Design and Development of Three Link Suspension System for a Quad Bike

Gaurav kadam[1], Mohit Kamble[2], Prajval Vaskar[3], Prathamesh Sangelkar[4]

[1,2,3,4]*Department of Mechanical Engineering, SAOE, Pune.*

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Abstract - *In Quad bike, particularly in sports type, Dependent suspension system is used as Rear suspension but there are some drawbacks of the dependent system such as lack of stability, high un-sprung weight, etc. To overcome these drawbacks independent suspension is used (Double wishbone, Three link, Multi-link). Now a-days Double wishbone independent suspension system is widely used in Quad bike because of its ease for manufacturing and cost. But it has some limitations too like it must be incorporated with anti-roll bar to resist rolling which adds the weight to system. So to keep the weight low with high performance suspension and more rolling stiffness three link suspension was designed. The paper describes the design and simulation of this type of suspension in quad bike.*

Key Words: **Independent Suspension, Three Link, CREO 3.0, Lotus Shark, ANSYS 16, Roll gradient.**

1. INTRODUCTION

Suspension system is a system which isolates the body and occupants from the irregularities of the road surface. It consists of tire, springs, shock absorbers, linkages. It is important for the suspension to keep the tire in contact with the road surface as much as possible, because all the road or ground forces act on the vehicle through the contact patches of the tires.

Functions of suspension system:

-Provide vertical compliance so the wheel can follow the uneven road, isolating the chassis from irregularities on the road.

-Maintain the wheel in the proper steer and camber attitudes to the road surface.

-React to the control forces produced by the tires- longitudinal (acceleration and braking) forces, lateral (cornering forces), and braking and driving torques.

-Keep the tire in contact with road with, minimal load variations.

- It should transfer minimum forces to the chassis.

-The properties of a suspension system are important to the dynamics of the vehicle primarily seen in the kinematic (motion) behaviour and its response to the forces and moments that it must from tires to the chassis. In addition, the other characteristics considered in the design process are cost, weight, package space, manufacturability, ease of assembly etc.

Dependent suspension system:

This type of suspension system acts as a rigid beam such that any movement of one wheel is transmitted to the other wheel. Also, the force is transmitted from one wheel to the other.

Independent suspension system:

Independent suspension allows each wheel to move vertically without affecting the opposite wheel. Independent suspension provides higher roll stiffness relative to vertical spring stiffness. In this roll center is easy to control by choice of geometry of control arms, the ability to control tread change with jounce and rebound, larger suspension deflections. It also has better resistance to steering vibrations.

1.1 Objective

The objective of this research is design and analysis of independent rear suspension for an all-terrain vehicle (quad bike). In rough terrains suspension plays vital role in ride comfort, load transfers and to some extent in safety too. The suspension was designed keeping all the vehicle performance requirements in these kind of terrains. Three link type of suspension was selected because of its added advantage to rolling stiffness and comparatively low weight.

1.2 Terminology

- **Degree of freedom:**

The assemblage of control arms in suspension system is to control the motion of the wheel in only one path i.e. bumps and droop. That path may also have camber change, castor change and toe change as the wheel moves. So the design of suspension system includes restraining the knuckle to limited motion in five directions.

- **Instant center of rotation:**

It is the point about which the all the links rotates. As in suspension the link moves up and down relative to the road conditions, instant center of rotation plays a vital role in limiting the camber change, castor change etc.

- **Instant axis:**

In three dimensional space instant centers are replaced by instant axis. It is obtained by connecting instant centers in front view and side view of the geometry. It is the axis about which the wheel moves. It doesn't change with the suspension travel.

- **Camber gain:**

It happens with change in the motion of wheel corresponding to bumps or droops. Length of the links, ICR affects the amount by which camber changes. The more the length of the suspension links less will be the camber change and vice versa. Similarly, the geometry having ICR at long distance have shorter camber gain as the geometry moves in arc having larger radius. Length of the arm also affects scrub of the tire i.e. lateral motion of the tire.

- **Roll gradient:**

The stiffness in roll is conveniently expressed in normalized form as degrees of roll per unit lateral acceleration (deg/g). This is called roll gradient.

