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OPTIMIZATION OF SUSPENSION

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Abstract— Body roll is one of the most common phenomenon that occurs in a car during high speed turnings on curve roads. Hence to avoid any kind of tilting in the vehicle body, while it is turning, he reaction forces due to centrifugal forces have to be negated. In this paper we will discuss about three mechanisms which can counter body roll and assist in efficient cornering of the vehicle thereby providing safety to the passengers and enhance better steering control.

Keywords— Rollover, Centrifugal force, MR Suspension, Toggle Mechanism, Hydro pneumatic suspension

I. INTRODUCTION

On a vehicle, the weight shift towards the outer wheels during turning is called body roll. When a suspension system is attached to the vehicle, it provides stability and absorbs shocks from roads. In the most basic level, it consists of a spring- mass damper system. Usually a suspension system consists of wheels, tires, springs, dampers, linkages, bushings, bearings, and steering system. The first workable suspension spring started with the rise in industrialization. Obadia Elliot was the first person to patent the spring suspension system. Each wheel had stable, strong and durable leaf springs and the horse carriage was attached to it. This started to find fame and so began the era of suspension. Leaf springs have been around the early Egyptians. Ford Model T also had the above mentioned suspension, which is being used by heavy vehicles even today. This was the first modern suspension system and advances grew with the improvement of road construction. The Americans introduced leather springs that swung and did not take jolts unlike the leaf springs in horse carriages. Coil springs first appeared in a production vehicle of Brush Motor Company which has been seen in most cars since it was introduced in 1906.

In 1920, Leyland motors used torsion bars as suspension. In 1922, independent suspensions were produced by Lancia Lambda. Today most cars have independent suspension. The mechanisms proposed in order to counter body roll over are:

1. MR suspension with tilting column system
2. Toggle mechanism controlled suspension with MR assistance system
3. Anti-roll Hydro pneumatic suspension system

II. MR SUSPENSION WITH TILTING COLUMN SYSTEM

In this design we might we made use of tilting column that could tilt cars at high speed while cornering so that the vehicle can be more stable and provide better steering control to the driver. This tilting column was designed to be placed on the Upper mounts of a double wishbone suspension so that the suspension actuation can be effectively controlled. This actuation of the tilting column was with the help of a DC stepper motor placed at the center of the column. Usage of an MR suspension which uses the electromagnetism to assist the damping.

MR fluid makes the suspension more adaptive and responds according to the given condition unlike the convention unlike the conventional suspension. A gearing system was also incorporated to provide suitable torque in order to counter road reactions and make the systems work efficiently.

III. TOGGLE MECHANISM CONTROLLED SUSPENSION WITH MR ASSISTANCE SYSTEM

Based on the previous calculation we worked on making use of the MR fluid effectively with the help of a toggle mechanism controlled suspension. The main aim is to actuate the toggle mechanism with the help of a motor, rack and pinion gear system. Toggle mechanism works on the application of large force through short distance and hence it provides a very good mechanical advantage thereby reducing load on the motor. This further helps in reducing the amount of force required by the motor to actuate toggle mechanism action. When toggle mechanism is used it multiplies the force by 4 up to 16 times depending upon the requirement of the force. For our mechanism we require a force multiplication of 4 times considering the factor of safety. Also with the usage of rack and pinion system we get a force and torque multiplication of 2-3 times.

ANTI ROLL HYDROPNEUMATIC SUSPENSION

Based on the principles of the fluid suspension. It consists of a spherical accumulator with a nylon membrane which consists of fluid and gas with a diaphragm separating it. In a few models it's been seen that the gas used is nitrogen at 75 Mpa pressure. The sphere here is an accumulator. In this model it has accumulators on both suspension at the rear transversely. There are cross-linked with circular pipes with the pipes begin at the top of right suspension and joins at the bottom of the other suspension and similarly the other pipe has been attached. The pipes consist of fluid that is the damper fluid. The top part of the damper is where the pipes originate from the right suspension and ends at the bottom of the damper of the left suspension.

During turning of the vehicle, the inner wheel damper moves up and the outer wheel damper moves downwards. The volume in the inner damper increases and the volume in the outer wheel damper decreases. There is a pressure of 7.27Mpa at the inner wheel and 17.46Mpa at the outer wheel developed. At normal or straight road conditions, the pressure is 9.47Mpa. Therefore to bring it to a point where there is no roll angle being provided the pressure has to be equalized to 9.47Mpa. This is achieved by taking advantage of the pressure difference.

