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Designing and Selection of Suspension Geometry and Shocks

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Abstract: This paper focuses on the designing of suspension geometry for and UTV. It also refers to the selection of shocks according to the application considering the vehicle parameters. Considering the important factors like camber, castor, wheel travel, etc a detailed procedure regarding this has been mentioned below. The vehicle is going to travel an off road, dealing with various troughs and crests with extreme cornering. We will be focusing on the smoothness of vehicle while traversing through such terrains.

The geometry is been designed in CATIA and SOLIDWORKS and static analysis of which is done in ANSYS. The purpose of the analysis is to ensure the design is safe at extreme load and abnormal impacts. The dynamic analysis of the vehicle is done in LOTUS software.

Thus designing of suspension parameters like A-arms and H arms, camber link, knuckle and various parameters like motion ratio, FVSA, etc are referred in the context.

Keywords: Camber, castor, CATIA, ANSYS, SOLIDOWORKS, LOTUS, Suspension geometry.

I. INTRODUCTION

The vehicle which is going to undergo off road conditions has a vital factor of comfort and smoothness of ride while traversing. The suspension geometry is designed in a manner where in the driver feels lesser impacts at various hurdles.

The suspension system is categorized into different types of geometry such as:

1) Conventional suspension system
2) Independent suspension system
3) Hydro elastic suspension system
4) Air suspension system

Thus considering independent suspension system better suit for this vehicle the procedure below relates to it. The independent suspension system is narrowed down to some of the geometries that are used in the market. The geometries are distinguished according to the linkages and mountings of the joints. Some of the systems are listed below.

A. Front Suspension Systems

1) Multi-link suspension system
2) Macpherson strut system
3) Double wishbone system
4) A arm suspension system

B. Rear Suspension System

1) Semi trailing suspension system
2) Trailing suspension system
3) H arm geometry

Thus, considering the requirement and simplicity of the geometry thus paper works on the double wishbones and H arm geometry. This basically consists of arms (linkages) ball joints, shock mountings and knuckle designing. Lower weight and easy manufacturing are some advantages of these systems.

II. OBJECTIVES

Considering various aspects that deal with suspensions, objectives are based on the design, analysis and execution of

A. To maintain wheel and road contact while turning
B. To have Optimum wheel and spring travel
C. To reduce unsprung mass of the vehicle
D. Over steer vehicle for terrain-based conditions.

III. SUSPENSION PARAMETERS

1) *Sprung Mass:* 420 kg
2) *Unsprung Mass:* 50 kg
3) *Total Weight:* 420 + 50= 470 kg (with driver)
4) *Kerb Weight:* 470-120 = 350 kg(without driver)
5) (Driver and navigator weight is considered as 60kgs)
6) *Front Track Width:* 58''inches
7) *Rear Track Width:* 60''inches
8) *Static Ride Height:* 10''inches
9) *Tire Diameter:* 25"inchES
10) *Wheel Travel Front:* 10 inches (4 inch bump, 6 inch droop)
11) *Wheel Travel Rear:* 8 inches (5inch bump, 3 inch droop)
12) *Front Spring Travel:* 6 inches
13) *Rear Spring Travel:* 4 inches
14) *Motion Ratio:*
15) *Front:* 6"/10"=0.6
16) *Rear:* 4"/8"=**0.5**
17) *Camber change Rate:*

CCR= TAN-1 (1/FVSA)

FRONT CCR

CCR= TAN-1(1/67.78)

= 0.84

REAR CCR

CCR= TAN-1(1/85.27)

= 0.67

Table I. Front suspension parameter

PARAMETER	VALUE
Camber Angle	-2.0^0 (Bump) -0.9^0 (droop)
Wheel travel	10 inches
Static ride height	8 inches
Motion Ratio	0.6

Table II. Rear suspension parameter

PARAMETER	VALUE
Camber Angle	-2
Wheel travel	8 inches
Static ride height	6 inches
Motion Ratio	0.5

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A. Front and Rear RCH Diagrams

