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Improved Vehicle Dynamics with Development in Suspension Geometry

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Abstract: The motive of undertaking this work of "Improved Vehicle dynamics with development in Suspension Geometry" is trying to increase the performance of the vehicle by keeping it stable and able to take all the loads coming from the ground and run comfortably. The key areas focused were to maintain minimal roll angle with controlling yaw/roll motions of vehicle and accelerate in a better way. The primary objective of the suspension system in this atv is to maximizing the contact between the tires and the road surface, providing steering stability and good handling, evenly supporting the weight of the vehicle (including the frame, engine, and body), and ensuring the comfort of passengers by absorbing and dampening shock provide safe vehicle control with free from vibrations. Design calculations are done for the geometry and frame as per the requirements. Nx11.0 has been chosen to design the components, Ansys solver is used for the analysis, lotus shark is used for the simulation, Manufacturing is done according to the design using all manufacturing tools with performing various operations and a run-virtual compliance test is performed for checking the vehicle dynamic performance and the vehicle is even tested in a rough-terrain.
Keywords: Camber; Toe; Vehicle Handling; Roll angle; Stable.

I. INTRODUCTION

A. Introduction to Suspension System

In 2008, 58% of all Sport Utility Vehicle (SUV) passengers involved in rollover accidents were fatally injured (Administration, 2008). In addition, in 2010 rollover accidents accounted for 35% of all vehicular occupant fatalities (Administration, 2010).

The risk of rolling over is largely affected by a vehicle's suspension system and how it counteracts external forces while the vehicle is in motion. Although suspension systems have made significant progress to better mitigate the risk of rollovers, these accidents still occur frequently and are incredibly dangerous to vehicle occupants. The development of a universal 4 bar linkage suspension system that addresses this issue will significantly reduce the number of rollover accidents each year.

This work focuses on a universal 4 bar linkage suspension system that will utilize semi-active technology allowing the operator to adjust the geometry of the suspension using an interface. Altering the geometry of the suspension system significantly impacts more than just the stability of the vehicle. It also affects the behaviour of the entire vehicle and how it will respond to different surfaces and manoeuvres. This adjustability is expected to allow for the system to be used in a variety of scenarios that will far exceed the state of the art in suspension systems today.

To alter the geometry of the suspension system, we utilize mechanisms to power instantaneous and independent motion in the linkage joints of the system. The position change of the instantaneous and independent motion in the linkage joints of the system. The position change of the linkage joints, as a result of the motion, occurs in the vertical (z) direction with respect to a predetermined origin (on a local nonrotating coordinate system). The operator determines the vertical position of the links through an interface. This allows the operator to adjust the system geometry to improve the efficiency and stability of the vehicle on a variety of terrains. An improved suspension system that effectively provides instantaneous adaptability would have potential in a plethora of fields and scenarios. This system could be used to improve performance and efficiency in drag racing, rock climbing, desert racing, military vehicles and more. In addition, the system would not be limited to strictly passenger vehicles; it could also improve the performance of mobile robots, toys, or any mechanical system that requires a suspension for its functionality. Thus, this suspension design is expected to make broad contributions far beyond our goal of improving vehicle safety.

B. Importance and Functions of the System

For the suspension it is important to keep the road and wheel in contact with contact as much as possible because all the ground forces acting on the vehicle transfer through the contact patches of the tires. The suspension should also protect the vehicle itself and any luggage or cargo from damage and wear.

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1) *Functions*
a) To prevent the impact forces from being transmitted to the vehicle body.
b) To preserve the stability of the vehicle in pitching or rolling.
c) To safeguard the occupants from road shocks.
d) To provide good road holding while driving, cornering and braking.
e) A suspension geometry must be designed to meet the requirements or ideals of the vehicle to be built, a lot of factors must be taken into consideration such as the roll rate, ride height, spring rates, etc.

The design of this system gets complex in a way because, while being restricted (controlled) in their motion path by the control arms, the wheel will have camber, caster and toe change. In such a scenario various component such as toe link, camber link, shock absorber mounting become extremely important. The work has been called out in a phase-wise manner to ease up the tasks for providing better results and also to allow for modification of the design if there may be a need.
Determination of the geometry and the type of setup to be used

2) *Considerations made in Designing Phase Were*
a) Independent nature.
b) Make within standard track width of 64".
c) Smaller packaging.
d) Fabrication limitations
e) Weight reduction.

3) *Considerations Made for a Selection of Spring and Damper System*
a) Motion Ratio
b) Installation Ratio
c) Spring rates
d) Damping characteristics
e) Weight of the Spring-Damper system
f) Initial compression of spring
g) Ride frequency

In this regard, various other parameters which are explained in later chapters have also been taken care off, some of them being, the position of roll centre, minimization of scrub radius, anti-squat, anti-dive. To avoid rollover the vehicle's centre of gravity has been put as low as possible, by doing this we have restricted the movement of the centre of gravity to an extent.
After establishing the design parameters the team has done different types of market surveys locally and on the internet to find components that are well suited for the purpose, the emphasis was on manufacturing most of the components to avoid outsourcing, although expensive, it would serve all the requirements as well as have clean engineering ethics rather than modifying an existing setup to suit ours.

II. DESIGNING PHASE

A. *Front Suspension designed for All-Terrain Vehicle*

1) The Double wish-bone setup was preferred as it met many of our requirements which were
a) Adjustability.
b) Ability to package in a small space.
c) Simpler construction.
d) Lightweight yet robust construction.

2) Main components in the suspension system are
a) Wishbones or "A" arms
b) Mounting tabs
c) Shock absorbers

A double-wishbone geometry which consists of two links that are used to connect the chassis on one end and to the upright at another end. The two links namely, upper wishbone and lower wishbone each of which is provided with two revolute joints at the chassis end and one rotational joint at the upright.

The design of suspension earlier was done using paper doll-models connected with threads to verify the motion, but in a more sophisticated way, the design has been done using various computer software's that provide better accuracy and analysis.

B. *Front Suspension Geometry*

The initial parameter for designing the front suspension was the track width which was set for 64" as per standards but in fabrication we have decided a track width of 60" as in the dynamic scenarios the overall change of the track width in vehicle should not affect any other parameters.

1) *Geometric Planning:* The setup started with a set of unknowns and a set of desired values, important unknown parameters for the design of components were:

a) Length of Lower A-arm
b) Length of Upper A-arm
c) Kingpin Inclination
d) Hub offset
e) Spindle offset
f) Angles of wishbones with the axis at desired ride height
g) Wheel offset
h) Motion Ratio
i) Shock Mounting clamp

These parameters were considered for designing the suspension components to undergo various analysis and dynamic simulation.

2) *Desired Suspension Characteristics Were*

a) Low scrub radius
b) Ride height
c) Height of roll centre
d) Avoid Bump steer

C. *Rear Suspension designed for All-Terrain Vehicle*

The rear suspension of the vehicle is different from the front suspension in the following ways:

- Must be designed to bear greater loads due to rear placement of the engine
- Must not allow great camber changes
- Allow the live-axle to be fixed without link-clash
- Must not allow toe changes
- Must not allow axle-plunge out.
- Must be independent.