V. CALCULATIONS

For this calculation, we have considered three different values of speeds: 50kmph, 100kmph and 150kmph. Since centrifugal force is the same for all these speeds at critical turning radius, calculations for only 50kmph (13.88 m/s) is shown. The specifications of Nissan Leaf is considered for this calculation. Considering the forces that act on the body while tilting are centrifugal force while turning resulting in reaction forces.

$$\mathrm{CF} = \frac{\mathrm{mv}^2}{R}$$
$$= \frac{[1520\times(13.88^2)]}{19.416}$$
$$= 15.082 \text{ KN}$$

The height $\mathrm{h} = 0.8$ m and diameter $\mathrm{d} = 1.62$ m

$$\text{Reaction:} = \frac{15.028*0.8}{1.62} = 7.421 \text{ KN}$$

For left turn:

Inner wheel (left)

$$\mathrm{R} = \mathrm{W} + \mathrm{Mr} + \mathrm{A}$$
$$6.726 = 3.444 + 1.2 + \mathrm{A}$$
$$\mathrm{A} = 2.082 \text{ KN}$$
$$\mathrm{A} = 2082 \text{ N}$$

Outer wheels (right)

$$\mathrm{R} + \mathrm{W} = \mathrm{Mr} + \mathrm{B}$$
$$6.726 + 3.444 = 1.2 + \mathrm{B}$$
$$\mathrm{B} = 8970 \text{ N}$$

For steering,

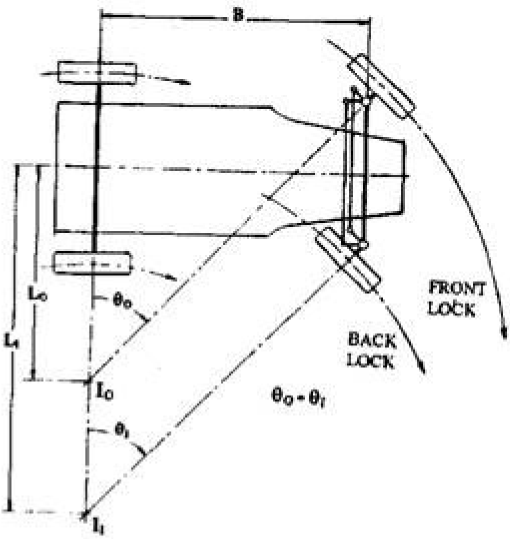


Fig V.I: Ackermann Steering

Using Ackermann steering formula, the radius of curvature of the road affects the steering angle. This radius of curvature is used in the centrifugal force formula used above.

$$R = \frac{b}{\sin\theta}$$

Where, R - Radius of curvature
b - Wheelbase
θ - Angle turned by inner wheel
considering various values of R, θ is calculated

$$11\text{m} = \frac{2.7}{\sin\theta i}$$
$$\theta i = 14.208°$$
$$a = 1.62 \text{ m}$$

14.208° is turned maximum by inner wheel.
Therefore,
For every 1̊ turn in steering angle, inner wheel rotates by 0.031̊
Consider a velocity V of 50 kmph

$$\text{At V} = 50 \text{ kmph}$$
$$R = 19.416\text{m}$$
$$R = \frac{b}{\sin\theta}$$
$$\theta i = 7.99°$$
$$\text{Tan } \theta_0 = \frac{R}{[\tan\theta]}$$

$\text{Tan } \theta i R + a$
$\theta_0 = 7.38$
Similarly at 100 kmph
$73.7\text{m} = \frac{2.7}{\text{Sin } \theta_i}$

$\theta_i = 2.0992°$
$$\frac{\text{Tan } \theta_0}{\text{Tan } \theta_i} = \frac{R}{R + a}$$

$\theta_0 = 2.05408°$
Consider V as 150 kmph
At V = 150 kmph
R = 166.64m
$166.64 = \frac{2.7}{\text{Sin } \theta_i}$

$\text{Sin } \theta_i = 0.016202592°$
$\theta_i = 0.92830785°$

$$\frac{\text{Tan } \theta_0}{\text{Tan } \theta_i} = \frac{R}{R + a}$$

$\text{Tan } \theta_0 = 0.0164744°$
$\theta_0 = 0.9438282°$

VI. TILTING MECHANISM I SECTION DESIGN AND CALCULATIONS

Construction of the I-section Tilting column:
An I section is considered 7.6 times stronger than a circular section of the same dimension
Reaction forces:

$$R_L = -R_R = \frac{M}{L} = \frac{\text{Diameter}}{\text{Length}}$$

Shear Forces:
$F_{LC} = F_{RC} = M$ (Shear forces aren't included because it is a couple)
Bending moment at the center is $M/2$
Bending formula is given by:

Model no	No. of phase	Step Angle	Rated Voltage	Current / Phase	No Load Run Frequency	Holding Torque	Rotor Inertia	Length	Shaft Dimension Φ D		Flange Dimension Φ D		KEY A	Wiring Diagram
Single shaft		degree	V	A	KHz	N.m	Kg-cm²	mm	mm		mm			
MST511C213 -X2AA6.0	2	1.8	120-310	6.0	≥20	27	33	165	19	-0.013 / -0.028	100	0 / -0.023	5x25	1
MST512C213 -X2AA7.0	2	1.8	120-310	7.0	≥15	40	48	230	19	-0.013 / -0.028	100	0 / -0.023	5x25	1
MST513C213 -X2AA7.0	2	1.8	120-310	7.0	≥12	50	60	270			100	0 / -0.023		1
On request	3	1.2	80-325	6.0	≥15	25	33	165	19	-0.013 / -0.028	100	0 / -0.023	5x25	
On request	3	1.2	80-325	6.0	≥15	37	48	230	19	-0.013 / -0.028	100	0 / -0.023	5x25	
On request	3	1.2	80-325	6.0	≥15	50	60	270			100	0 / -0.023		
On request	5	0.72	120-310	5.0	≥20	20	33	165	19	-0.013 / -0.028	100	0 / -0.023	5x25	2
On request	5	0.72	120-310	5.0	≥20	30	48	230	19	-0.013 / -0.028	100	0 / -0.023	5x25	2
On request	5	0.72	120-310	5.0	≥15	40	60	270			100	0 / -0.023		2

Fig V.III: Orthographic projections of tilter column mechanism

$$\frac{M}{I} = \frac{f}{Y} = \frac{E}{R}$$

Where M= moment
I= moment of inertia
f=stress
Y= distance of extreme fiber from neutral axis
E= Young's modulus
R= Radius of curvature

$$\frac{bd^3 - h^3(b-t)}{12}$$

Where $h = (D - 2t)^3$

$$Z = \frac{I}{Y} = \frac{I \times 2}{D}$$

Moment of resistance $= M_R = \frac{f \times I}{Y}$

$f = 435$ Mpa $B = 50$ $D = 80$

$$16050.345 = \frac{50 \times 803 - (50-t)(80-2t)^3}{80}$$

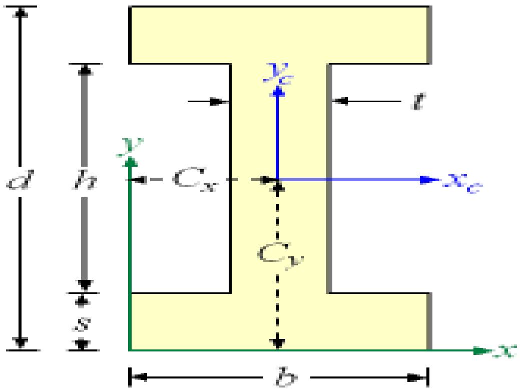


Fig V.II: I section

$1.284 \times 106 = 25.6 \times 106 - (50 - t)\,(80 - 2t)^{3}$

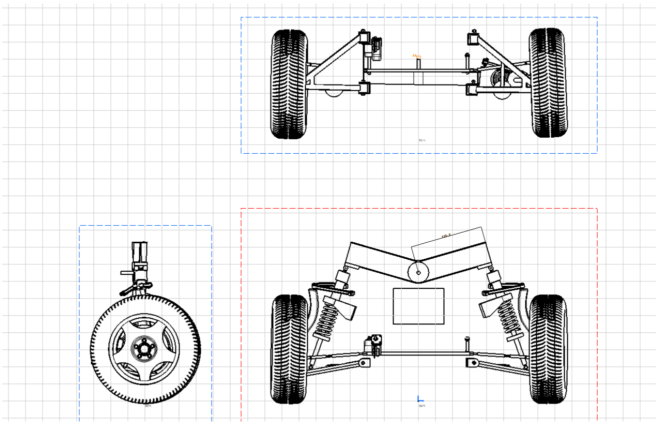
$24.316 \times 106 = (50 - t)\,(80 - 2t)^{3}$