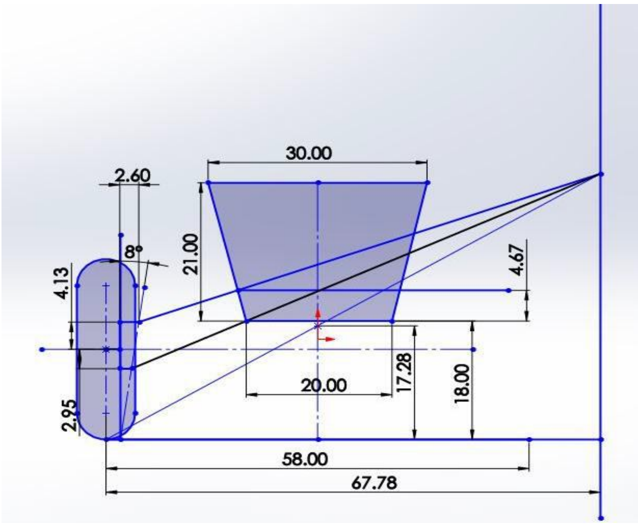

Fig. 1 Front RCH Diagram

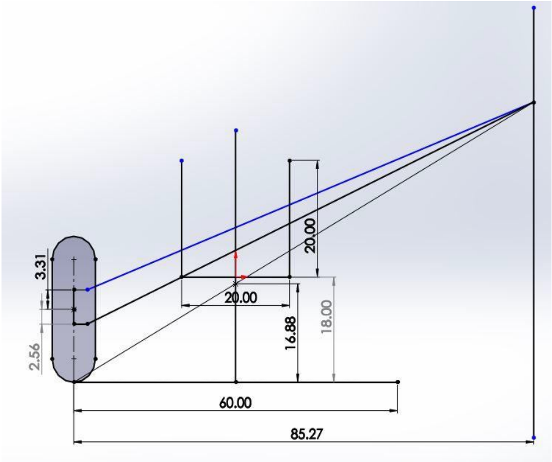

Fig. 2 Rear RCH Diagram

IV. DESIGN METHODOLOGY

The overall purpose of a suspension system is to absorb impacts from coarse irregularities such as bumps and distribute that force with least amount of discomfort to the driver. We completed this objective by doing extensive research on the front and rear suspension arm's geometry to help reduce as much body roll as possible. Proper camber and caster angles were provided to the front wheels. The shocks will be set to provide the proper dampening and spring coefficients to provide a smooth and well performing ride.

TABLE III
suspension parameters

Camber angle	-2.0
Castor angle	6
Toe In	5
KPI	8 degree
Ride Height	8 inch
Scrub radius	2.33 inch

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A. Material Used

A arms and H arms	AISI 4130
Knuckles	EN 19

V. FRONT SUSPENSION

The front suspension consists of an SLA double wishbone with mounting of strut to lower wishbone. The tire needs to gain negative camber in a rolling situation, keeping the tire flat on the ground.