With the above premise, the types of suspensions were reviewed again, following which the decision came down to the selection between double wishbone suspension and a semi-trailing arm with upper and lower links design. The double-wishbone suspension although very adjustable and lightweight would not be a match here as the rear track width limited to 58" which meant that the packaging space will not be enough, the absence of chassis support members to mount the arms led us to the design of semi-trailing arm, with upper and lower links.

1) *Rear Suspension Geometry:* Semi-Trailing arm with upper and lower links suspension provides all the required camber and toe control since it is possible to alter one parameter at a time without affecting others, whereas in double wishbone were moving a hard point or changing its position it affects a minimum of two parameters.

a) *Components Used in the rear Suspension*

- Semi-Trailing arm
- Trailing arm mount
- Upper Link
- Lower Link
- Rear Shocks

b) Desired Suspension Characteristics Were
- Extend to off-Roading
- Allows vehicle to flex more
- Able to move more easily in varying angles at off-roading
- Minimal toe control

D. Selection of Shocks

The shock absorber calculations are done according to the vehicle weight and spring stiffness was chosen accordingly from calculating the sprung mass and un-sprung mass of the vehicle which were 130kgs and 50 kgs. Since, it was an adaptive suspension system type designed for better performance. We have chosen fox float 3 Shocks which are quite very adjustable according to the terrain and stiffness was able to change as per the scenario meant.

Fox float 3 air shocks, which use air as springs, instead of heavy coil springs which are good in load optimizing air technology.

A high-performance, velocity-sensitive air sleeve with shimmed damping system is installed in it. Fox float 3 air dampers contain high viscosity index shock oil and high-pressure nitrogen gas separated by an internal floating piston system. This helps to ensure fade-free damping and stabilised in most riding conditions.

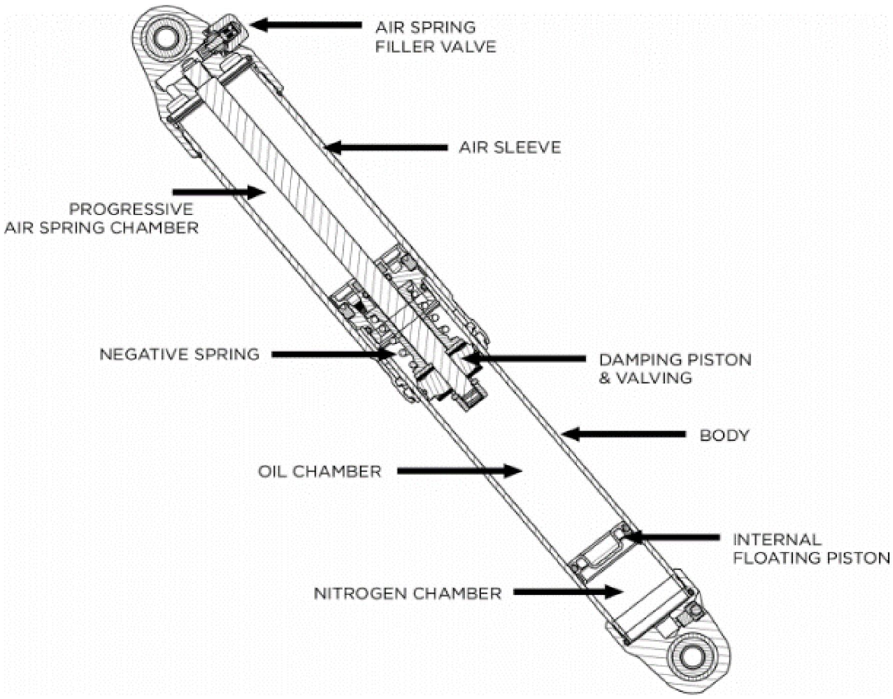

Figure 1: FOX FLOAT 3

Float 3 shocks are built using 6061-t6 aluminium for light weight and strength. The damper shaft is super-finished for long seal life and low friction. All of the wipers and seals are engineered specifically. The damper shafts, wipers and seals are installed within the air sleeve to ensure it is free from dirt, water and ice

Figure 2: Air pump used for Fox Float 3 to adjust stiffness (0-150 psi).

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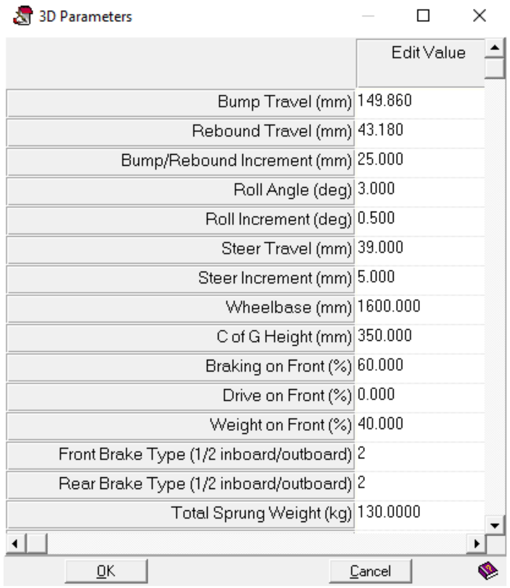
III. SIMULATION

All the required suspension parameters are given as input to the lotus simulation software according to the All-terrain vehicle dimensions such as the 3D parameters as mentioned.



3D Parameters

	Edit Value
Bump Travel (mm)	149.860
Rebound Travel (mm)	43.180
Bump/Rebound Increment (mm)	25.000
Roll Angle (deg)	3.000
Roll Increment (deg)	0.500
Steer Travel (mm)	39.000
Steer Increment (mm)	5.000
Wheelbase (mm)	1600.000
C of G Height (mm)	350.000
Braking on Front (%)	60.000
Drive on Front (%)	0.000
Weight on Front (%)	40.000
Front Brake Type (1/2 inboard/outboard)	2
Rear Brake Type (1/2 inboard/outboard)	2
Total Sprung Weight (kg)	130.0000

OK Cancel

Figure 3: Input 3D parameter given.

The Hardpoints are varied several times according to the required roll centres are achieved and making sure maintaining all the roll centres, scrub radius, king pin inclination, camber, caster and toe are achieved according to the designed values.

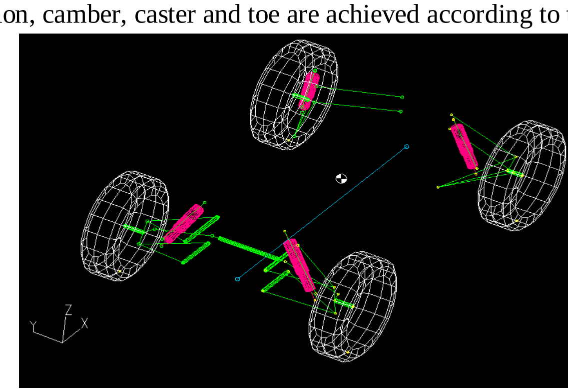


Figure 4: Isometric View of the Suspension System.