$24.316 \times 106 = (25.6 \times 106 + 8t^4 - 1360t^{3} + 86400t^2 - 243200t)$

$8t^4 - 1360t^3 + 86400t^2 - 243200t + 1284000 = 0$

t = 3.056 mm

Closest commercially available I section beam is B=76mm, D=127mm, t= 4mm



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V.II TOGGLE MECHANISM DESIGN AND CALCULATIONS

Calculation for motor

For 20° angle between spring and toggle arm (the optimum angle obtained for force multiplication in toggle mechanism)

θ = 19.494°

α = 20°

a = 206.412 mm

h = 70.597 mm

consider a right angled triangle

$$\cos\alpha = \frac{\text{adjacent}}{\text{hypotenuse}}$$

$$\cos 20° = \frac{193.964}{\text{hypotenuse}}$$

Therefore, Hypotenuse = 206.412 mm

TABLE V.I: STEPPER MOTOR SPECIFICATIONS

$\tan \alpha = \frac{\text{opposite}}{\text{hypotenuse}}$
Therefore, opposite = 70.597 mm
Using a 50 Nm torque motor,
T = F × r
50 = F × .028 m
Therefore, F = 1762 N
Using the formula,
$P = \frac{F \times a}{2 \times h}$
For different values of h ranging from 0mm to 70mm we get a graph for values of which is shown in the analysis results of this mechanism.
The rack and pinion is placed at an angle of 12.5 degrees to the direction of the movement of the toggle mechanism.
Design for Pinion:
Diameter of pinion d_1 = 56mm
Teeth of pinion Z = 14
Module m = d/z = 4mm
Face width = 40mm for module (m) 1-20
Mass of pinion: 0.78 – 0.86kg
For 20° involute teeth:

i) Addendum ha $= 1 \times \mathrm{m} = 1 \times 4 = 4\mathrm{mm}$
ii) Dedendum hf $= 1.25\mathrm{m} = 5\mathrm{mm}$
iii) Whole depth h $= 2.25\mathrm{m} = 9\mathrm{mm}$
iv) Clearance $= 0.25\mathrm{m} = 1\mathrm{mm}$
v) Outside diameter of pinion $d_{a1} = (Z1 + 2) = 16 \times 4 = 64$ mm
i) Tooth thickness $s = \frac{\pi \times m}{2} = \frac{\pi \times 4}{2}$
ii) Working depth $h_1 = 2\mathrm{m} = 2 \times 4 = 8\mathrm{mm}$
iii) Circular pitch $p = \pi\mathrm{m} = 12.567\mathrm{mm}$
iv) Root diameter of pinion $d_{f1} = d_1 - 2hf = 56 - (2 \times 5) = 46\mathrm{mm}$

$P = \pi \times 56$
$= 175.93$mm

One rotation of pinion gives 175.93mm displacement of rack
0.4 rotation of pinion gives 70.597mm

