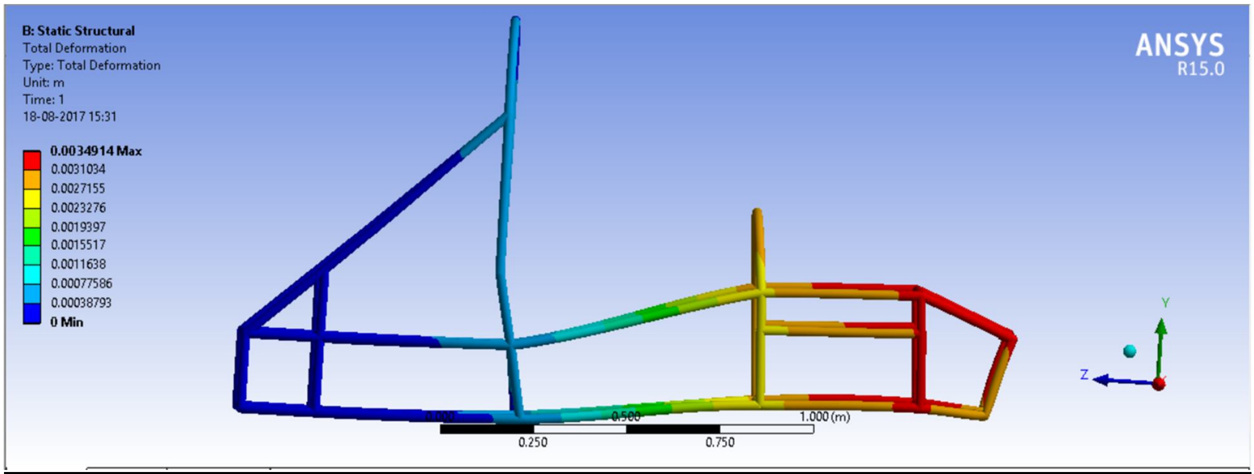
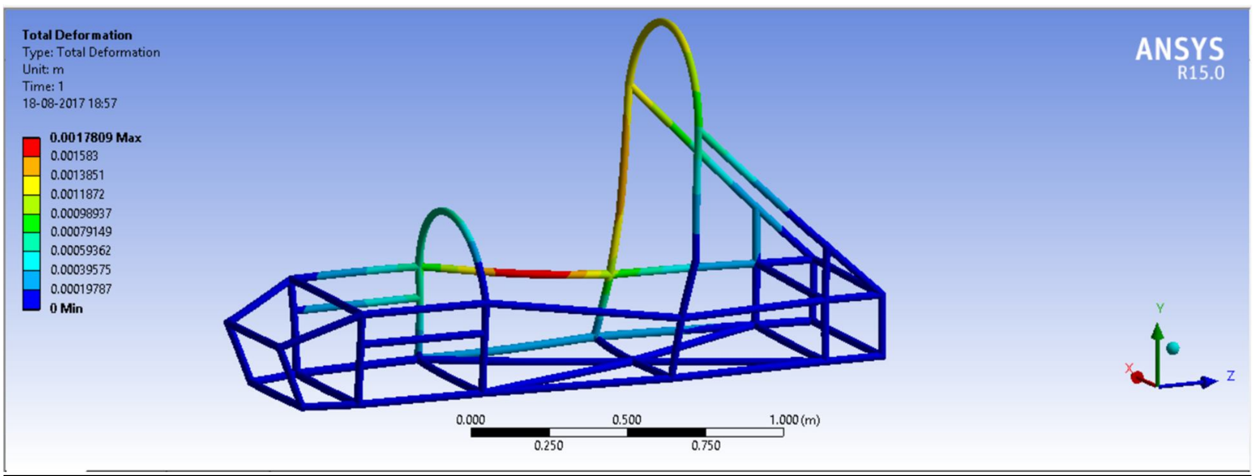
III. IMPACT ANALYSIS OF FRAME

A. Impact analysis Front



B. Side impact analysis



IV. SUSPENSION

A. Introduction

The suspension system is the term given to the system of springs, shock absorbers and linkages that connect a vehicle to its wheels. When a tire hits an obstruction, there is a reaction force and the suspension system tries to reduce this force. The size of this reaction force depends on the unsprung mass at each wheel assembly. In general, the larger the ratio of sprung weight to unsprung weight, the less the body and vehicle occupants are affected by bumps, dips, and other surface imperfections such as small bridges. A large sprung weight to unsprung weight ratio can also impact vehicle control. The main role of the suspension system is as follows:

a) It supports the weight of the vehicle
b) Provides a smoother ride for passengers.
c) Protects the vehicle from damage
d) Keeps the wheels firmly pressed to the ground for better traction.
e) It isolates the vehicle from road shocks.

There are three basic components in any suspension system:

f) Springs
g) Dampers
h) Anti-sway bars

The following types of suspension systems are generally available in the market: 1. Mechanical Suspension System:

1) Independent Suspension

a) Leaf spring Suspension

b) MacPherson Suspension

c) Wishbone Suspension

2) Dependent Suspension

a) Rigid Axle Suspension

b) Electric Suspension System

c) Magnetic Suspension System

B. Selection Of Suitable Suspension System

The selection of the suspension system which will best satisfy the requirements of formula hybrid vehicle was carried out. Out of the many available suspension systems in the market, the Double Wishbone Suspension System was selected for the formula hybrid vehicle. This selection was done based on the following basic parameters:

1) Load bearing capacity

2) Flexibility

3) Cost

4) Technical aspects: Camber, Stiffness, Rolling

5) Availability of parts and components

C. Design

The design procedure for the chosen suspension system is divided into two stages:

1) *Primary design:* Basic design and development of Suspension System and components. Modified design parameters based on an approximation of Dynamic Conditions. Static testing and analysis

2) *Secondary design:* Mathematical modeling of finalized hybrid vehicle Dynamic testing and analysis Modification of Design Parameter based on Dynamic Testing results.

The following components are to be designed:

a) Wishbone

b) Knuckle

The design procedure for the components of the Suspension system is dependent on the suspension geometry (as shown in Fig.1 and Fig.2); found out by taking into considerations the design constraints.