The Following two figures shows the simulation of maximum and minimum travel of the suspension system.

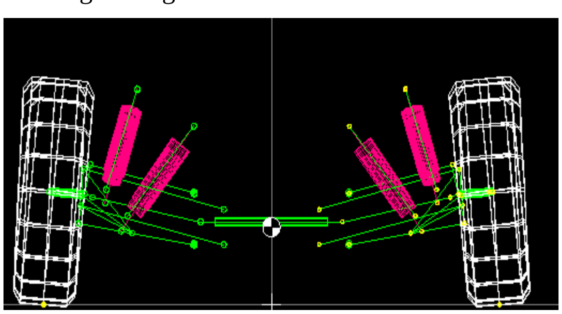


Figure 5: Maximum droop Scenario, no link Clash Detected

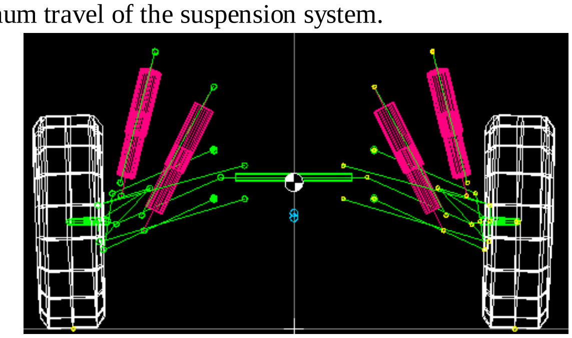


Figure 6: Maximum Bump and side Roll Scenario, no link Clash Detected

From the above analysis it is noted that the design satisfies the requirements and does not have any aberrant effects, the link geometry found to be very good.

The following graphs gives more in-detail view of the wheel travel changes v/s the geometry

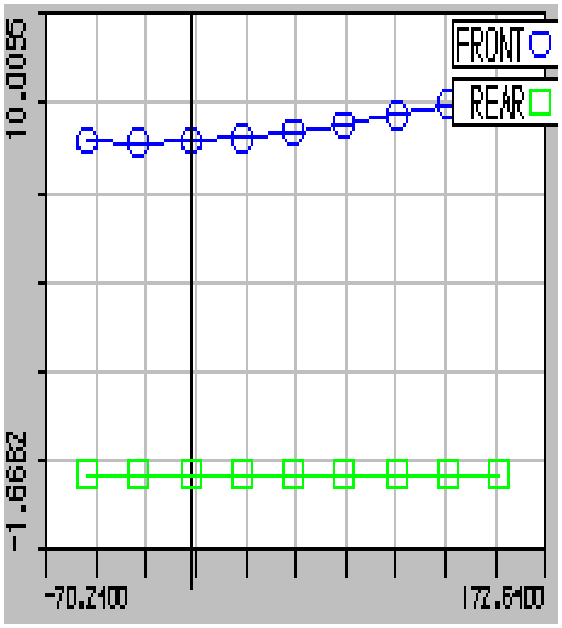


Figure 7: Wheel Travel V/S King Pin Inclination

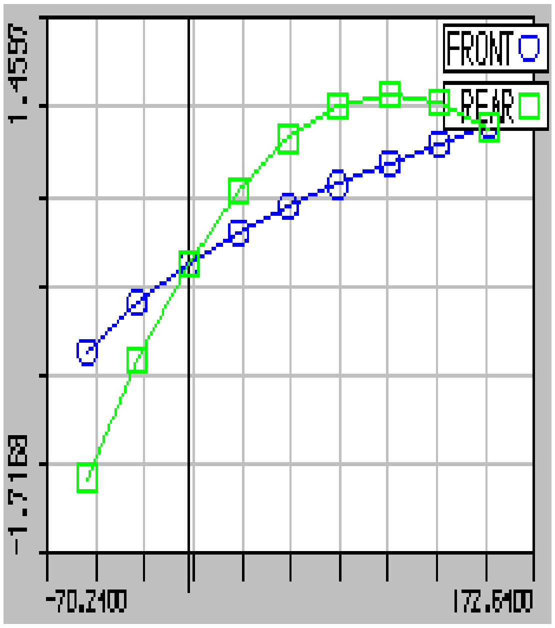


Figure 8: Wheel Travel V/S Toe Change

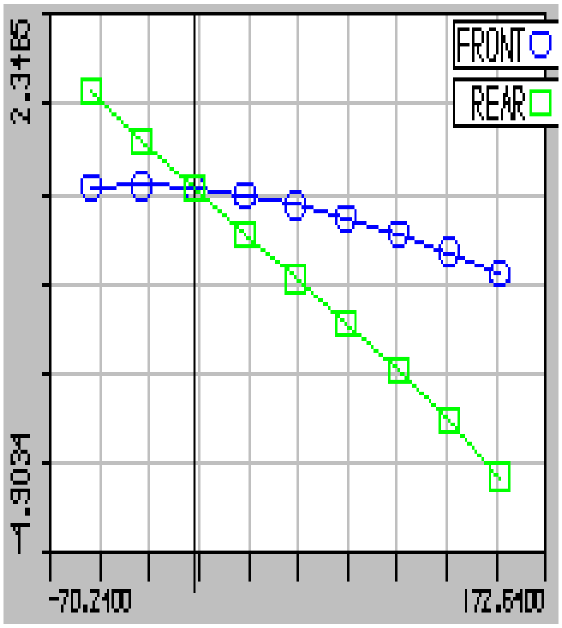


Figure 9: Wheel Travel V/S Camber Change

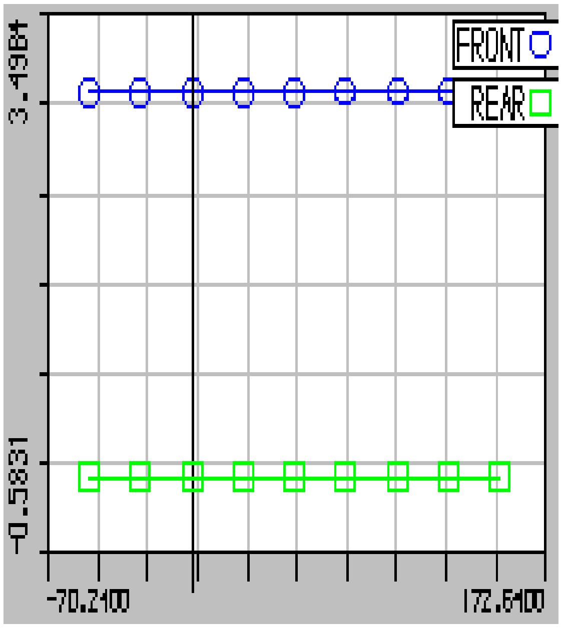


Figure 10: Wheel Travel V/S Castor Change

A.   RUN-Virtual Compliance Test

Simulation has offered tremendous progress in vehicle development. time and effort for iterations on vehicle design have dramatically decreased by using this.