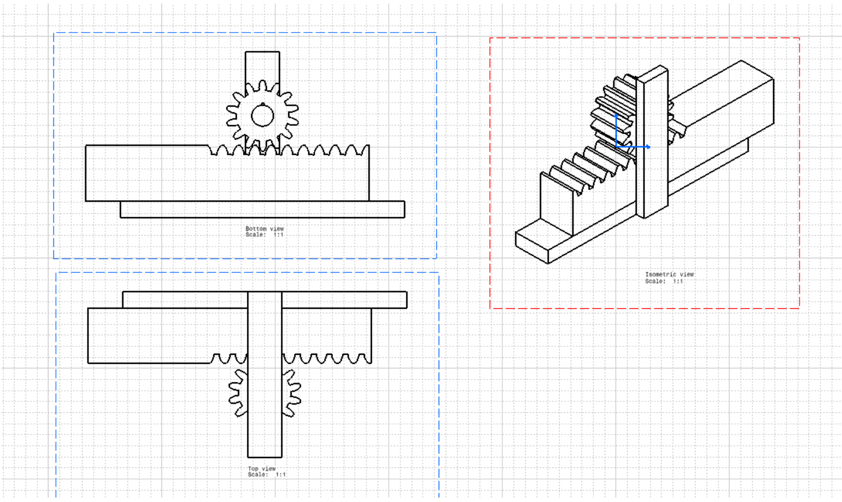

Fig V.IV: Orthographic projections of Rack and Pinion

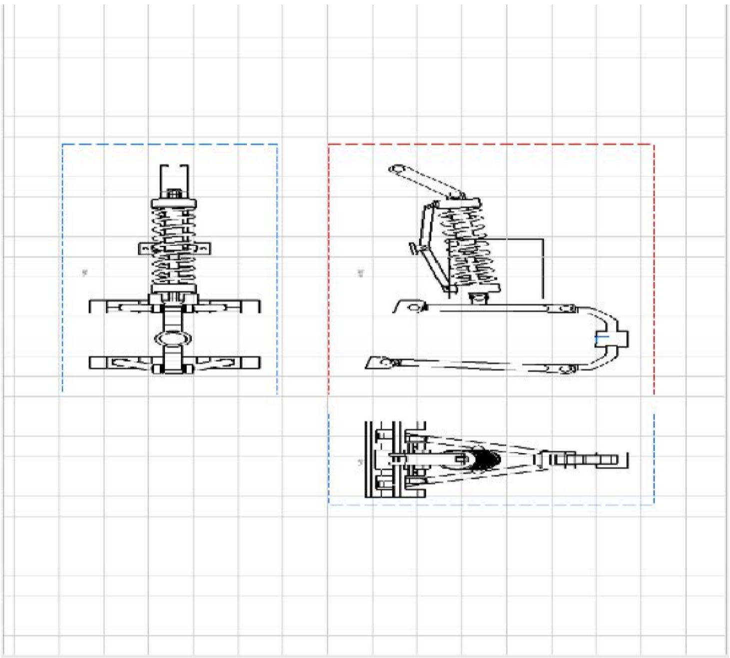

Fig V.V: Orthographic Projections of Toggle mechanism

V.III ANTI-ROLL HYDRO PNEUMATIC SUSPENSION DESIGN AND CALCULATION

Density = 912 kg/m^3
Acceleration due to gravity = 9.81 m/s^{-2}
At static position:
P_s = pressure at static condition
P_i = pressure at inner wheel
P_o = pressure at outer wheel
d = density of fluid

$$P_s = \frac{\text{force }(m*g)}{\text{Area of damper}} + d*g*h$$

At turning:

$$P_i = \frac{\text{inner wheel resultant force}}{\text{Area}} + d*g*h_i = 7.277 \text{ Mpa}$$

$$P_o = \frac{\text{outer wheel resultant force}}{\text{Area}} + d*g*h_o = 17.46 \text{ Mpa}$$

$$\frac{9000}{5.3*10^{-5}} + \frac{(912*9.81*220)}{10^{6}} = \frac{2800}{5.3*10^{-4}} + d*g*(224) + P_{on\ piston}$$

Pressure on the piston is 10.183Mpa.
Pushing the piston down on the inner wheel and up on the outer wheel.
Give pressure of 10 Mpa on accumulator to get the car to an original position.

$$\frac{(9000 + 5300)}{5.3*10^{-5}} + d*g*(249) = \frac{2800}{5.3*10^{-4}} + d*g*(194) + P_{on\ piston}$$

New pressure= 20.24
Difference in pressure = 10.06 Mpa
Considering the sprung weight the roll angle can be given as:

$$\Phi = \underline{4 * W_{sp} * f * h}$$

$a^2 * s_t$

Where Φ is roll angle

W_{sp} is sprung weight

f is centripetal acceleration

a is wheel track

s_t is wheel stiffness

During turning:

Density = 912 kg/m^3

Acceleration due to gravity = 9.81 m/s^{-2}

At static position:

P_s = pressure at static condition

P_i = pressure at inner wheel

P_o = pressure at outer wheel

d = density of fluid

the total pressure is the sum of pressure by damper and also the pressure by the liquid:

$$P_s = \frac{\text{force } (m*g)}{\text{Area of damper}} + d*g*h$$

At turning:

$$P_i = \frac{\text{inner wheel resultant force}}{\text{Area}} + d*g*h_i = 7.277 \text{ Mpa}$$

$$P_o = \frac{\text{outer wheel resultant force}}{\text{Area}} + d*g*h_o = 17.46 \text{ Mpa}$$

$$\frac{9000}{5.3*10^{-5}} + \frac{(912*9.81*220)}{10^6} = \frac{2800}{5.3*10^{-4}} + d * g * (224) + P_{\text{on piston}}$$

Pressure on the piston is 10.183Mpa.

Pushing the piston down on the inner wheel and up on the outer wheel.