A. Modelling of Front Knuckle

Fig. 3 Front Knuckle Model in SOLIDWORKS

B. Analysis of Front Knuckle

Forces applied

Braking	1.9M N-mm
Suspension	5G
Cornering	4G

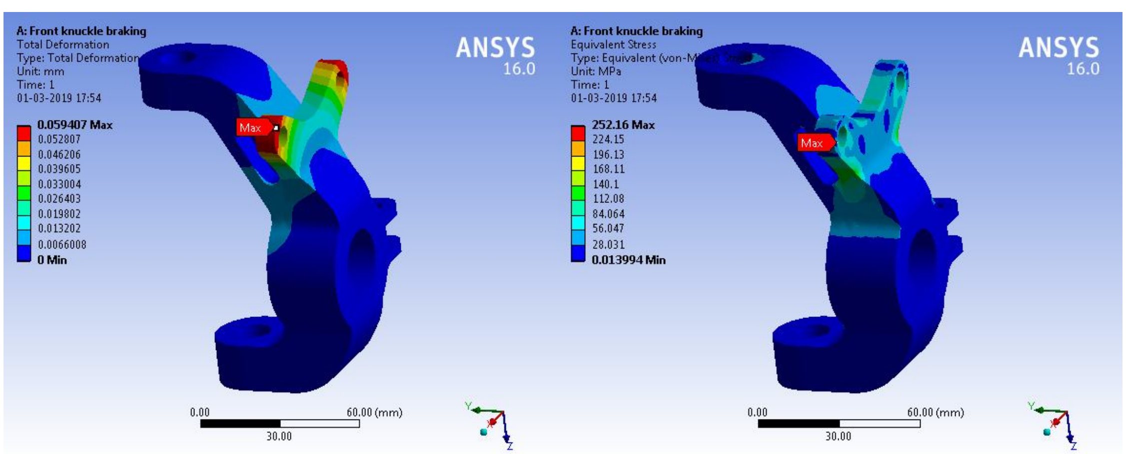

Fig. 4 Stress Analysis and Total Deformation of Front Knuckle while Braking

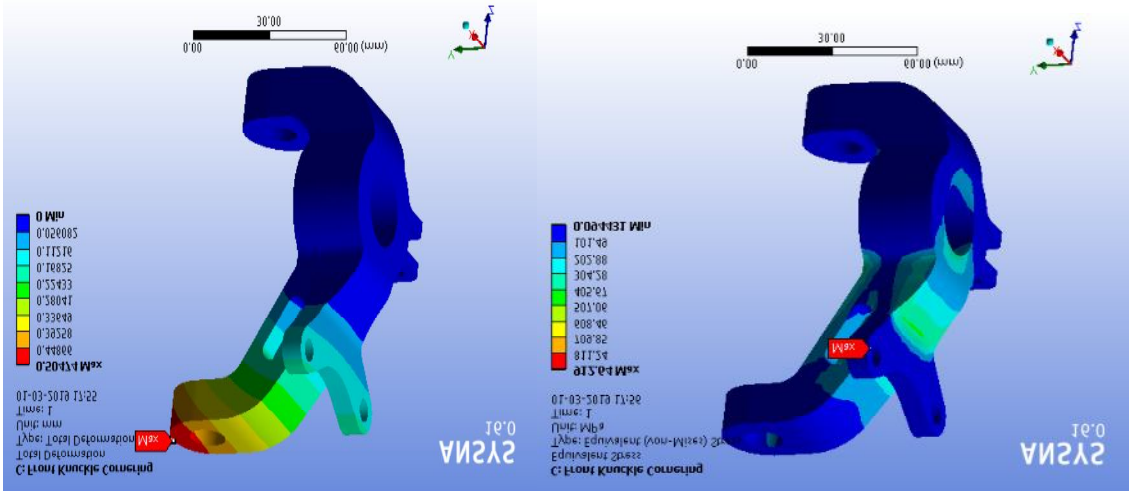

Fig. 5 Stress Analysis and Total Deformation of Front Knuckle while Cornering

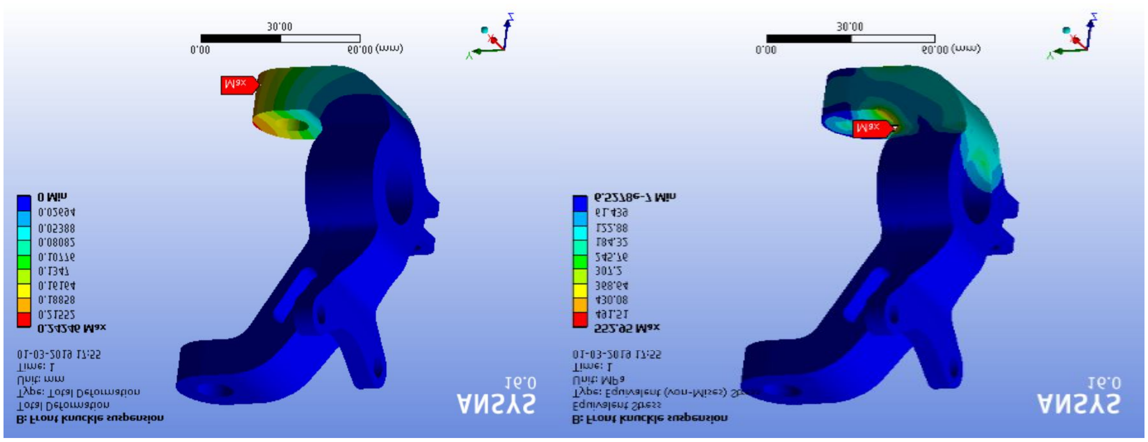

Fig. 6 Suspension Deformation and Suspension Stress

C. Modelling of A-Arms

Fig. 7 Front Wishbone Design in SOLIDWORKS

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D. Analysis of A-Arms

LOAD	5G
MAX STRESS	964.09