2. DESIGN OF THREE LINK SUSPENSION

2.1 Iteration

The design is based on Instant axis concept. Instant axis was obtained by connecting ICRs in both side view and front view. Various constraints such as chassis, differential, mounting tabs etc. were considered. Iterations were done by changing the position of ICR and the link angles. One with minimum camber change, wheelbase change and track width change was finalized. Knuckle pivot points were assumed considering the packaging of the wheel assembly. Iterations were done using CREO 3.0.
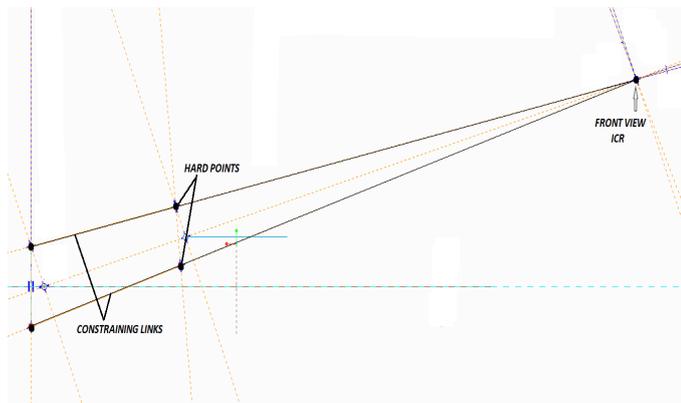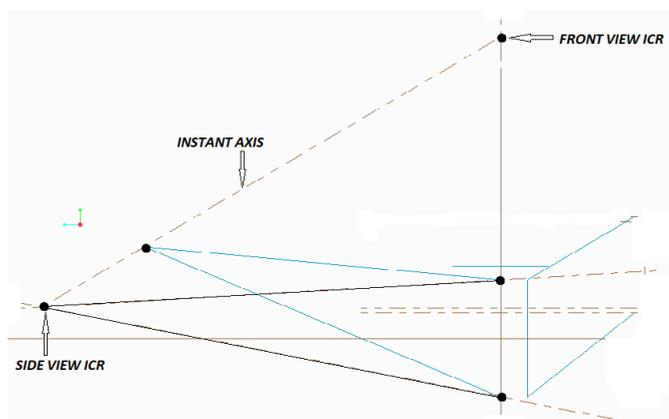

Fig 1- Front view ICR
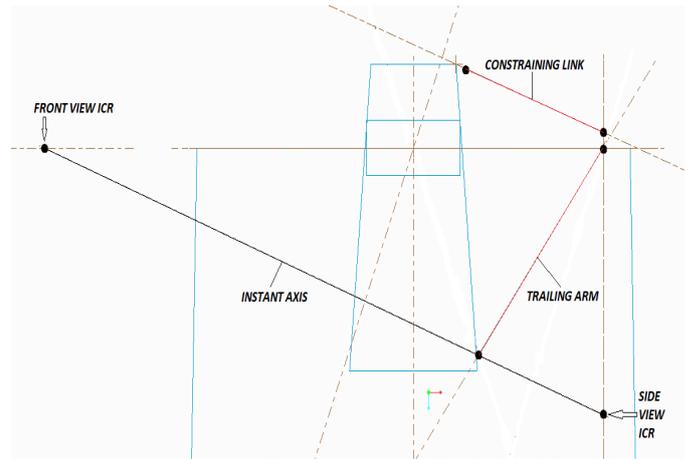

Fig 2- Side view ICR

Fig 3- Top view ICR

2.2 Simulation

The suspension simulation was done in LOTUS SHARK. The position of the hard points was imported from the link design in CREO. It gave various dynamic parameters of the suspension system, summary of which is illustrated via images below.
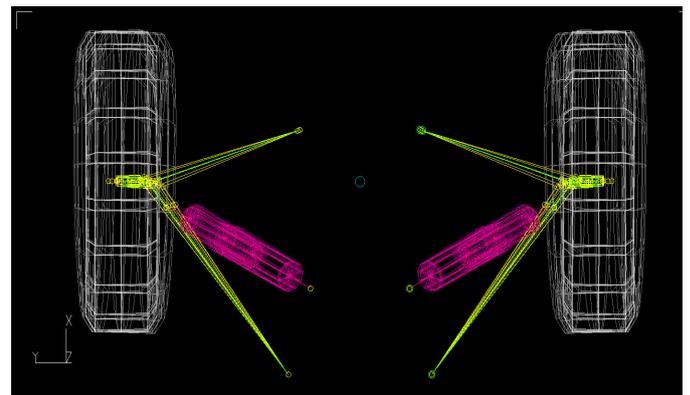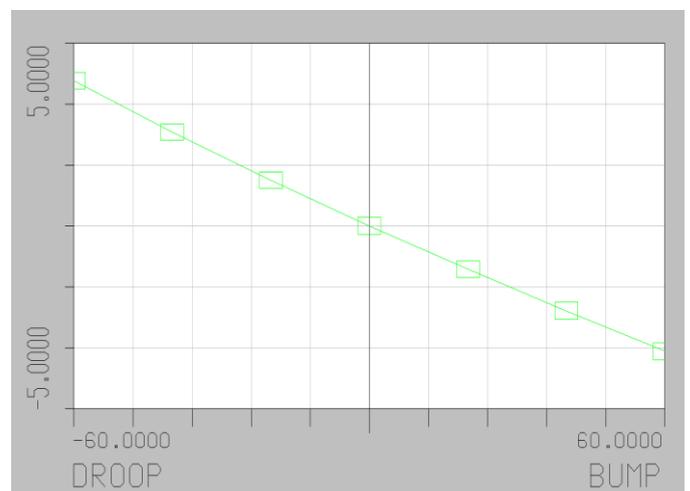