Give pressure of 10 Mpa on accumulator to get the car to an original position.

$$\frac{(9000 + 5300)}{5.3*10^{-5}} + d*g*(249) = \frac{2800}{5.3*10^{-4}} + d*g*(194) + P_{\text{on piston}}$$

New pressure= 20.24

Difference in pressure = 10.06 Mpa

Considering the sprung weight the roll angle can be given as:

$$\Phi = \frac{4 * W_{sp} * f * h}{a^2 * s_t}$$

Where Φ is roll angle

W_{sp} is sprung weight

f is centripetal acceleration

a is wheel track

s_t is wheel stiffness

During turning:

The roll angle of the vehicle is formulated for various pressures

Inner wheel

$$\Phi = \frac{4 * 7.277*10^6*5.4*10^{-4}* 10.24 *0.8}{1.62^2 * 331}$$

$= 0.145^\circ$

Outer wheel

$$\Phi = \frac{4 * 7.277*10^6*5.4*10^{-4}* 10.24 *0.8}{1.62^2 * 331}$$

$= 0.349^\circ$

On a straight road roll angle is zero

During actuation:

Inner wheel

$$\Phi = \frac{4*(7.277 - 10 + (912*9.8*224*10^{-6}))* 10.24 *0.8* 5.3 * 10^{-4})}{331 * 1.62^2 *10^3}$$

$= 0.185* 10^{-6\circ}$

Outer wheel

$$\Phi = \frac{4*((-17.46+10)+(912*9.81*220*10^{-6}))*10.24*0.8*5.3*10^{-4}}{331*1.62^{2}*10^{3}}$$

$= -0.1099 * 10^{-4}$

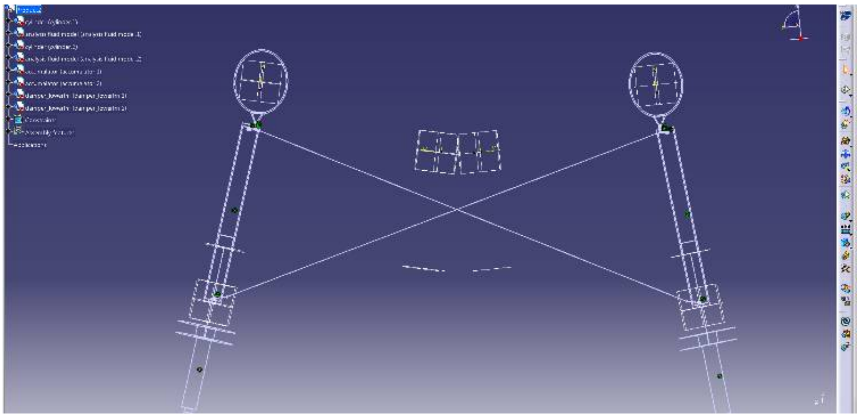

Fig V.VI: Wireframe view of Hydro-pneumatic anti-roll mechanism

VI. ANALYSIS RESULTS

VI.I MR Suspension with tilting column

Based on the calculation, design, analysis and simulation of the system we found a suitable solution to the problem. For the system to work efficiently a very powerful motor with a torque of 750 Nm with minimal weight is to be used. Our mechanism did not work based on the calculations made prior to the design. In spite of this we designed the system in order to examine the points of failure as a part of research study and future references. With the help of this study and research a further optimal and efficient design can be developed.

VI.II Toggle mechanism controlled suspension with MR assistance system

The below graph shows us the main advantage of this mechanism, i.e., the force produced by this mechanism exponentially increases with the decrease in the angle of the link with respect to the suspension damper axis.Hence this system can be used in high load conditions, such as in commercial vehicles like trucks, buses and trailers. The analysis of this mechanism is done in FUSION 360, to ensure that there is no mechanical failure occurring, when actuated. The material used for this analysis is AHSS Steel.