Total deformation	0.68195

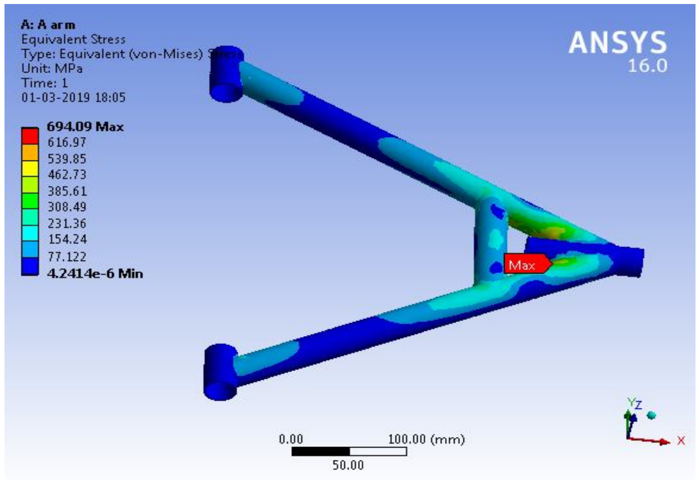

Fig. 8 Front Wishbone Stress Deformation

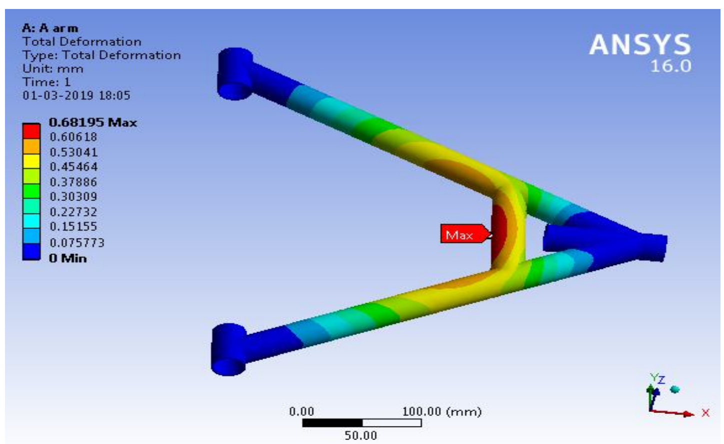

Fig. 9 Front Wishbone Total Deformation

VI. REAR SUSPENSION

A. Rear Knuckle Modelling

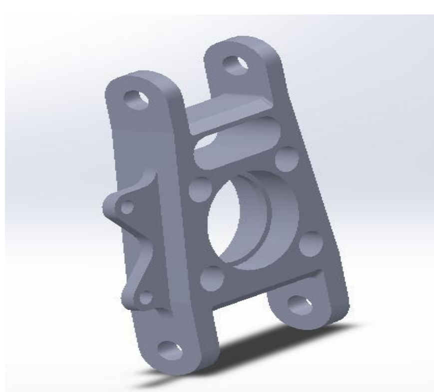

Fig. 10 Rear Knuckle Design in SOLIDWORKS

B. Rear Knuckle Analysis

Force applied

Braking	4.3M N-mm
Suspension	5G
Cornering	3G

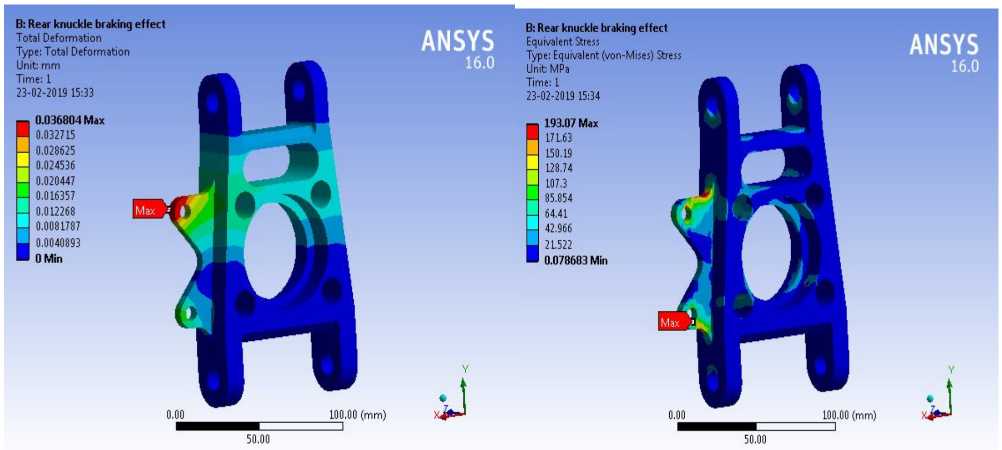

Fig. 11 Total Deformation and Total stress analysis of Rear Knuckle While Braking

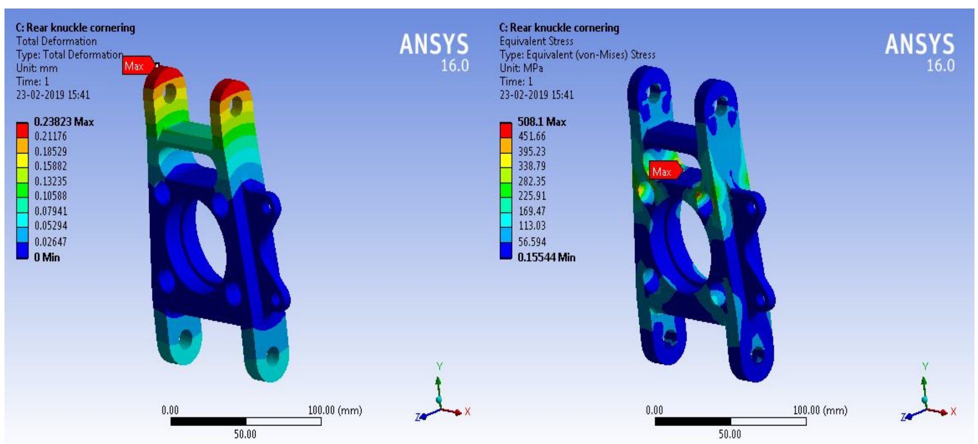

Fig. 12 Total Deformation and Total stress analysis of Rear Knuckle While Cornering