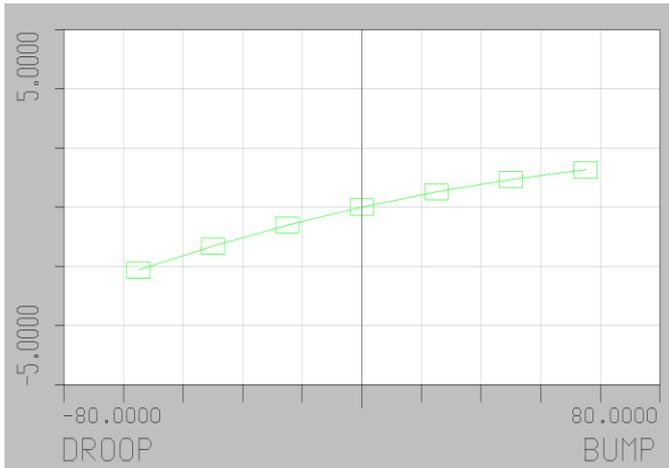**Fig 4-** Line diagram in LOTUS

Chart -1: Camber change with travel

Chart -2: Toe change with camber
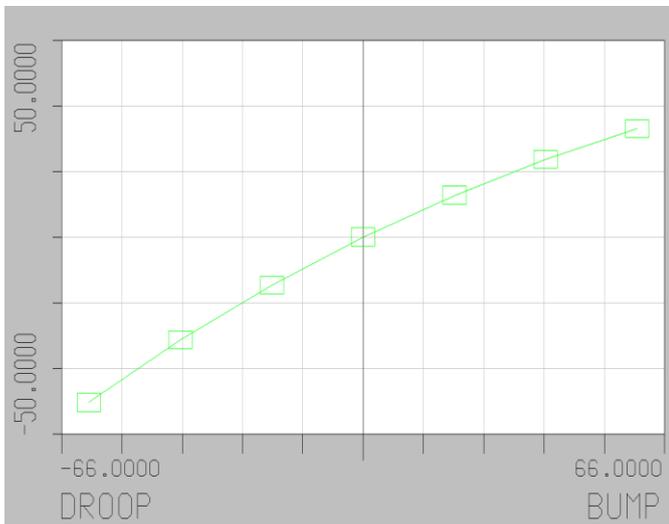
Chart -3: track change with travel

BUMP TRAVEL (mm)	CAMBER ANGLE (deg)	TOE ANGLE (deg)
-60.00	3.9650	-1.7833
-50.00	3.2497	-1.4291
-40.00	2.5594	-1.0990
-30.00	1.8914	-0.7919
-20.00	1.2436	-0.5068
-10.00	0.6137	-0.2431
0.00	0.0000	0.0000
10.00	-0.5993	0.2229
20.00	-1.1857	0.4261
30.00	-1.7606	0.6100
40.00	-2.3254	0.7749
50.00	-2.8813	0.9210
60.00	-3.4295	1.0487
70.00	-3.9710	1.1579
80.00	-4.5070	1.2489
90.00	-5.0385	1.3218
100.00	-5.5665	1.3766

Table -1: Incremental suspension values

BUMP TRAVEL (mm)	HALF TRACK CHANGE (mm)	WHEELBASE CHANGE (mm)
-60.00	-41.93	-2.04
-50.00	-33.74	-1.61
-40.00	-26.07	-1.23
-30.00	-18.89	-0.88
-20.00	-12.17	-0.57
-10.00	-5.88	-0.27
0.00	0.00	0.00
10.00	5.48	0.26
20.00	10.59	0.51
30.00	15.33	0.74
40.00	19.73	0.98
50.00	23.79	1.21
60.00	27.53	1.45
70.00	30.95	1.69
80.00	34.06	1.94
90.00	36.88	2.19
100.00	39.41	2.46

Table -2: Incremental suspension values

BUMP TRAVEL (mm)	ANTI DIVE (%)	ANTI SQUAT (%)
-60.00	57.43	8.81
-50.00	58.37	9.46
-40.00	59.40	10.15
-30.00	60.51	10.86
-20.00	61.70	11.61
-10.00	62.97	12.39
0.00	64.33	13.20
10.00	65.78	14.05
20.00	67.32	14.94
30.00	68.95	15.86
40.00	70.68	16.84
50.00	72.52	17.86
60.00	74.46	18.93
70.00	76.52	20.06
80.00	78.70	21.25
90.00	81.01	22.50
100.00	83.46	23.82