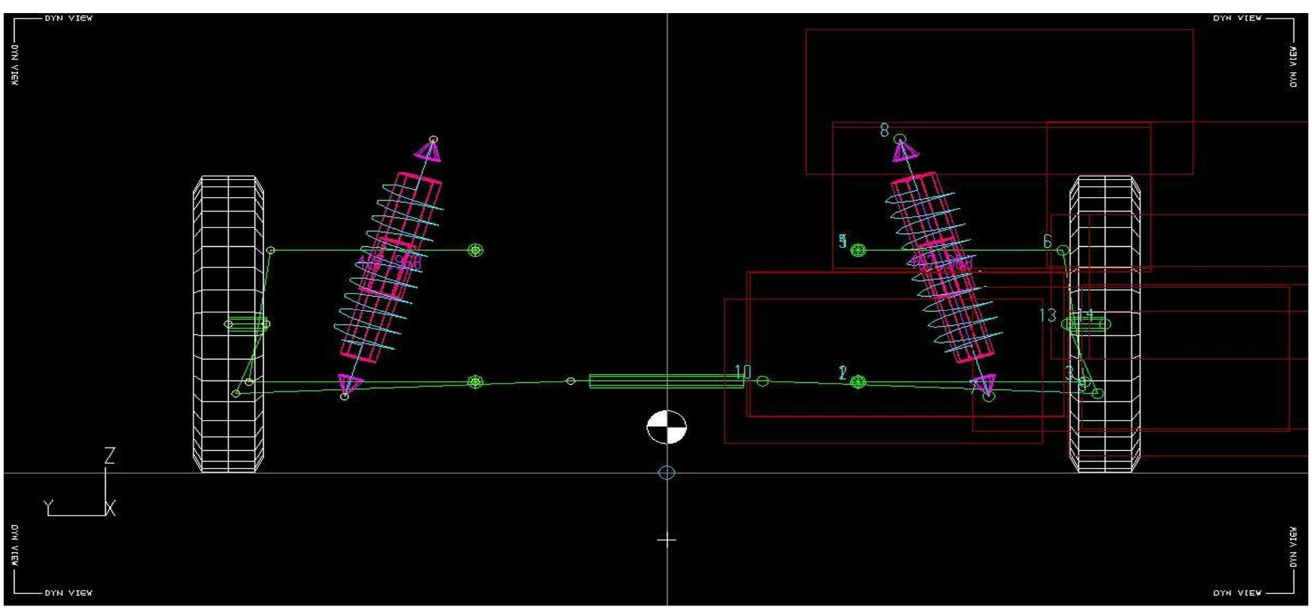


Fig.1 Front suspension geometry

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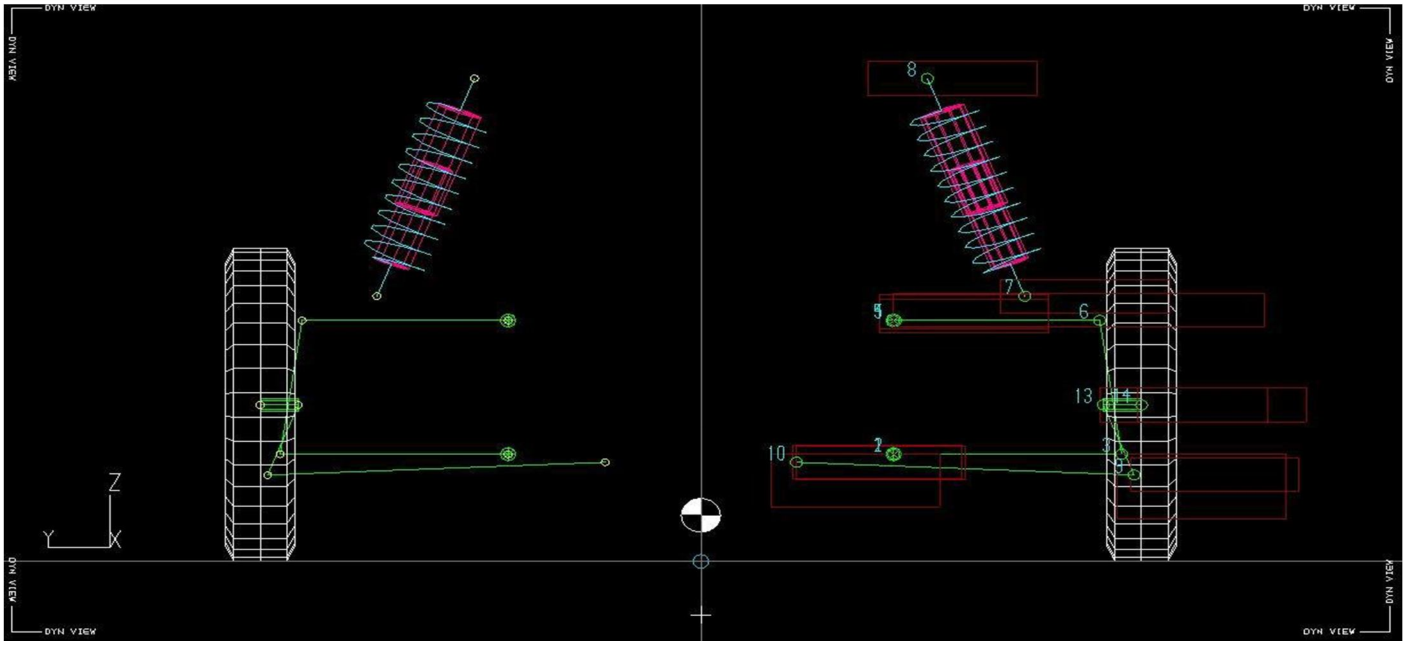


Fig.2 Rear suspension geometry

Assumptions made:

1) Factor for static to the dynamic condition:3
2) According to the mass distribution of 60:40 (Rear: Front)
3) Mass per wheel (Front) = 42 kg
4) Mass per wheel (Rear) = 63 kg

D. Front suspension

The angle of inclination of the strut = 60 (from horizontal)
Point of attachment of strut = 10” (254 cm) from chassis end (from suspension geometry) Reaction force acting from the ground on the wheel = (Mass per wheel * 9.81) N = (42 kg * 9.81) N = 412.02 N

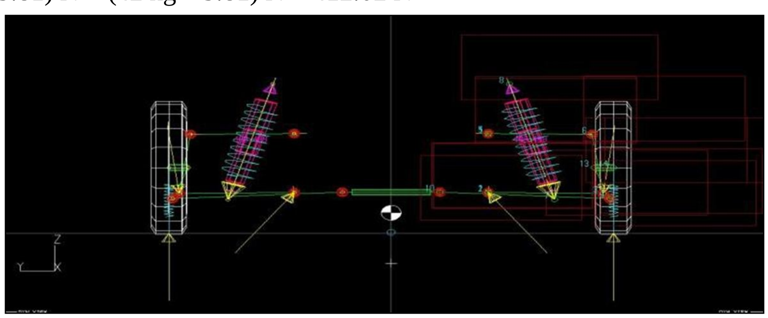


Fig.3. Forces on front

Horizontal distance of reaction force from hinge point = 31.8cm ….from suspension geometry Horizontal distance of strut attachment point from hinge point = 26.9cm
By taking moment about hing points, 412.02*31.8= Spring Force * 26.9
Spring Force = 487.07N
Considering the dynamic factor, the Dynamic force acting on the spring = 1461.21 N
According to the ride conditions and road quality for a hybrid vehicle, it is concluded that the optimum spring travel should be approx. 3
Hence, Required Spring Stiffness =Dynamic Spring Force/Spring Deflection = 1461.21/62.5 = 23.37 N /mm Angle Correction Factor(ACF) = cosine(16.367)=0.96

Motion ratio = (26.9/31.8)^2=0.71 Wheel
Rate = 11.30
Suspension frequency = 187.8*((11.30/42)^0.5)
= 97.45/60 = 1.624 Hertz

E. Mounting Points

X (mm)	Y (mm)	Z (mm)		
3819.00	-247.00	238.91	POINT:1	Lower wishbone front pivot
4179.00	-247.00	238.91	POINT:2	Lower wishbone rear pivot
4092.00	-539.34	238.91	POINT:3	Lower wishbone outer ball joint
4092.50	-247.00	438.91	POINT:4	Upper wishbone front pivot
4332.00	-247.00	438.91	POINT:5	Upper wishbone rear pivot
4092.50	-511.34	438.91	POINT:6	Upper wishbone outer ball joint
4146.50	-415.84	216.91	POINT:7	Damper wishbone end
4180.00	-300.88	606.91	POINT:8	Damper body end
4214.50	-556.34	221.31	POINT:9	Outer track rod ball joint
4245.50	-123.84	240.31	POINT:10	Inner track rod ball joint
4092.50	-516.84	326.41	POINT:11	Wheel spindle point
4092.50	-565.84	326.41	POINT:12	Wheel centre point
4030.00	-255.84	208.31	POINT:13	Part 1 C of G
4170.00	-335.84	463.31	POINT:14	Part 2 C of G
4230.00	-340.84	233.31	POINT:15	Part 3 C of G
4130.00	-535.84	288.31	POINT:16	Part 4 C of G

Camber Angle (deg):	0.00
Toe Angle {Plane} (deg):	0.00
Toe Angle {SAE} (deg):	0.00
Castor Angle (deg):	0.14
Castor Trail (hub) (mm):	-0.28
Castor Offset (grnd) (mm):	0.84
Kingpin Angle (deg):	7.97
Kingpin Offset (w/c) (mm):	38.75
Kingpin Offset (grnd) (mm):	7.25
Mechanical Trail (grnd) (mm):	0.84
ROLL CENTRE HEIGHT (mm):	0.22

LHS WHEEL (-ve Y)
TYPE 1 Double Wishbone, a damper to lower wishbone
INCREMENTAL GEOMETRY VALUES