The designed vehicle is taken on to a test rig as shown in the figure and all the four wheels are placed over the jacks where the test rigs and jacks are moved into various scenarios according to the travels set for the designed vehicle and simulates according to a real-time scenarios making all the permutations and combinations such as each individual wheel is moved into bump and droops, two wheels, all wheels at a time and even with turning off the front wheels too and it is evaluated whether it can sustain or not.

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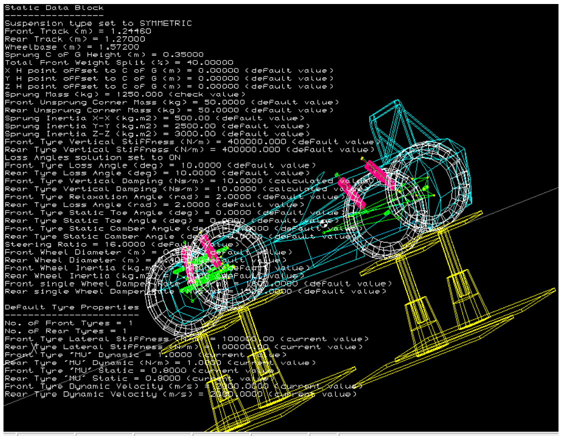

Figure 11: Run Virtual Compliance Test Starting

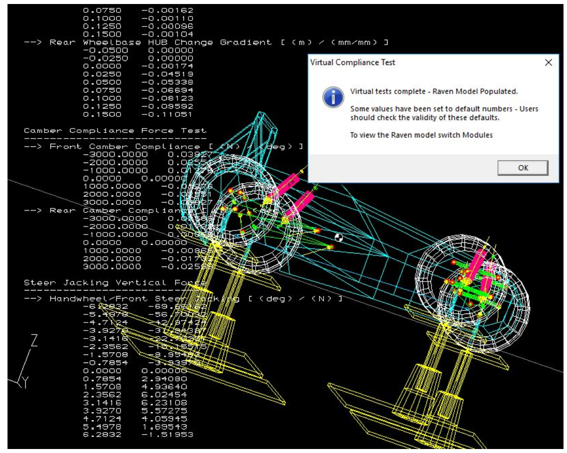

Figure 12: Run Virtual Compliance Test Successfully completed

The test was completely successful and checked where there were no errors or no linkage breaks or scenario where it couldn't pass through. Therefore, after the simulation process, the design of the combination suspension of front and rear is deemed to be fit for use in the All-Terrain Vehicle, the body roll and steer characteristics have shown that the suspension will be capable of doing the required job even on the inhospitable terrain. Provided with the zero-droop rear design, the vehicle will not encounter any mechanical problems such as axle plunge-out in case of high-speed cornering or large potholes.

IV. MODELLING PHASE

A. Front Suspension Components

The following dimensions for the modelling of upper and lower control arms, Various shapes for control arms were considered initially. Since the control arm is the link between tire and body of the vehicle, it needed to be stiff and strong to support also control the tire motion. There were a lot of variants in the design of control arms in which few are provided by suspension analyser. The length of them was based on front nose dimensions, track width and various other performance significant parameters. The model ought to reflect the lengths mentioned below in the drafts of upper and lower A-arm.

1) *Lower A-arm:* The lower A-arm shock mounting plate was given according to the motion ratio calculated from the suspension geometry to have the required bump to wheel travel.
2) *Upper A-arm:* The upper A-arm was made enough wide that it would get the shocks fit easily into it and make sure that at any point of the wheel travel the shock doesn't make contact with the A-Arm.

Since the shock was given an angle correction factor of 30°, according to the hardpoints and from the mounting point of a shock to the lower A-arm and to the chassis end. The upper A-arm is made enough wide that it fits the shock easily and doesn't make any contact in motion too.

B. Rear Suspension components

The semi-trailing arm which connects the chassis and the rear wheels is important in design point of view as any changes reflect the power transmission ability and many other suspension characteristics. As in our case, it's not just a tube but other joints connected to it.

Some of the consideration that are to be brought down before the beginning of modelling phase are:

1) Length of the semi-trailing arm is such that it projects the cv shafts in perpendicular to travel of the vehicle. As any deviation from it will lead to performance loss
2) Should accommodate the brake calliper
3) Should have a provision at the end of it to pass through cv joint. Also, it should be such that even at any articulation it shouldn't be interference to trailing arm.
4) At appropriate motion ratio, it should have the capacity to hold the air shock firmly also allowing relative rotation.
5) Should not deflect either way when cornering. Ideally should not allow any chamber change.
6) Should not deform in heavy loading. The diameter and tube be considerably adjusted to loading.
7) A single link should be able to locate the wheels without the need of any other links like camber control link etc.

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C. Upper and Lower Link

The upper and lower links are designed in the different lengths according to the performance and the designed suspension geometry to get the best camber travels while cornering and bump travel.

The small difference of mm between those is adjusted by the Heim's joint because it's easy to fabricate same length pipes for both the links for ease of fabricating it and jigs also can be easily prepared for it.

V. ANALYSIS PHASE

A. Front Suspension Components

Material and geometry data

Material	
Assignment	steel 4130
Nonlinear Effects	Yes
Thermal Strain Effects	Yes
Bounding Box	
Length X	330. mm
Length Y	67.7 mm
Length Z	348.91 mm
Properties	
Volume	1.6524e+005 mm³
Mass	1.2971 kg
Centroid X	-0.26096 mm
Centroid Y	1.297 mm
Centroid Z	178.92 mm
Moment of Inertia Ip1	13782 kg·mm²
Moment of Inertia Ip2	27420 kg·mm²
Moment of Inertia Ip3	13901 kg·mm²
Statistics	
Nodes	502302
Elements	287920
Mesh Metric	Element Quality
Min	0.165889299649668
Max	0.999988804005109
Average	0.785124540064304
Standard Deviation	0.113911613090989

Lower A-arm

Sizing	
Size Function	Adaptive
Relevance Center	Coarse
Element Size	1.50 mm
Initial Size Seed	Assembly
Transition	Fast
Span Angle Center	Coarse
Mesh Based Defeaturing	On
Defeature Size	Default
Minimum Edge Length	2.30 mm
Quality	
Check Mesh Quality	Yes, Errors
Error Limits	Standard Mechanical
Target Quality	1.e-003
Smoothing	High
Mesh Metric	Element Quality
Min	0.16589
Max	0.99999
Average	0.76512
Standard Deviation	0.11391

Upper A-arm

Sizing	
Size Function	Adaptive
Relevance Center	Coarse
Element Size	2.0 mm
Initial Size Seed	Assembly
Transition	Fast
Span Angle Center	Coarse
Mesh Based Defeaturing	On
Defeature Size	Default
Minimum Edge Length	31.4160 mm
Quality	
Check Mesh Quality	Yes, Errors
Error Limits	Standard Mechanical
Target Quality	1.e-003
Smoothing	Medium
Mesh Metric	None
Inflation	
Use Automatic Inflation	None
Inflation Option	Smooth Transition
Transition Ratio	0.272
Maximum Layers	5
Growth Rate	1.2
Inflation Algorithm	Pre
View Advanced Options	No

Figure 13: Analysis data and mesh functions of front suspension components.