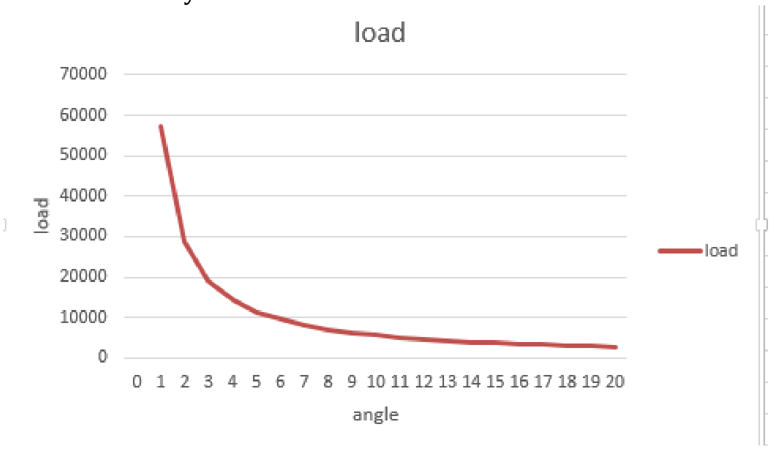

Fig VI.I: Load vs Angle of the link

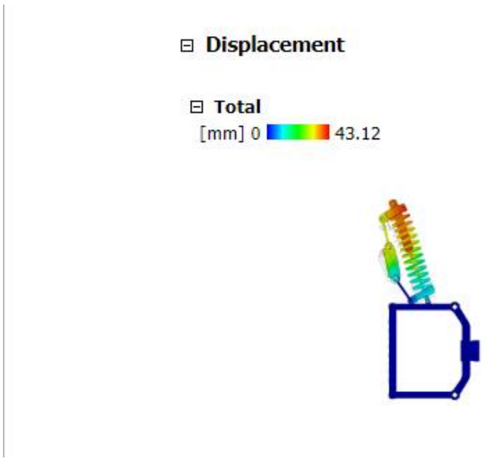

Fig VI.II: Displacement

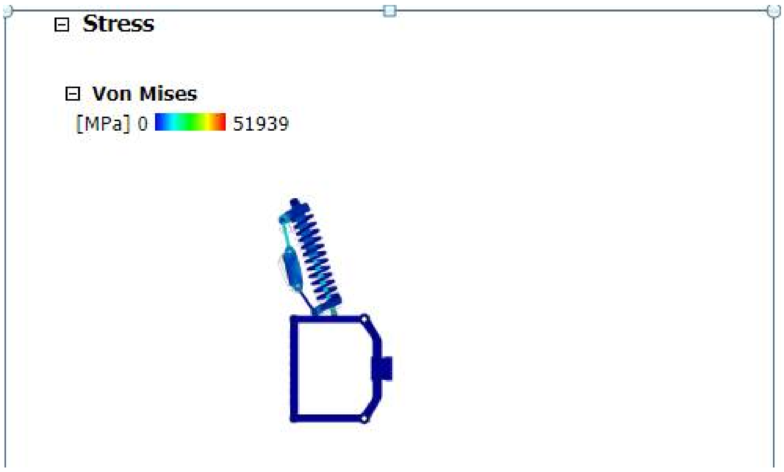

Fig VI.III: Von Misses Stress

As we can see from the above analysis that the maximum displacement occurs in the piston of the damper. When reaction force of 9000N from road is transferred to the damper, a force of 1700N is given to the toggle mechanism. This relatively small force is multiplied, as we saw in the previous graph, to oppose the 9000N force and hence body roll.

VI.III Anti-roll Hydro-pneumatic suspension system

The analysis of this mechanism was done on FUSION 360. With the help of Event Simulation Analysis, we found that the mechanism was subjected to mostly compressive stresses.

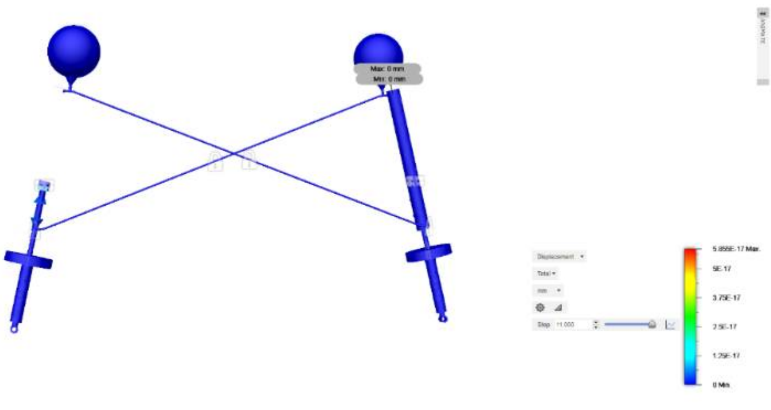

Fig VI.IV: Displacement Analysis

The displacement analysis shows us that the piston in the inner wheel damper displaces by 27mm for the given pressure of 10MPa. The stress analysis results show the mechanism to be safe as the safety factor did not go less than 4. The forces used in this analysis were considerably less as this mechanism is tested for passenger vehicles. Hence this mechanism cannot be used for heavy duty purposes, as pressure values at those conditions would be very for this mechanism to work effectively. The pressure and roll angle relation was also plotted to see to that the pressure changes in a linear manner with respect to the roll angle of the inner and outer wheel.