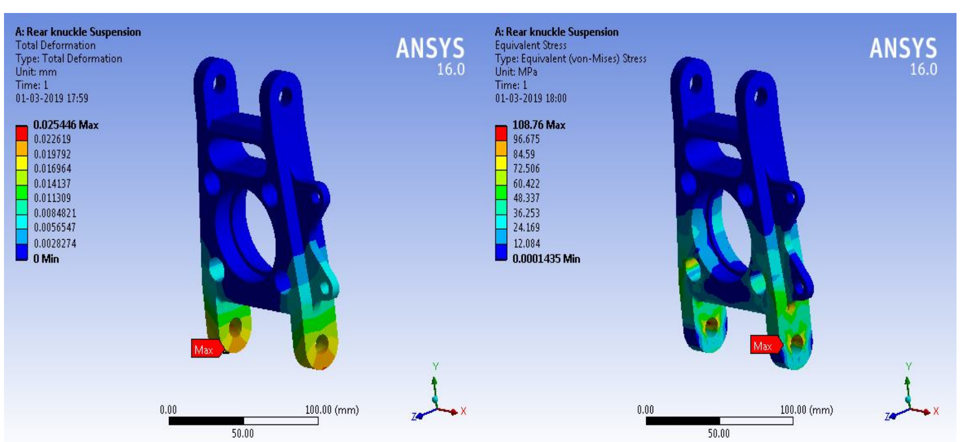

Fig. 13 Total Deformation and Total stress analysis of Rear Knuckle Due to Suspension

C. Modelling of Rear Arms

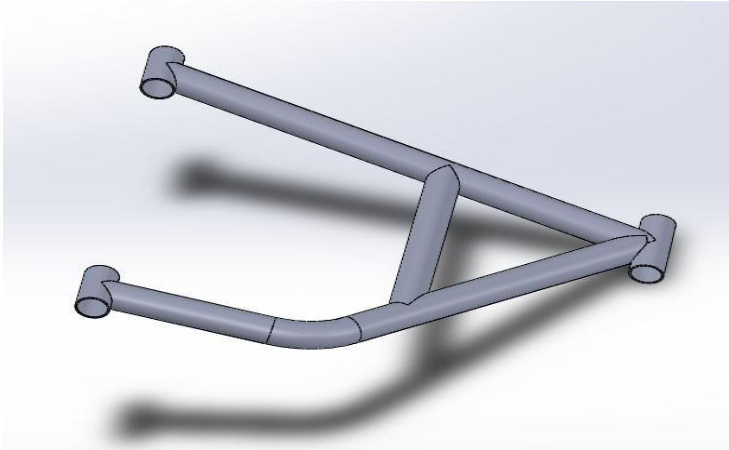

Fig. 14 H-Arm Design in SOLIDWORKS

D. Analysis of H-Arms

Forces Applied

Load	3G
Stress deformation	927.66
Total deformation	2.30447 mm

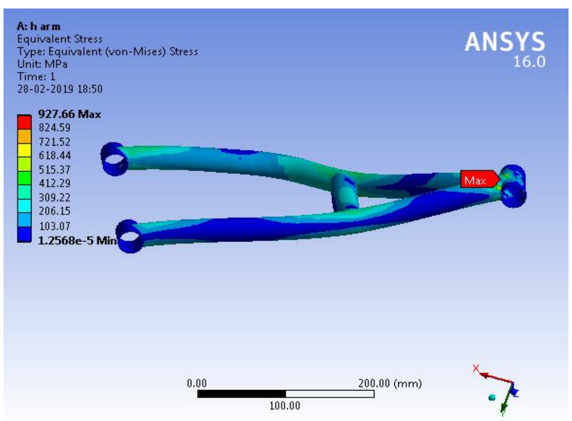

Fig. 15 Total Stress Analysis of H-Arms

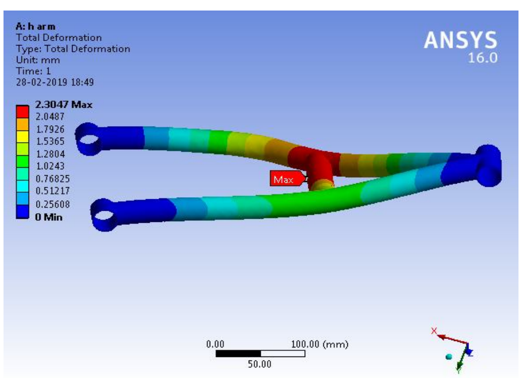

Fig. 16 Total Deformation of H-Arms

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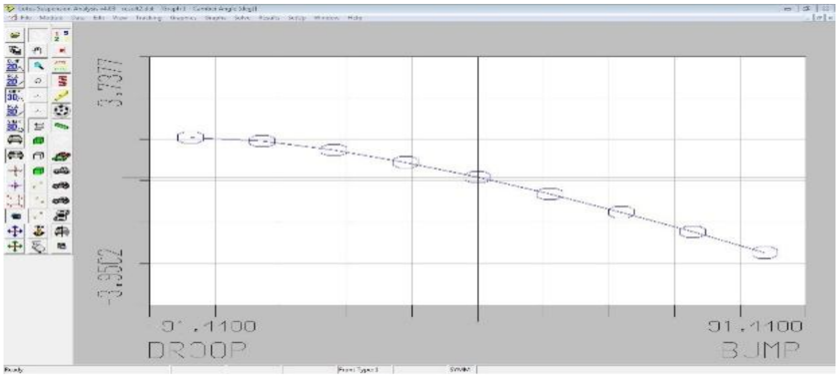
VII. LOTUS ANALYSIS OF FRONT AND REAR SUSPENSION

Lotus simulation software is software in which we can create our vehicle to simulate and transverse while applying the dynamic conditions in 3D state. The software thus gives us a idea of the how will the vehicle parameters like camber and toe change while considering the jounce, droop and jounce it undergoes while simulation. Thus, we can check the simulation for 3D roll, 3D steer 3D bump and so on. According to the results, and the objective we set for the team we are certainly to our goals. Below are the images of the analysis of caber and toe change while simulation.