Loading conditions are given taking into consideration of worst scenario where the maximum force that can be acting on the components when the vehicle may land on one single wheel after an obstacle.

Calculation

Total weight of the vehicle :179kgs.

Driver Weight :70kgs (Approx. Average Weight).

Total Weight :249kgs.

Force= m*a = 249*9.8 = 2440.2N.

From the toggle point solver in the lotus, the direction of force acting on the vehicle is known and, in that direction, the max load scenario is considered and applied as shown below keeping the fixed supports which are connected to chassis.

1) *Upper A-arm:* The stresses were minimal compared to the lower A-arm since the upper A-arm gives the support for camber gains in bump travels and keep the wheel in required angle and very fewer forces will be acting on this arm since it doesn't have a shock mounting point on it.

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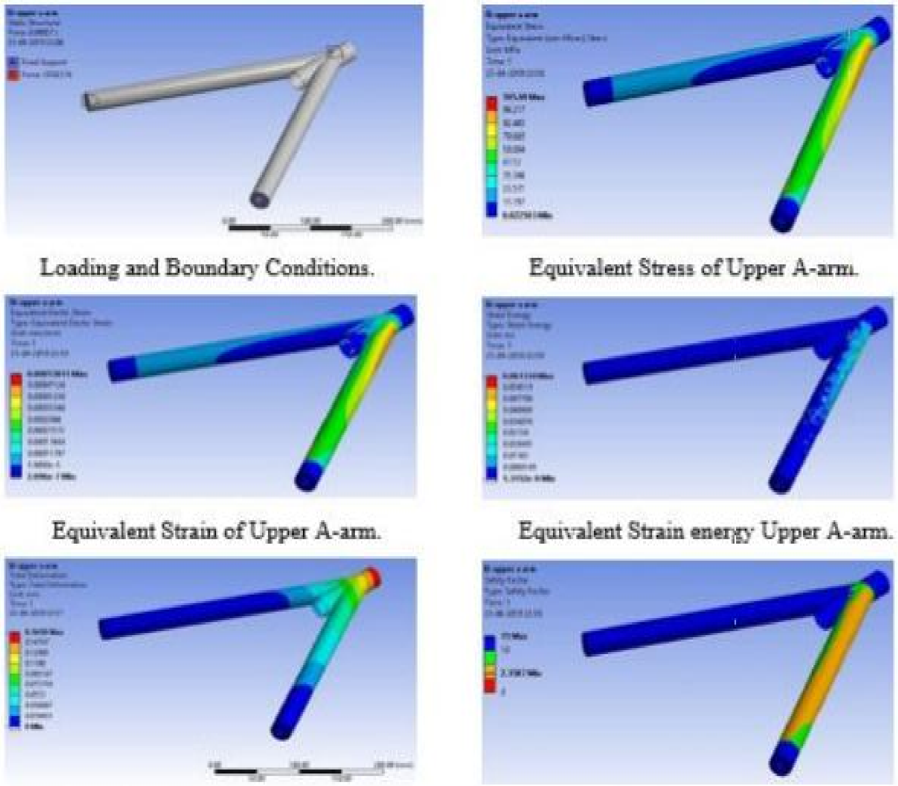


Loading and Boundary Conditions. Equivalent Stress of Upper A-arm.

Equivalent Strain of Upper A-arm. Equivalent Strain energy Upper A-arm.

Total Deformation Upper A-arm. Factor of Safety Upper A-arm.

Figure 14. Analysis results of Upper A- arm.

From all the above analysis results it is evident that the upper A-arm doesn't undergo through direct forces since much force is completely taken by shock and lower A-arm, so the F.O.S is high compared to lower A-arm.

B. Rear Suspension Components

Material and geometry data

Material	
Assignment	steel 4130
Nonlinear Effects	Yes
Thermal Strain Effects	Yes
Bounding Box	
Length X	460.02 mm
Length Y	169.93 mm
Length Z	135.74 mm
Properties	
Volume	1.7737e+005 mm³
Mass	1.3924 kg
Centroid X	292.57 mm
Centroid Y	96.982 mm
Centroid Z	-1.4222 mm
Moment of Inertia Ip1	1296.3 kg·mm²
Moment of Inertia Ip2	26423 kg·mm²
Moment of Inertia Ip3	25331 kg·mm²
Statistics	
Nodes	1106349
Elements	617913
Mesh Metric	None

Semi-trailing arm

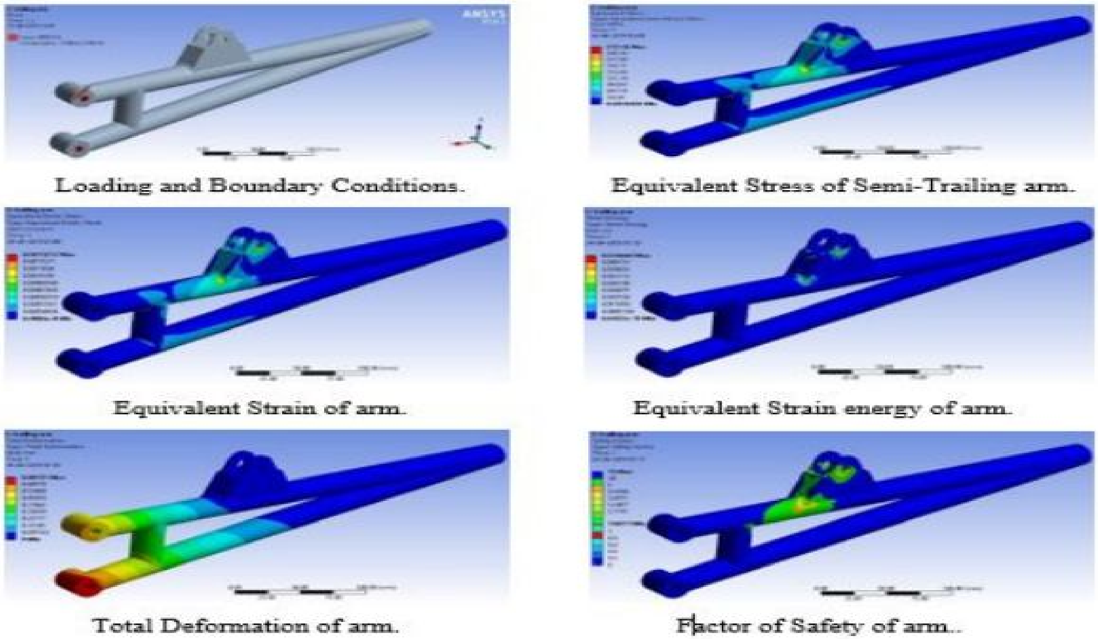
Sizing	
Size Function	Adaptive
Relevance Center	Coarse
Element Size	1.0 mm
Initial Size Seed	Assembly
Transition	Fast
Span Angle Center	Coarse
Mesh Based Defeaturing	On
Defeature Size	Default
Minimum Edge Length	6.640e-003 mm
Quality	
Check Mesh Quality	Yes, Errors
Error Limits	Standard Mechanical
Target Quality	1.e-003
Smoothing	High
Mesh Metric	None
Inflation	
Use Automatic Inflation	None
Inflation Option	Smooth Transition
Transition Ratio	0.272
Maximum Layers	5
Growth Rate	1.2
Inflation Algorithm	Pre
View Advanced Options	No