Bump Travel (mm)	Camber Angle (deg)	Toe Angle {SAE} (deg)	Castor Angle (deg)	Kingpin Angle (deg)	Damper1 Ratio (-)	Spring1 Ratio (-)
60.00	-0.1866	-2.1169	0.1433	8.1566	1.742	1.742
40.00	-0.0836	-1.2104	0.1433	8.0520	1.790	1.790
20.00	-0.0216	-0.5071	0.1432	7.9902	1.838	1.838
0.00	0.0000	0.0000	0.1432	7.9696	1.884	1.884
-20.00	-0.0003	0.0489	0.1433	7.9699	1.893	1.893
-40.00	-0.0851	0.4234	0.1433	8.0560	1.971	1.971
-60.00	-0.2000	0.3224	0.1433	8.1706	2.012	2.012

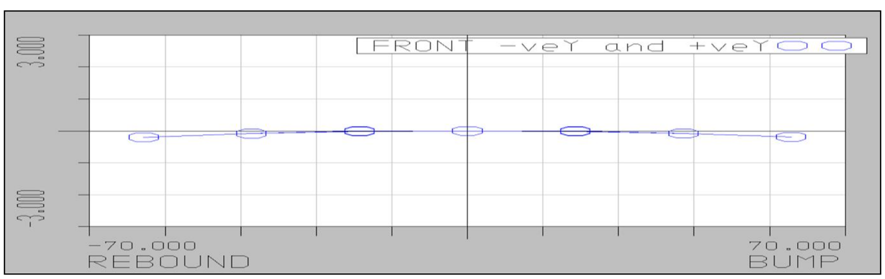
RHS WHEEL (+ve Y)

TYPE 1 Double Wishbone, a damper to lower wishbone

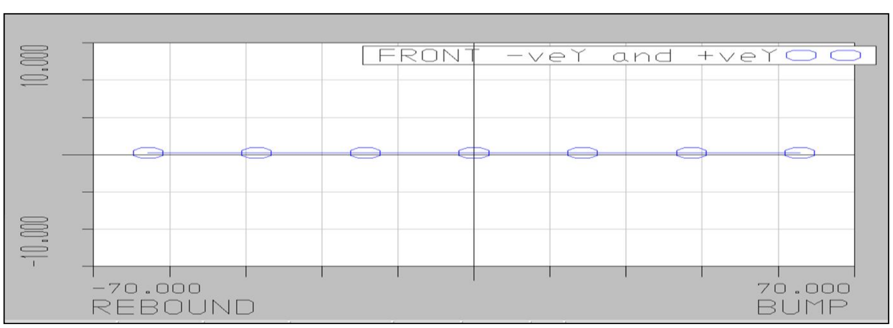
F. *Incremental Geometry Values*

Bump (mm)	Camber Angle (deg)	Toe Angle {SAE} (deg)	Castor Angle (deg)	Kingpin Angle (deg)	Damper1 Ratio (-)	Spring1 Travel Ratio (-)
60.00	-0.1866	-2.1169	0.1433	8.1566	1.742	1.742
40.00	-0.0836	-1.2104	0.1433	8.0520	1.790	1.790
20.00	-0.0216	-0.5071	0.1432	7.9902	1.838	1.838
0.00	0.0000	0.0000	0.1432	7.9696	1.884	1.884
-20.00	-0.0202	0.3119	0.1432	7.9907	1.928	1.928
-40.00	-0.0851	0.4234	0.1433	8.0560	1.971	1.971
-60.00	-0.2000	0.3224	0.1433	8.1706	2.012	2.012

1) *Camber Graphs*



2) *Castor Angle*



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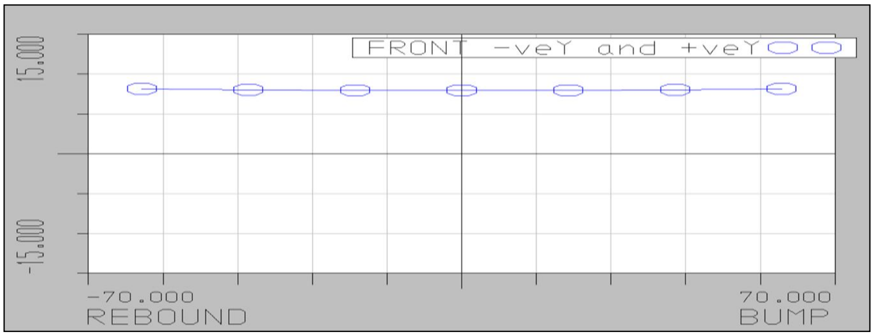
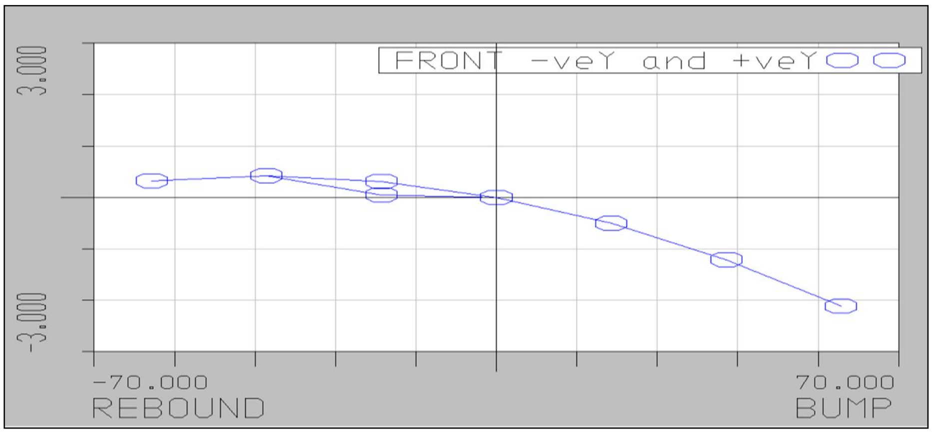
3) Kingpin Angle



4) Toe Angle



G. Rear Suspension

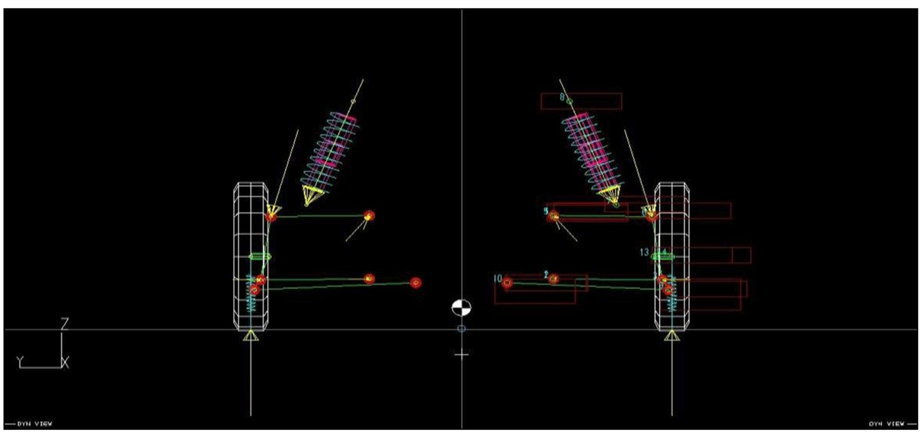


Fig.4. Forces on rare

Reaction force acting from the ground on the wheel = (Mass per wheel * 9.81) N = (63 kg * 9.81) N = 618.03 N Horizontal distance of reaction force from hinge point = 31.75cm ….from suspension geometry Horizontal distance of strut attachment point from hinge point = 26.8cm

By taking moment about hinge points,618.03*31.75 = Spring Force * 26.8

Considering the dynamic factor, Dynamic force acting on the spring = 2196.54 N

According to the ride conditions and road quality for hybrid vehicle , it is concluded that the optimum spring travel should be approx. 3

Hence, Required Spring Stiffness =Dynamic Spring Force/Spring Deflection = 2196.54/62.5 = 35.14 N/mm