. Fig. 17 Simulation of Camber Change in LOTUS

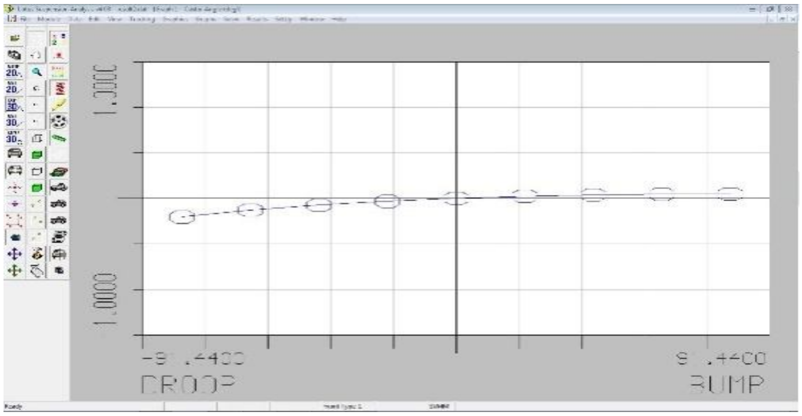
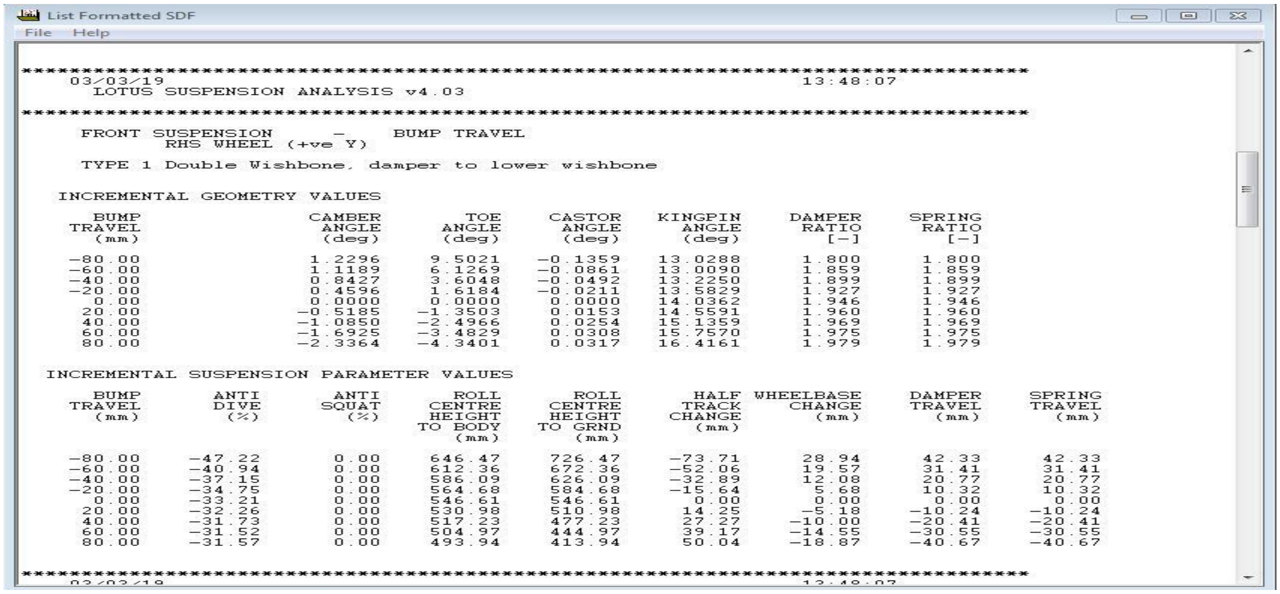


Fig. 17 Simulation of Castor Change in LOTUS

A. Results from LOTUS Software



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LOTUS SUSPENSION ANALYSIS v4.03

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)	DAMPER RATIO [-]	SPRING RATIO [-]
-80.00	1.2296	9.5021	-0.1359	13.0288	1.800	1.800
-60.00	1.1189	6.1269	-0.0861	13.0090	1.859	1.859
-40.00	0.8427	3.6048	-0.0492	13.2250	1.899	1.899
-20.00	0.4596	1.6184	-0.0211	13.5829	1.927	1.927
0.00	0.0000	0.0000	0.0000	14.0362	1.946	1.946
20.00	-0.5185	-1.3503	0.0153	14.5591	1.960	1.960
40.00	-1.0850	-2.4966	0.0254	15.1359	1.969	1.969
60.00	-1.6925	-3.4829	0.0308	15.7570	1.975	1.975
80.00	-2.3364	-4.3401	0.0317	16.4161	1.979	1.979