Links

Sizing	
Size Function	Adaptive
Relevance Center	Medium
Element Size	2.0 mm
Initial Size Seed	Assembly
Transition	Fast
Span Angle Center	Coarse
Mesh Based Defeaturing	On
Defeature Size	Default
Minimum Edge Length	31.4160 mm
Quality	
Check Mesh Quality	Yes, Errors
Error Limits	Standard Mechanical
Target Quality	1.e-003
Smoothing	Medium
Mesh Metric	None
Inflation	
Use Automatic Inflation	None
Inflation Option	Smooth Transition
Transition Ratio	0.272
Maximum Layers	5
Growth Rate	1.2
Inflation Algorithm	Pre
View Advanced Options	No

Figure 15: Analysis data and mesh functions of rear suspension components

1) *Semi-trailing Arm:* Initially, there were several modifications to be done at the mounting plate since the whole load was acting up there and an extra supporting plate was given to the mount to withstand the stresses.

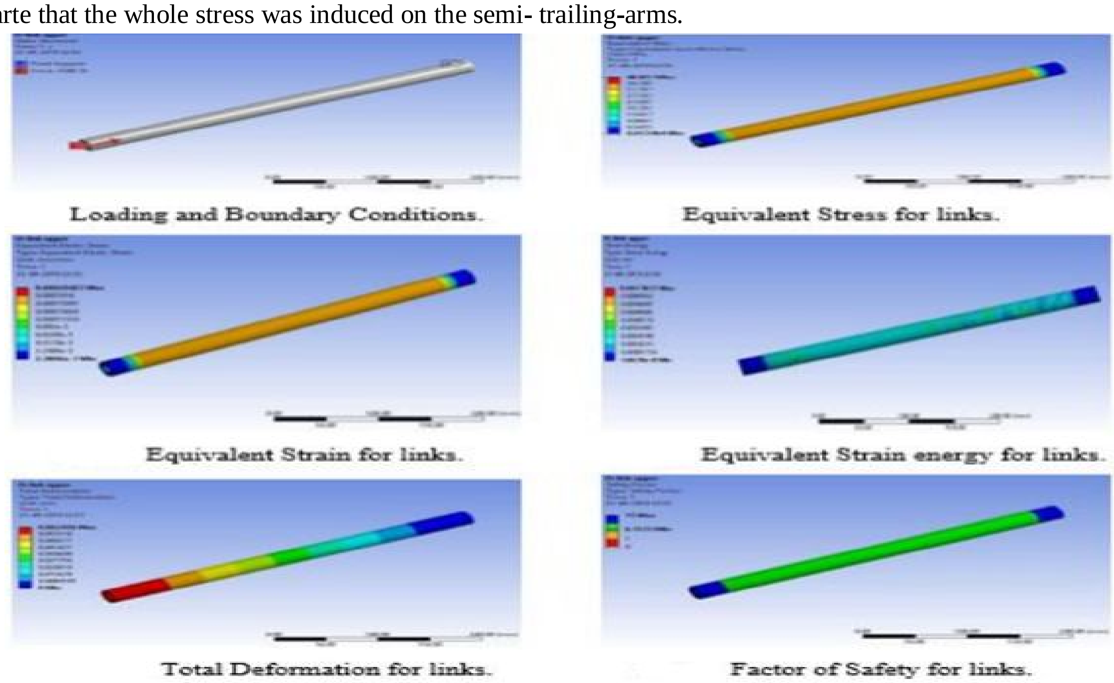


Loading and Boundary Conditions. Equivalent Stress of Semi-Trailing arm.
Equivalent Strain of arm. Equivalent Strain energy of arm.
Total Deformation of arm. Factor of Safety of arm..

Figure 16:. Analysis results of Semi-trailing arm.

Factor of safety was a bit less compared to the front suspension parts since it was a long beam, but it could easily take up the all the loads after adding those extra members in between the pipes.

2) *Upper and Lower Link:* The stresses observed were very minimal since these links only act as toe adjusters and results clearly demonstarte that the whole stress was induced on the semi- trailing-arms.



Loading and Boundary Conditions. Equivalent Stress for links.
Equivalent Strain for links. Equivalent Strain energy for links.
Total Deformation for links. Factor of Safety for links.

Figure 17. Analysis results of links.

The links turns up to have the highest F.O.S at some regions since there were very minimal forces acting upon it as it was an extra support given to the semi-trailing arm. The main purpose was added as it can be used for toe-adjustment.

VI. FABRICATION PHASE

A. *Design for Manufacturing and Assembly*

DFMA Plan was chosen for the Fabrication phase.

DFMA is a combination of two methodologies, One is designed for manufacturing and another one is the design of the assembly. Overview of design for manufacturing and assembly techniques, which are used to minimize product cost through design and process improvements.

DESIGN OF MANUFACTURING + DESIGN OF ASSEMBLY =
DESIGN FOR MANUFACTURING AND ASSEMBLY

1) *Design for Manufacturing*

DFM is concerned with reducing overall part production cost and minimizes the complexity of manufacturing operations.

The Main Objective of DFM are:

a) Estimate the manufacturing cost, so that we can reduce the cost.
b) Reduce the cost of components.
c) Reduce the costs of assembly.

2) *Design for assembly*

DFA is the method of design of product for ease of assembly. It is a tool used to assist the design teams in the design of products that will transition to productions at a minimum cost focusing on the number of parts, handling and ease of assembly. It is only concerned with reducing product.

The Main Objectives of DFA are:

a) Minimize part count.
b) Design parts with self-fastening features.
c) Minimize reorientation of parts during assembly.
d) Design of parts for reuse, handling, & insertion.
e) Emphasize ‘Top-Down’ assemblies.
f) Standardize parts(minimum use of fasteners).
g) Encourage modular design.
h) Design for a base part to integrate as many as other components.
i) Design for component symmetry for insertion.