Angle Correction Factor(ACF) = 0.93

Motion ratio = (26.8/31.75)^2=0.844

Wheel Rate = 23.05

Suspension frequency = 1.8 Hertz

H. Mounting Points

X (mm)	Y (mm)	Z (mm)		
3819.00	-247.00	238.91	POINT:1	Lower wishbone front pivot
4179.00	-247.00	238.91	POINT:2	Lower wishbone rear pivot
4092.00	-539.00	238.91	POINT:3	Lower wishbone outer ball joint
4092.50	-247.00	438.91	POINT:4	Upper wishbone front pivot
4332.00	-247.00	438.91	POINT:5	Upper wishbone rear pivot
4092.50	-511.00	438.91	POINT:6	Upper wishbone outer ball joint
4145.00	-415.00	475.00	POINT:7	Damper wishbone end
4180.00	-290.00	800.00	POINT:8	Damper body end
4214.50	-555.06	208.00	POINT:9	Outer track rod ball joint
4245.50	-122.56	227.00	POINT:10	Inner track rod ball joint
4092.50	-515.56	313.10	POINT:13	Wheel spindle point
4092.50	-564.56	313.10	POINT:14	Wheel centre point
4030.00	-254.56	195.00	POINT:15	Part 1 C of G
4170.00	-334.56	450.00	POINT:16	Part 2 C of G
4230.00	-339.56	220.00	POINT:17	Part 3 C of G
4130.00	-534.56	275.00	POINT:18	Part 4 C of G

I. Static Values

Camber Angle (deg):0.00

Toe Angle {Plane} (deg):0.00

Toe Angle {SAE}(deg):.0.0

Castor Angle (deg):0.14

Castor Trail (hub) (mm):-0.31

Castor Offset (grnd) (mm):0.90

Kingpin Angle (deg):7.97

Kingpin Offset (w/c) (mm): 35.9

Kingpin Offset (grnd) (mm): 3.19

Mechanical Trail (grnd) (mm): 0.90

ROLL CENTRE HEIGHT (mm):0.23

LHS WHEEL (-ve Y): TYPE 6 INCREMENTAL GEOMETRY VALUES Double Wishbone, a damper to upper wishbone

Bump Travel	Camber Angle	Toe Angle	Castor Angle	Kingpin Angle	Damper1 Ratio	Spring1 Ratio

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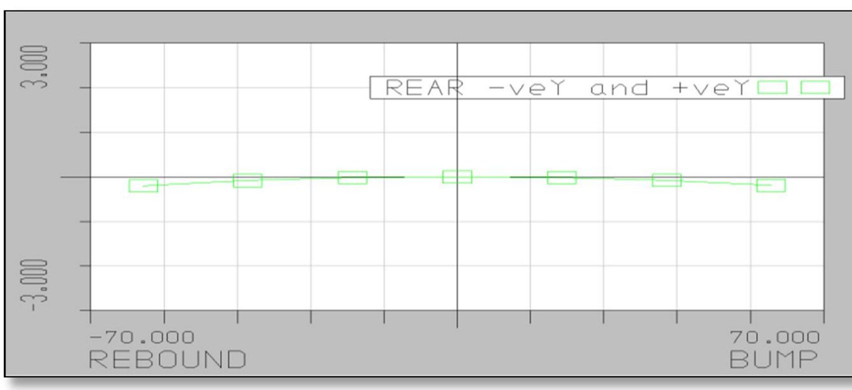
(mm)	(deg)	{SAE} (-) (deg)	(deg)	(deg)	(-)	
60.00	-0.1872	-2.1026	0.1433	8.1570	1.519	1.519
40.00	-0.0839	-1.2044	0.1433	8.0522	1.534	1.534
20.00	-0.0216	-0.5059	0.1432	7.9903	1.549	1.549
0.00	0.0000	0.0000	0.1432	7.9696	1.563	1.563
-20.00	-0.0202	0.3143	0.1432	7.9907	1.576	1.576
-40.00	-0.0853	0.4322	0.1433	8.0562	1.586	1.586
-60.00	-0.2005	0.3420	0.1433	8.1711	1.593	1.593

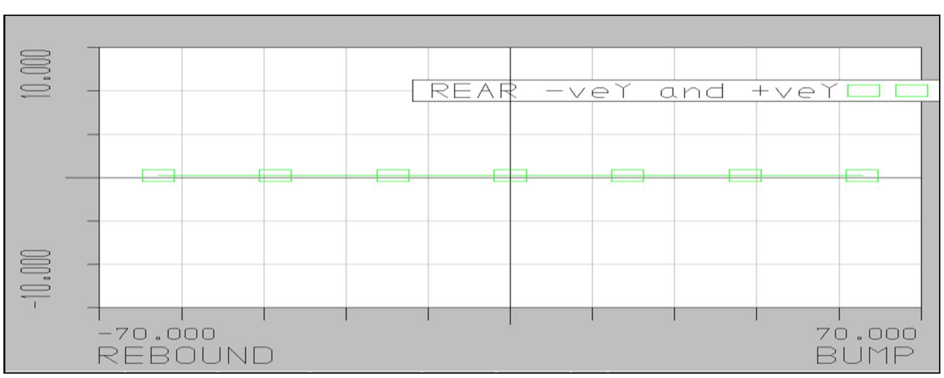
RHS WHEEL (+ve Y): TYPE 6 Double Wishbone, a damper to upper wishbone
INCREMENTAL GEOMETRY VALUES

Bump Travel (mm)	Camber Angle (deg)	Toe Angle {SAE} (deg)	Castor Angle (deg)	Kingpin Angle (deg)	Damper1 Ratio (-)	Spring1 Ratio (-)
60.00	-0.1872	-2.1026	0.1433	8.1570	1.519	1.519
40.00	-0.0839	-1.2044	0.1433	8.0522	1.534	1.534
20.00	-0.0216	-0.5059	0.1432	7.9903	1.549	1.549
0.00	0.0000	0.0000	0.1432	7.9696	1.563	1.563
-20.00	-0.0202	0.3143	0.1432	7.9907	1.576	1.576
-40.00	-0.0853	0.4322	0.1433	8.0562	1.586	1.586
-60.00	-0.2005	0.3420	0.1433	8.1711	1.593	1.593

1) *Graphs Camber*



2) *Castor Angle*



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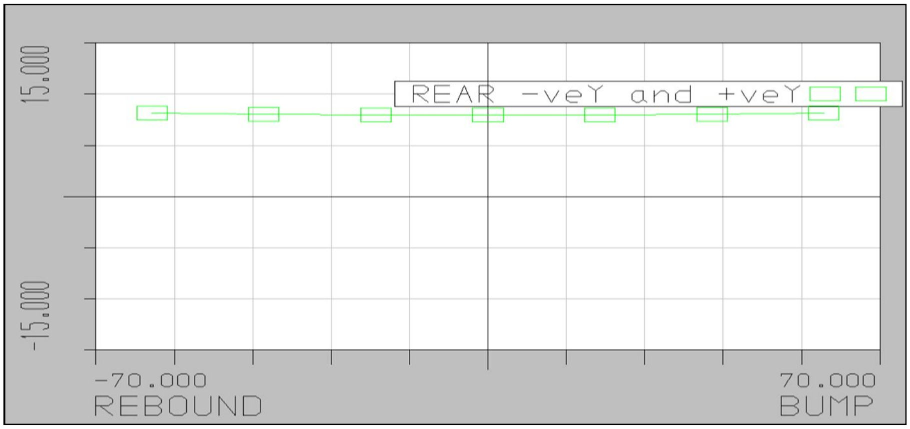
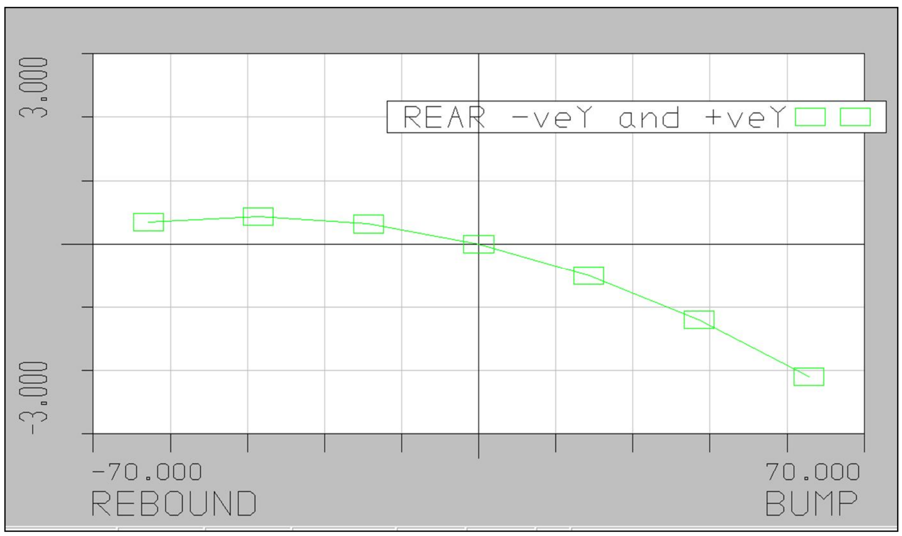
3) Kingpin Angle