INCREMENTAL SUSPENSION PARAMETER VALUES

BUMP TRAVEL (mm)	ANTI DIVE (%)	ANTI SQUAT (%)	ROLL CENTRE HEIGHT TO BODY (mm)	ROLL CENTRE HEIGHT TO GRND (mm)	HALF TRACK CHANGE (mm)	WHEELBASE CHANGE (mm)	DAMPER TRAVEL (mm)	SPRING TRAVEL (mm)
-80.00	-47.22	0.00	646.47	726.47	-73.71	28.94	42.33	42.33
-60.00	-40.94	0.00	612.36	672.36	-52.06	19.57	31.41	31.41
-40.00	-37.15	0.00	586.09	626.09	-32.89	12.08	20.77	20.77
-20.00	-34.75	0.00	564.68	584.68	-15.64	5.68	10.32	10.32
0.00	-33.21	0.00	546.61	546.61	0.00	0.00	0.00	0.00
20.00	-32.26	0.00	530.98	510.98	14.25	-5.18	-10.24	-10.24
40.00	-31.73	0.00	517.23	477.23	27.27	-10.00	-20.41	-20.41
60.00	-31.52	0.00	504.97	444.97	39.17	-14.55	-30.55	-30.55
80.00	-31.57	0.00	493.94	413.94	50.04	-18.87	-40.67	-40.67

Fig. 18 Computer Generated Data from LOTUS Software

Thus, the results state the camber the vehicle will have while traversing through the terrains. This indeed explains if the vehicle sustains the motion of suspension at extreme dynamic conditions. Thus we found out that this rate is acceptable for us depending on what we had assumed, which was around (1-2 degree) of camber change. Thus, we came to the conclusion that we have achieved our objectives.

VIII. SELECTION OF SUPENSIONS

Steps to identify which suspension is better suited:
MR- Motion ratio
Sr – Spring rate
Rf – ride frequency
Ms – sprung mass

1) Motion ratio = Wheel travel / spring travel
   Front
   Motion ratio = 6 / 9 = 0.66
   Rear
   Motion ratio = 4/ 8 = 0.5

2) Spring rate = $4*\pi^2*Rf^2*Ms* MR^2$
   Front (Rf = 3.25)
   $4* \pi^2*(3.25)^2* (40)*(0.6)^2$
   = 27.02 N/mm
   Rear (Rf=2.6)
   $4* \pi^2*(2.6)^2* (45)*(0.5)^2$
   = 29.08 N/mm

3) Wheel rate = Spring rate / MR^2
   Front = 27.02 / 0.6^2 = 7.5 N/mm
   Rear = 29.08/ 0.6^2 =11.6 N/mm

4) Roll Rate = π (TW)*$(WR)^2$ / 180*(WR+WR)
   Front = π (58)*$(7.5)^2$ / 180*(7.5+7.5) = 379.60 Nm /deg
   Rear = π (64)*$(11.6)^2$ / 180*(11.6+11.6) = 647.8 Nm / deg

5) Critical damping = sqt(4*Sr * Ms)
   Front = 2317.05 kg/ sec
   Rear = 2664.24 kg/ sec

6) Damping Coefficient = Critical Damping * Damping Ratio
   = 2317.05 * 1.5 =3475.36 kg / sec (front)
   = 2664.24 * 1.3 = 3623.12 kg/ sec (rear)

7) Damping force = Damping Coefficient * Velocity @0.3
   = 3475.36 * 0.3 = 1024.25N (front)
   = 3623.12*0.3 = 1086.23 N

8) Damper stroke = Damping Force / Sr
   = 1024.5 /270.2 = 3.78 inches (Front)
   = 1086.24 /290.8 = 4.1 inches (Rear)

Thus, considering the Damper stroke that we required below suspensions were considered suitable.
The Polaris RZR 570 has been our priority from the start. The suspension has shown us a good performance still the comparison remains.

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Table IV. Suspension Selection

SUSPENSIONS CONSIDERED			
SR.NO.	MODEL OF VEHCILE	MODEL NO.	SPECS
1	RZR S 800 REAR	7043419	SHOCK FOX PODIUM(Piggyback disc and rebound adjustment) 21.43" (EXT)* 6.22" (stroke)-11.5" body
2	RZR XP 900 REAR	70437794	SHOCK FOX PODIUM(Piggyback disc only) 22.7" (EXT)* 7.6" (stroke)2.5" body
3	RZR 570 REAR	7043759	SHOCK SACHS 19.0" (EXT)*6.7" (Stroke)-2.0" body. Adjustable spring

The RZR 570 rear was selected due to the low budget. But this won't hamper our performance as the 19 inches shocks has been used for a vehicle which has a weight of 600kg which is a Polaris model. Thus, giving us a idea how much the shocks can sustain. Even the length seems to be optimum for our vehicle. Thus RZR 570 rear SHOCKS will be used for our vehicle.