B. *Fabrication*

Several jigs were designed and used as shown in the below figures to obtain high accuracy. Ensuring that the hard points of the vehicle doesn’t change. The clamps are been manufactured by laser cutting to have a proper notch which fits the frame of the vehicle.The sketch was designed over a cardboard and the required pipe lengths are cut and notched for a firm fit to avoid gaps. According to the designed sketch and clamps were used to fix on to it so that it wouldn’t encounter any disturbance after the welding operation is done. Using a long M10 size bolt to keep the hardpoints aligned in the position and the welding operation was performed. So, it wouldn’t get any disturbances in the structure of a-arm.The clamps were laser cut. Since, a millimeter change in the hardpoints leads to a huge change in the suspension geometry which would affect the vehicle performance.Tig welding was used for the welding operation since it has great strength and smooth finish over the AISI4130.

Figure 18: Setup of jigs, Fixtures for manufacturing feasibility.

In the assembly to the chasis high tension M10 bolts of SAE grade-10.8 were used with spring washers placed on either sides along with nylon locknuts.

Front Suspension system. Rear Suspension system.

Suspension system in CAD Model. Fabricated suspension system to Chassis.

Figure 19: Assembled suspension system to the chassis.

C. Design Failure Mode Effect and Analysis

Design Failure Mode Effect and Analysis (DFMEA) is a systematic group of activities used to determine (how to recognize and evaluate potential systems), products or process failures. DFMEA identifies the effects and outcomes of failures, actions that could eliminate or mitigate the failures and provides a historical written record of the work performed. To perform this test initially the particular component which is to be evaluated is considered. After the design is done, we need to find out the mode of the failure and a severity rating is given to it and the cause of that gives the occurrence value and the effect of that failure mode gives the detection and the risk priority number is calculated accordingly The standard chat of severity, occurrence and detection can be found in the reference. Analysing the mode, effect, cause of the failure the severity, occurren, detection is calculated and the risk priority number is calculated as follows:

1) *Severity:* For the components such as Springs,A-arms, links, trailing-arms, damper. The mode of failure is Surging/ buckling, bending & breakage, leakage of Oil. For these kind of failure mode the severity is aroud ~7.
2) *Occurrence:* For the components such as Springs,A-arms, links, trailing-arms, damper. The cause of failure is arised by excess loading, axial stress > yield stress, mechanical failure. For these kind of failure cause the occurence is aroud ~4.
3) *Detection:* For the components such as Springs,A-arms, links, trailing-arms, damper. The effect of failure is seen in damage to system/comfort, rough operation of vehicle / damages to system, damage to shocks. For these kind of failure observed the detection is aroud ~3.

Risk Priority Number (R.P.N) = Severity*Occurrence*Detection = 7*4*3 = 84.

Here comes the main analysis which is done by making the necessary changes to increase the detection value where certain precautions are taken to avoid that particular mode of failure by which the occurrence and detection value comes down. Simultaneosuly the calculated risk priority number comes down.

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D. Design Validation and Plan

Design validation is the process of evaluating the software during or at the end of the product development, to ensure the manufactured system satisfies the specification in the end-user application or product and environment.
The below flow chart shows the validation process:

Requiement→Input→Output→Product
↕ ↕
Design Validation.

For the parameter such as camber, caster, toe and king pin inclination. The acceptance criteria considered was by simulating in several suspension software's & analysing them by virtual-compilance test method validation.
For the parameter such as a-arms, semi-trailing arms and links. The acceptance criteria considered was by performing stress analysis which is done using various analysing softwares where the factor of safety is used in validation.
For the parameter such as drop test. The acceptance criteria considered was to check the suspension components were in proper condition or not after the test where validation method was to performed by dropping the vehicle from a height of 2meters.

VII. RESULTS AND DISCUSSION

A. Suspension Travels

Since the length of the shock was only 16.2 inch then the angle correction factor was designed to 30° and stiffness, the motion ratio was set accordingly.
Front Motion Ratio: 0.63
Rear Motion Ratio: 0.54
From the above motion ratios, the suspension travel achieved was 8" for front and rear. The front was kept at 50PSI and rear was set to 70PSI and the below graph shows the suspension travel corresponding to the force acting on it.

B. Wheel Travels V/S Geometry changes

This was the main stage where we concentrated and worked on for a longer period of time and for which the results came out really well as thought of the below tables of the front and rear suspension gives the bump travel w.r.t suspension geometry.

Front Suspension V/S Bump Travel.

FRONT SUSPENSION - BUMP TRAVEL
RHS WHEEL (+ve Y)

TYPE 1 Double Wishbone, damper to lower wishbone

INCREMENTAL GEOMETRY VALUES

BUMP TRAVEL (mm)	CAMBER ANGLE (deg)	TOE ANGLE (deg)	CASTOR ANGLE (deg)	KINGPIN ANGLE (deg)
-50.00	0.7770	-0.2763	2.9982	9.2113
-25.00	0.4163	-0.1116	3.0012	9.5632
0.00	0.0000	0.0000	3.0049	9.9737
25.00	-0.4692	0.0747	3.0093	10.4389
50.00	-0.9917	0.1237	3.0144	10.9589
75.00	-1.5702	0.1552	3.0205	11.5358
100.00	-2.2098	0.1762	3.0275	12.1742
125.00	-2.9174	0.1926	3.0358	12.8810
150.00	-3.7027	0.2109	3.0456	13.6654

Rear Suspension V/S Bump Travel.

REAR SUSPENSION - BUMP TRAVEL
LHS WHEEL (-ve Y)

TYPE 10 Trailing Arm, upper and lower rear links

INCREMENTAL GEOMETRY VALUES

BUMP TRAVEL (mm)	CAMBER ANGLE (deg)	TOE ANGLE (deg)	CASTOR ANGLE (deg)	KINGPIN ANGLE (deg)
-50.00	0.8728	-0.9421		
-25.00	0.4215	-0.4017		
0.00	0.0000	0.0000		
25.00	-0.4088	0.2668		
50.00	-0.8208	0.4022		
75.00	-1.2519	0.4093		
100.00	-1.7180	0.2912		
125.00	-2.2354	0.0509		
150.00	-2.8217	0.0080		

Figure 20: Wheel Travel with respect to suspension geometry.

Achieving the castor, kingpin inclination with a minimal change of 1° from its mean value which was kept almost constant throughout the travel, brought out good results for the front suspension providing the proper feedback and steering effort same all the time.
Here the camber and toe changes were set according to the roll centres of the vehicle and made sure that the vehicle doesn't roll over or get toppled in the travels. We designed our camber angle accordingly where initially it was kept 0° where it was tending to accelerate in a better way and calculating the cornering force it was designed to achieve a min of 2° negative camber where the vehicle doesn't roll over and all set for faster cornering's.