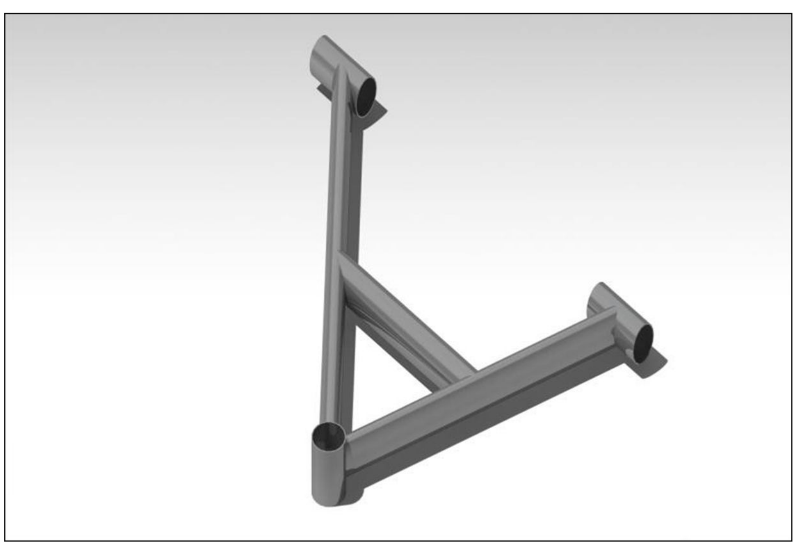


4) Toe Angle



J. Designing Of Wishbone

1) Cad Model Of Front Lower Wishbone



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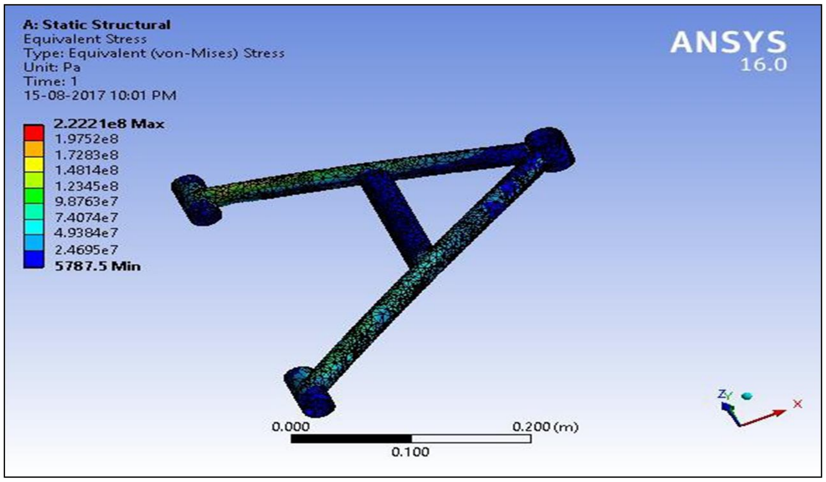
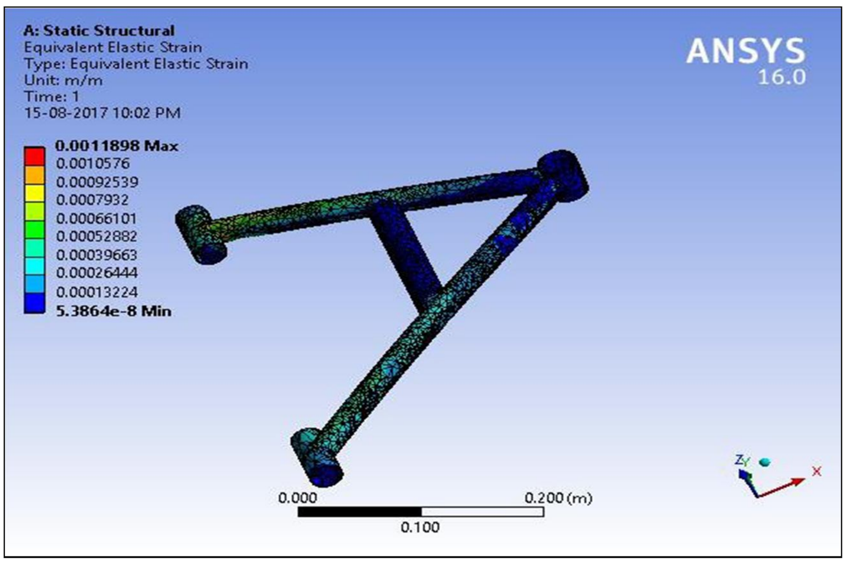
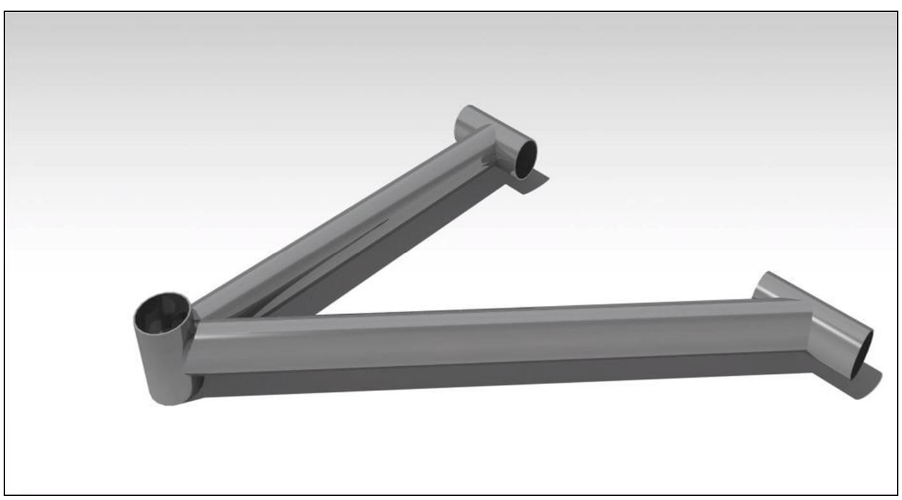
2) Analysis Of Front Lower Wishbone