IX. ACTUAL ASSEMBLY

A. A-Arms

Fig. 19 Actual Manufactured A-Arms

Fig. 20 Actual Rear Assembly

Fig. 21 Actual Front Assembly

X. CONCLUSIONS

Thus, identifying all the factors that were required to build the geometry, we have built our suspension system. Thus, modelling and analysing that CAD models we manufactured the structure. The objectives which we laid down are accomplished as stated in the lotus results, describing the castor and camber change thus functioning of the vehicle, following with optimum oversteer.

Completing the dynamic analysis, the OEM shocks were selected for the vehicle which satisfied our requirement. We have achieved lower unsprung mass of the vehicle. Achieving the targets set the study has been made regarding the suspension system for a two-seater utility task vehicle.

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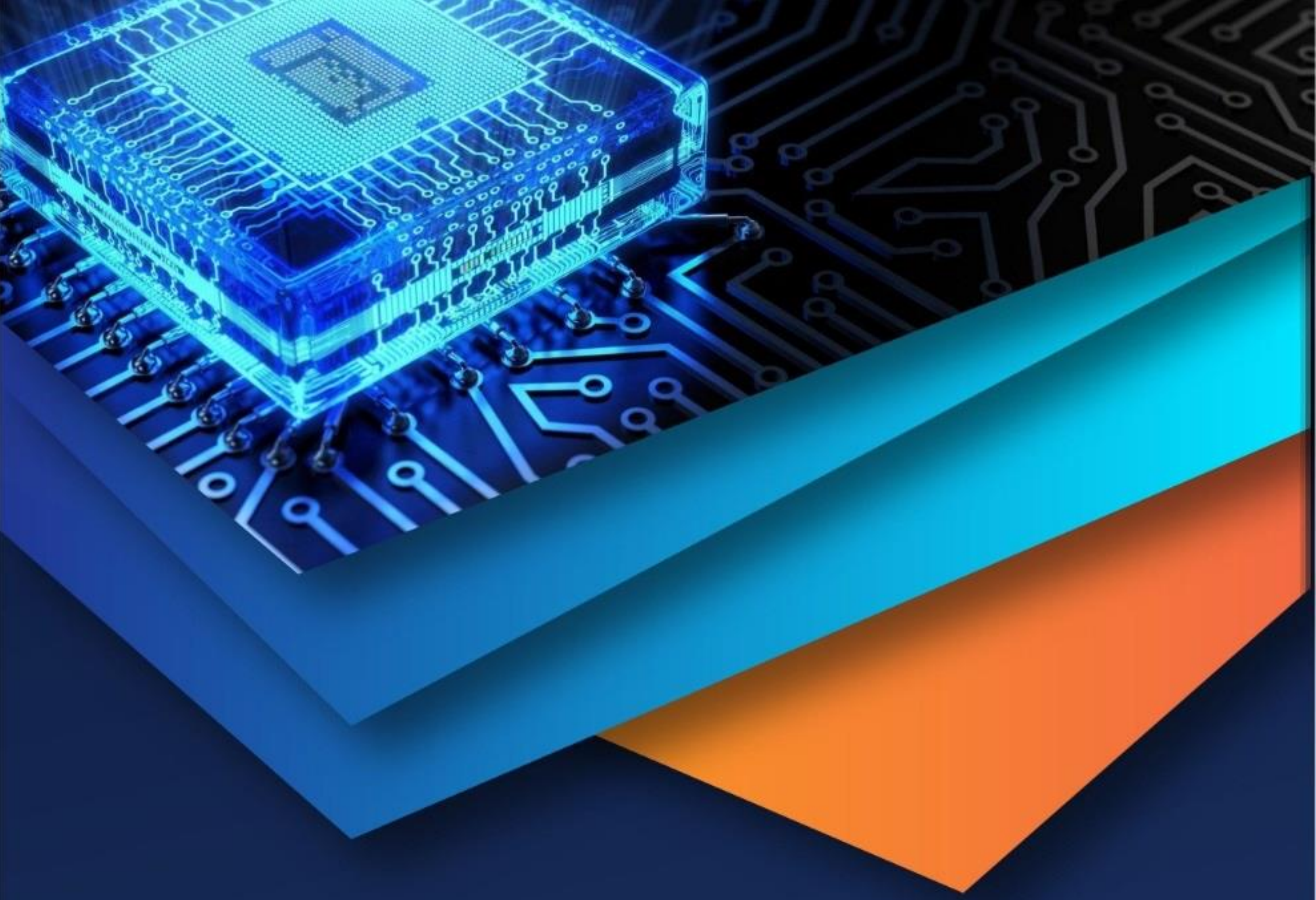

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