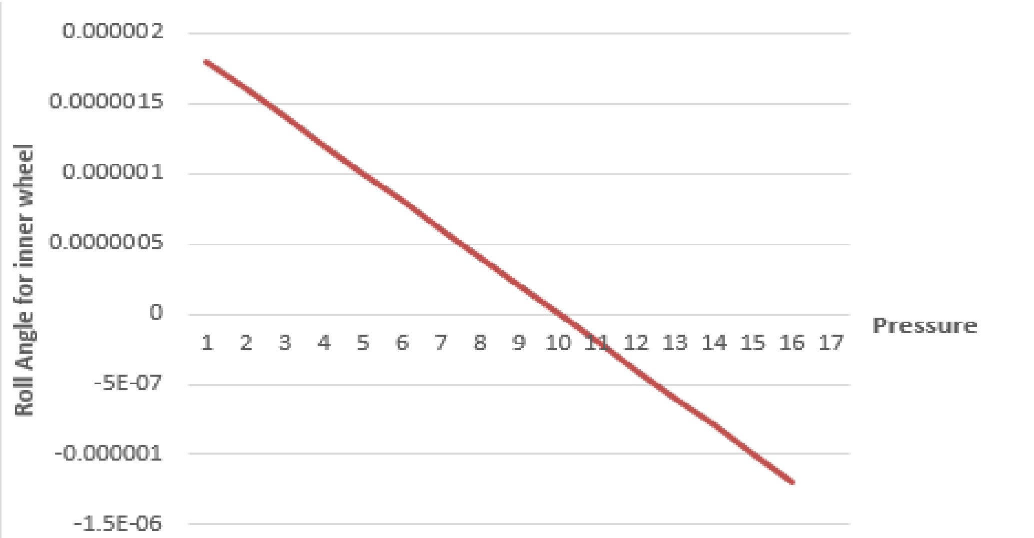

Fig VI.V: Roll angle vs Pressure (inner wheel)

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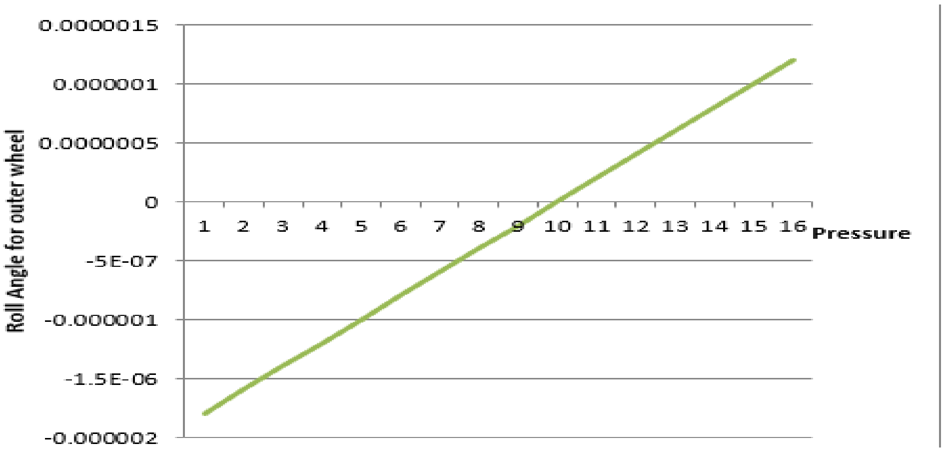

Fig VI.VI: Roll angle vs Pressure (outer wheel)

As we can see from the two graphs that, as the roll angle is changing, simultaneously pressure is given to the system to keep the vehicle flat and steady. Also since any of the wheel, the right or the left can be the inner or outer wheel, the two graphs are complimenting each other.

VII.CONCLUSION

The main objective of this project is to design and optimize the suspension design in a four wheeler to help the vehicle avoid body roll during high speed cornering. The three mechanisms presented in this project, two of the designs are successful and one mechanism is partially successful. The Tilter mechanism although the simple in design, requires a motor of very high torque to produce the output, that will resist the body roll. The Toggle mechanism and the Hydro pneumatic mechanism are mechanisms that can be implemented to achieve the objective.

Mechanism	Tilting column	Toggle mechanism	Hydro-pneumatic mechanism
Working	An I-section tilting column is placed on the suspension system, which is actuated by a high torque motor.	A toggle link mechanism is used to tilt the body of the vehicle, by using a motor and rack and pinion arrangement.	This system uses hydro-pneumatic systems to vary the pressure in the damper, hence controlling its stiffness.
Pros	Can produce large amounts of force. Suitable for Electrical vehicles.	Generates large output force for low input. Mechanical advantage.	Simple mechanism Takes less space
Cons	Requires motor that generates large amounts of torque	Takes lots of space.	Generates relatively less force
Applications	Electrical vehicles and SUV's.	Heavy commercial vehicles.	Light passenger vehicles.

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