3) Cad Model Of Front Upper Wishbone



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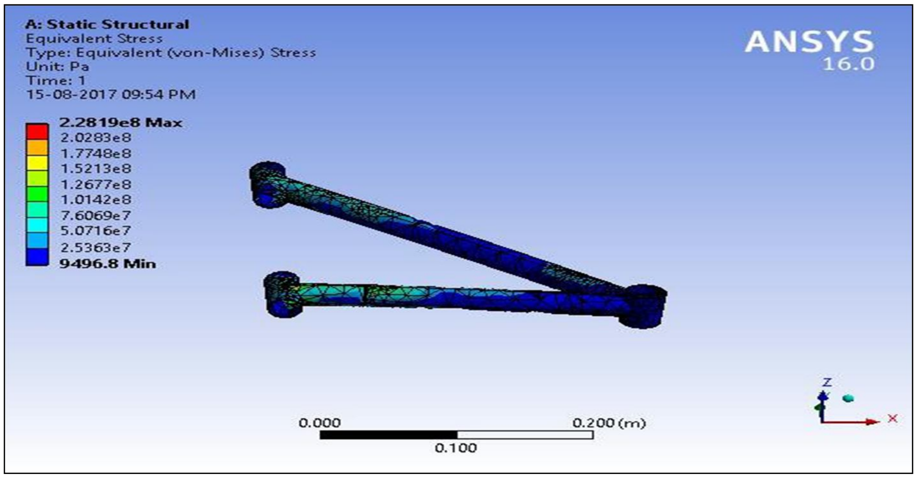
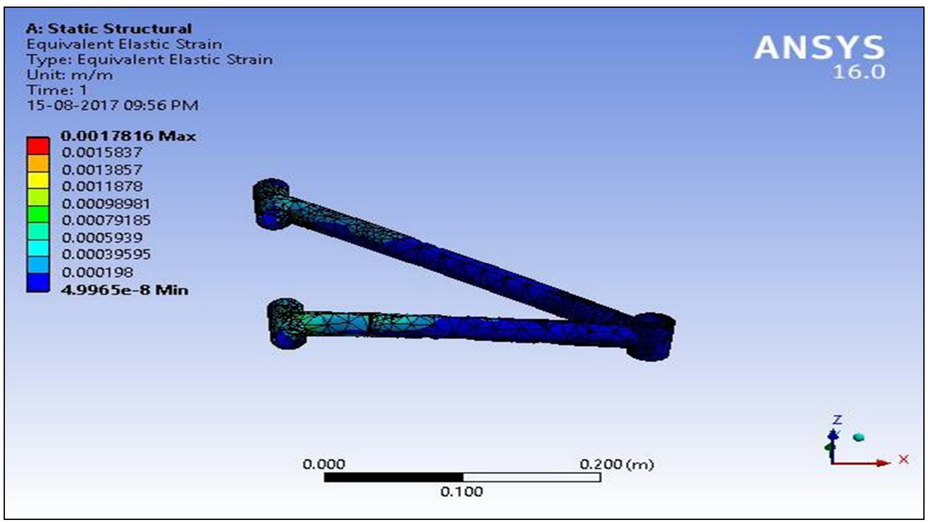
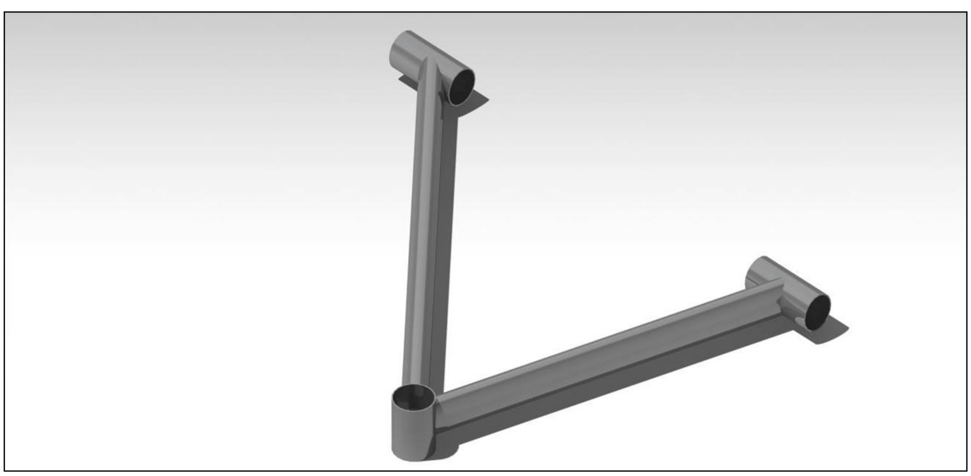
4) *Analysis Of Front Upper Wishbone*





5) *Cad Model Of Rear Suspension Lower Wishbone*



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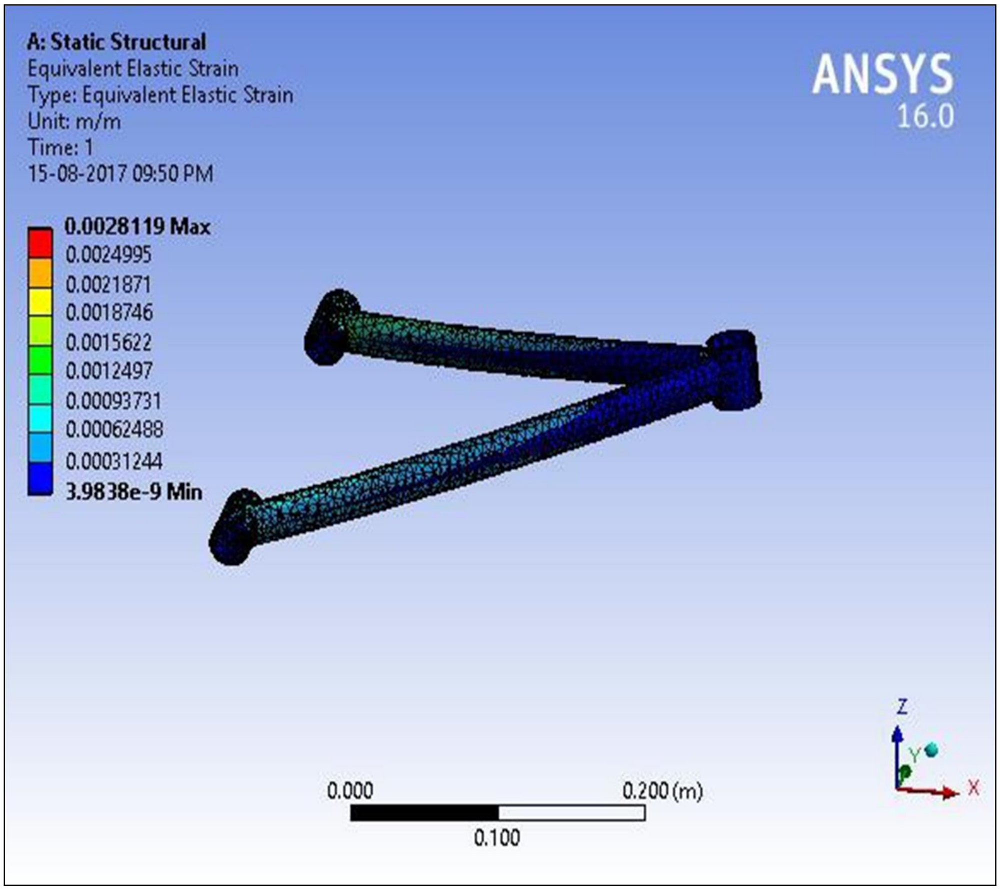
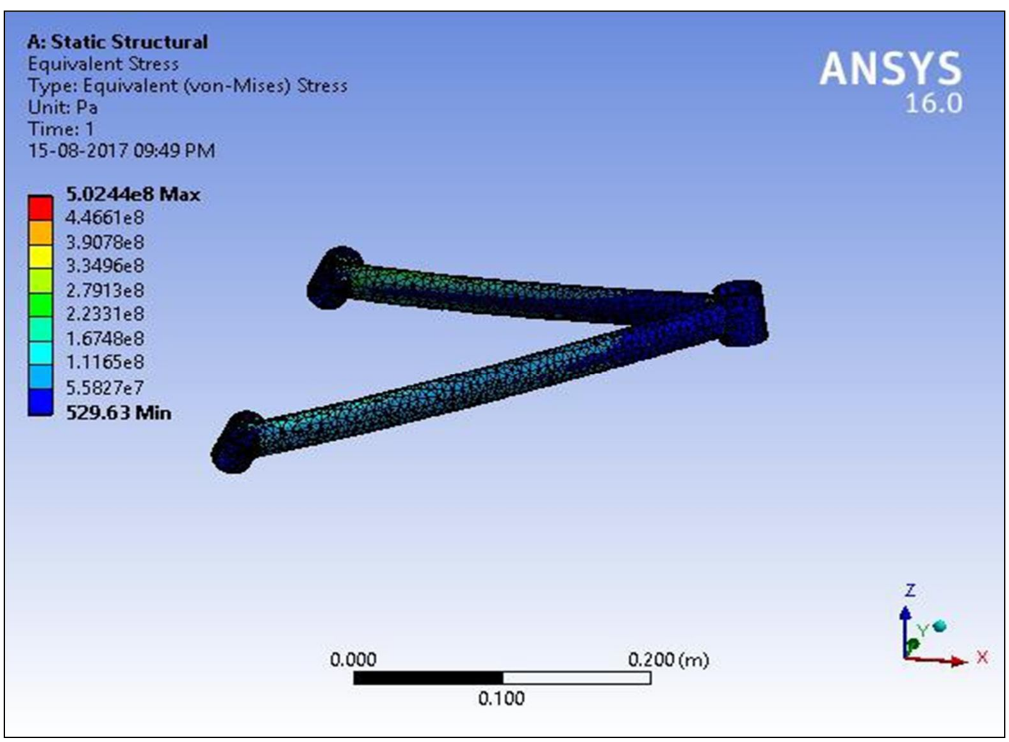
6) Analysis Of Rear Lower Wishbone