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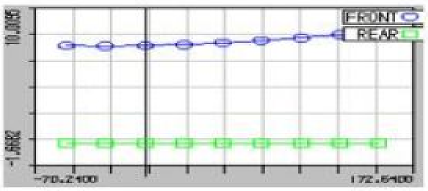
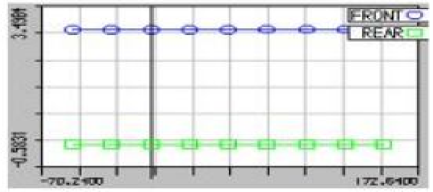
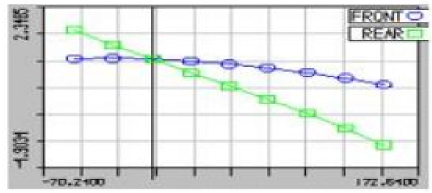


Bump Travel V/S King Pin Inclination.



Bump Travel V/S Castor.

Bump Travel V/S Toe Angle.



Bump Travel V/S Camber Angle.

Figure 21: Graphical results of suspension parameters.

C. Finite Element Analysis of the Components

The FEA results are tabulated below and it is observed that the factor of safety of all components are above 1.5 which indicates that the components can withstand the loads and sustain the rough-terrain.

Component	Maximum (von –mises) stress (MPA)	Max Equivalent Strain (mm/mm)	Maximum deformation (mm)	Strain Energy (mJ)	Factor of safety
Lower A-arm	215.47	0.0010775	0.14283	0.11936	2.13
Upper A-arm	272.56	0.0015217	0.68151	0.076669	1.68
Semi-Trailing Arm	105.99	0.0005301	0.1659	0.061334	2.35
Upper and Lower Link	40.827	0.00020427	0.062456	0.007303	6.12

Table 1: Finite Element Analysis Results.

VIII. TESTING PHASE

The vehicle was initially driven into a muddy terrain then slowly into small bumps, logs, Cement blocks were taken on and when it sustained well, we even took testing to severe-level by taking sharp turns, driving from steps, landing on a single wheel and some of the images were captured and shown below.

Getting Down the Steps.

Taking Sharp Fast turns.

Bump Travel overcoming blocks.

Droop Travel overcoming blocks.

Figure 22: Testing in various scenarios.

IX. CONCLUSION

At the end of the day, the team has developed a unique design for each and every component that is designed, analysed and simulated for all kind of situations making it suitable for the rough terrain. For the creation of this design, specification and ability to meet several computer-aided drafting, analysis, testing & development, manufacturability, serviceability, system integration is done and the vehicle is evaluated on how it works. Each of the parts developed by the team is evaluated using various validation plans, constant testing and refinement were done using DFMEA. From the support of our college, this work made our team to achieve our suspension system fabricated as designed and with the intent of performing well in all the rough terrains.

X. SCOPE FOR FUTURE WORK

A. By increasing the CV shaft articulation in rear side, the suspension travel can be much more increased and Ground clearance can be increased.
B. Choosing bigger shock absorbers and designing with much better angle correction factor.
C. Much effective design with better motion ratios gives out much more wheel travels.
D. Optimization of the components.
E. More different suspension systems can be studied and worked which may increase the performance such as adaptive suspension system with electronic components integrated to it.

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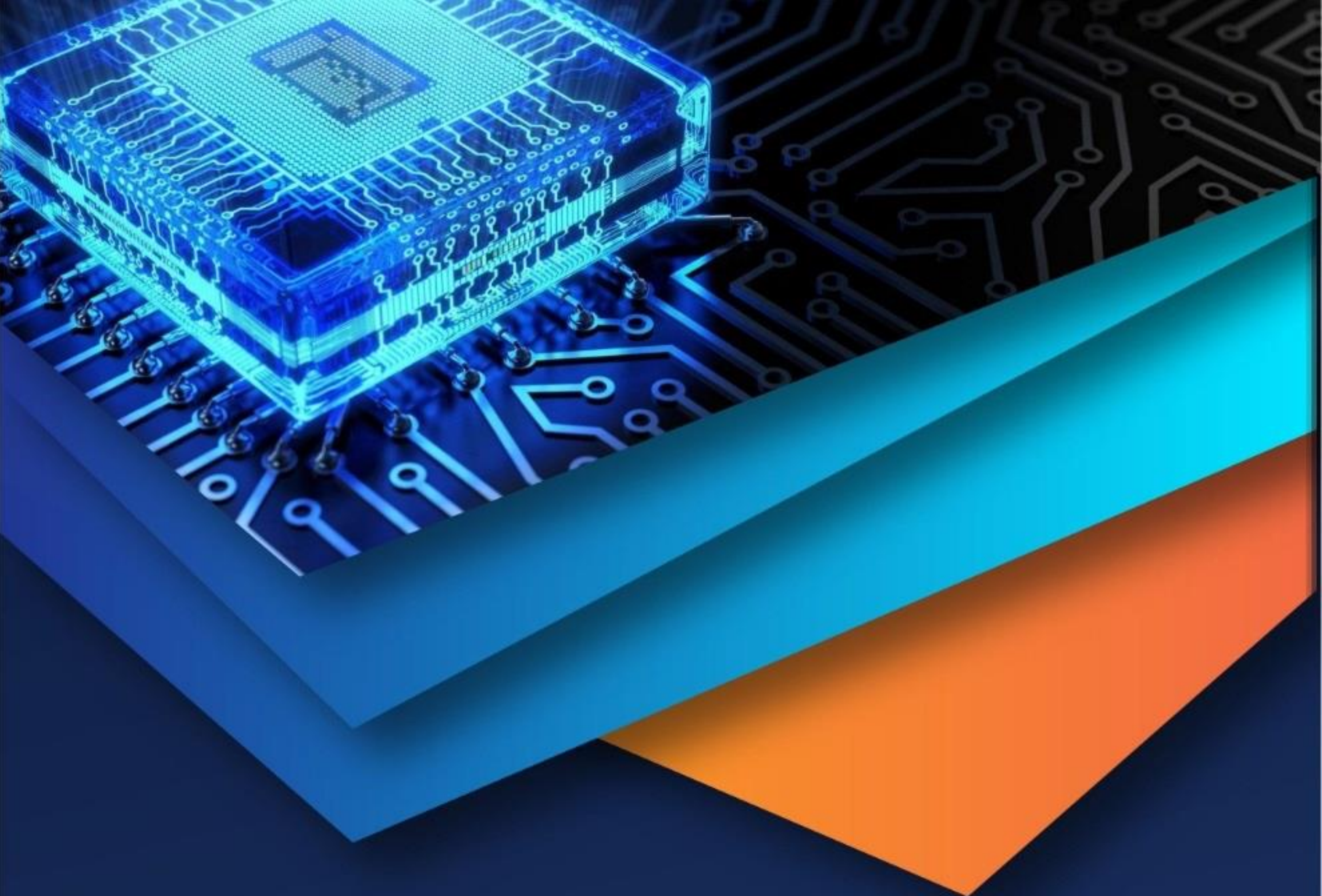

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