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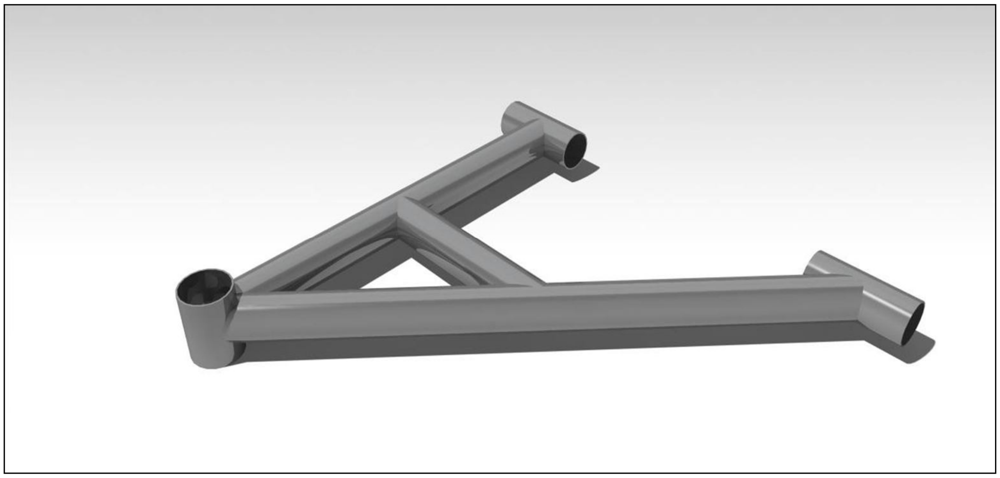
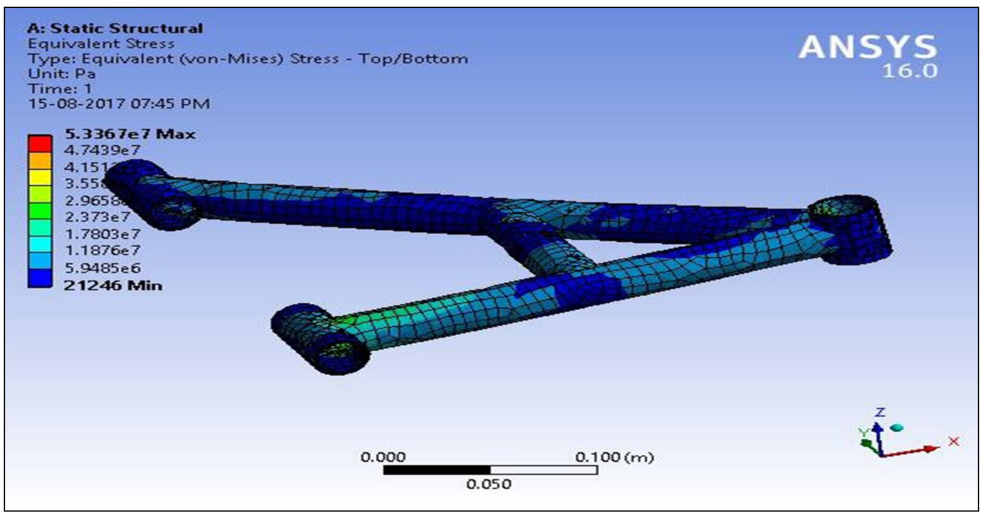
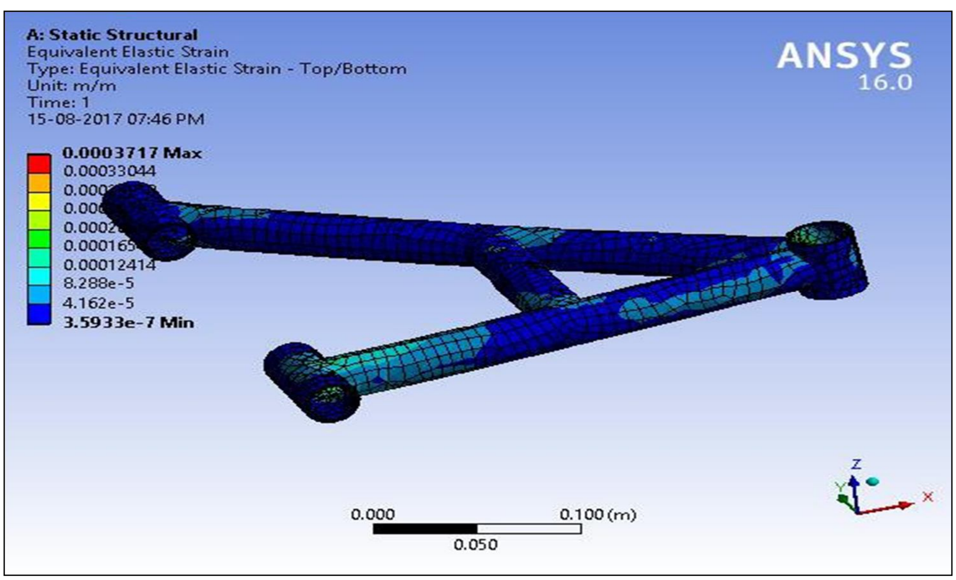
7) Rear Upper Wishbone



8) Analysis Of Rear Upper Wishbone





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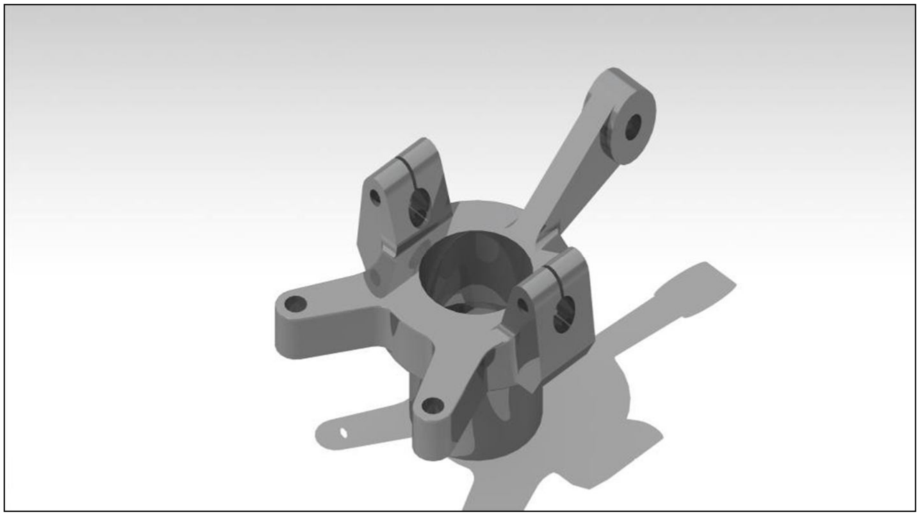
9) Design Of Knuckle



V. STEERING SYSTEM

A. Introduction

The controlling behavior of a vehicle is influenced by the performance of its steering system. The steering system consists of the steering wheel, steering column, and a linkage system. Our vehicle is controlled by the rack movement using pinion direct from the steering column to the steering arm.

B. Design Procedure

From the studies of all different types of steering and the mechanism which could fulfill our requirements, we have decided to used simple RACK AND PINION steering system. To achieve perfect steering and correct steering angle, two types of mechanisms, have been devised namely

1) Davis

2) Ackermann

Out of this Ackermann, mechanism is almost universally used because of following reasons: Ackermann system is comparatively simple to design and posses fewer problems both in practical and theoretical aspects. Ackermann system uses turning pair arrangement with provide system less frictional surfaces thereby decreasing frictional and wear losses, Ackermann system gives close approximations to results provided in real life.

C. *Selection Of Steering Gear*

Steering gears are used to convert the rotational motion of the steering wheel into the linear motion, which is needed to turn the wheels. It provides least steering ratio, which makes it more handy and efficient during turning. For design consideration purposes it has been seen that use of Rack and pinion gear is widely used compared to other steering gears. This is because of following reasons:

D. Advantages Of The Rack And Pinion Steering Gear Over Other Mechanisms

A rack and pinion is a type of linear actuator that comprises a pair of gears which convert rotational motion into linear motion. A circular gear called "the pinion" engages teeth on a linear "gear" bar called "the rack"; rotational motion applied to the pinion causes the rack to move relative to the pinion, thereby translating the rotational motion of the pinion into linear motion. Rack and pinion steering system provide less backlash and greater feedback compared to other steering gears (especially mechanical linkage type). It has a more compact and robust design compared to worm and worm wheel-type steering gears. It is cheap and readily available.

E. Steering Geometry Calculations

1) Steering geometry data:

WHEELBASE (L) – 1687.5mm

WHEEL CENTRE DISTANCE (B) – 220mm

DISTANCE FROM FRONT AXIS TO TRACK ROD – 180mm

Above data are assumed or fixed according to desirable steering geometry and other restrictions on the basis of analysis

TRACK WIDTH (W) – 1315 mm

STEERING ARM LENGTH (r) – 125 mm

TRACK ROD (D) – 395 mm

PIVOT in ANSYS.

a) $\cot\Phi - \cot\Theta = B/L$.=..EQ1

b) $\sin(\alpha+\Theta) - \sin(\alpha-\Phi) = 2\sin(\alpha)$ EQ2

c) TURNING RADIUS INNER FRONT = $L/\sin\Theta - (W-B)/2$ EQ3

d) TURNING RADIUS OUTER FRONT = $L/\sin\Phi + (W-B)/2$ EQ4

e) STEERING ARM ANGLE = $\arctan(t/2L)$ EQ5

Φ – OUTER WHEEL LOCK

Θ – INNER WHEEL LOCK

α – STEERING ARM ANGLE

$\alpha = 21.28$ DEGREES by EQ5

$\Phi = 22$ DEGREES by EQ2 (When $\Theta = 30.52$ degrees (closest to the correct steering angle)) see the excel sheet for the other angles considered.

T.R.I.F = 2775.4 mm

T.R.O.F = 5052.2 mm by EQ3 & EQ4 RESPECTIVLY.

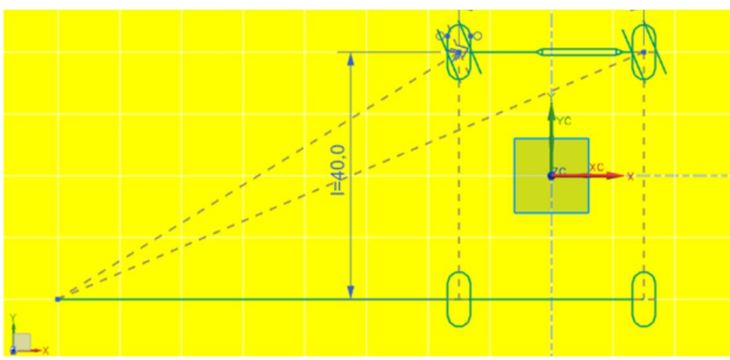
RACK DETAILS:

STEERING RATIO – 9:1

RACK LENGTH – 220mm

FOR STEERING ARM UP TO 4 INCH

TIE ROD LENGTH: 395mm



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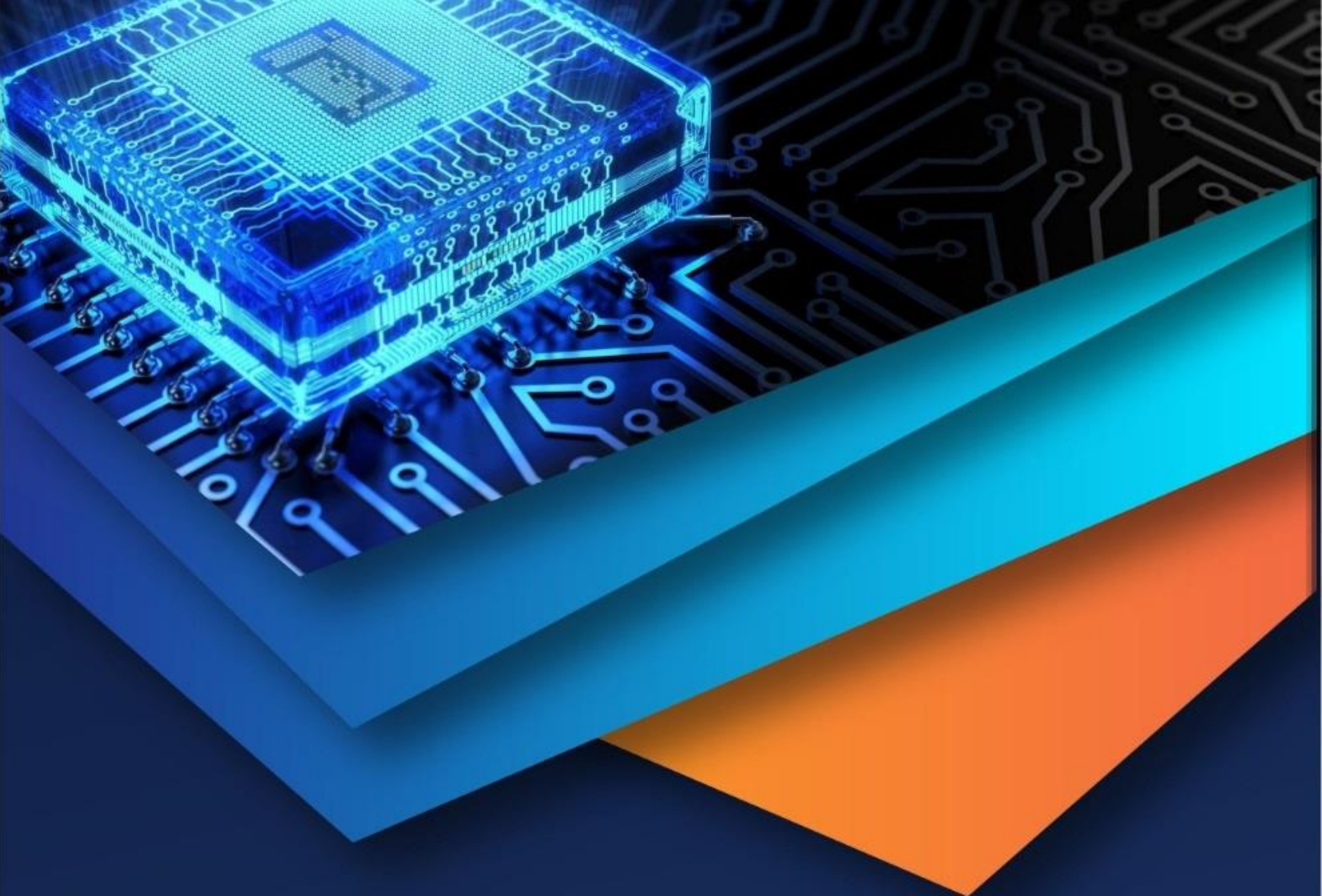

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