Table -3: Incremental suspension values

2.3 Link analysis on ANSYS 16.0

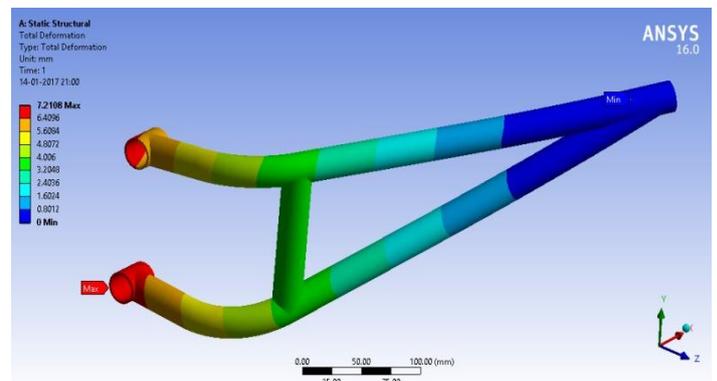Pipe Material - AISI 4130, Dimensions – OD=25.4, Thickness=1.6mm
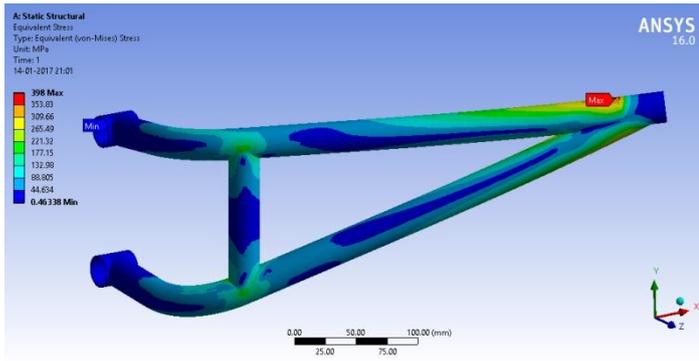

Fig 5- Total deformation

Fig 6- Von misses stress

Table -4: Result

Sr. No	Parameters	Values Obtained
1	Maximum deformation	7.21mm
2	Equivalent stresses	398Mpa

2.4 Calculations for Ride rate and Roll rate

Front Track width (t_f) = 3.5 feet

Rear Track width (t_r) = 3.33 feet

Wheelbase (l) = 3.83 feet

Height of C.G from the ground (h) = 1.44 feet

Distance between C.G to Roll axis (H) = 0.8 feet

Cornering radius (R) = 10 feet

Speed of vehicle (V) = 15 Km/h

Angle of banking (α) = 0°

Weight on front left tire $(W1)$ = 100 lb

Weight on front right tire $(W2)$ = 100 lb

Weight on rear left tire $(W3)$ = 122.22 lb

Weight on rear right tire $(W4)$ = 122.22 lb

Total weight on Front tires (W_f) = 200 lb

Total weight on rear tires (W_r) = 244.44 lb

Total Weight of vehicle (W_t) = 444.44 lb

Calculation for location of C.G

Location of C.G from rear axle (b) = $(W_f * l)/W_t$

$$= (200*3.83)/444.44$$

$$= 1.723 \text{ feet}$$

Location of C.G from front axle (a) = l-b

$$= 3.83-1.723$$

$$= 2.107 \text{ feet}$$

Lateral acceleration = V^2/Rg

$$=192.51/ (10*32.2)$$

$$=0.59 \text{ g's}$$

As there is no banking of road

Effective weight (W') = Total weight (W_t)

Therefore, Effective weight on front side= Total weight on front side

$$(W_f') = (W_f)$$

$$= 200 \text{ lb}$$

Effective weight on rear side= Total weight on rear side

$$(W_r') = (W_r)$$

$$= 244.44 \text{ lb}$$

Weight transfer due to lateral acceleration

Assume the vehicle is taking right turn

Weight transfer on front axle = (lateral acceleration *height of C.G.* front axle load)/front track width

$$= 0.59*1.44*200/3.5$$

$$= 49.19 \text{ lb}$$

W_{FO}= 100+49.49

$$= 149.49 \text{ lb}$$

W_{FI}= 100-49.49

$$= 50.51 \text{ lb}$$

Weight transfer on rear axle = (lateral acceleration *height of C.G.* front axle load)/ rear track width

$$= 0.59*1.44*244.44/3.33$$

$$= 62.36 \text{ lb}$$

W_{RO} = 122.22+62.36

$$= 184.58 \text{ lb}$$

W_{RI}= 122.22-62.36

$$= 59.86 \text{ lb}$$

Change from the static level measured on the ground

W_{FO} = 149.49-100=49.49 lb

W_{FI} = 50.51-100 = -49.49 lb

W_{RO} = 184.59-122.22= 62.36 lb

W_{RI} = 59.86-122.22= -62.36 lb

Assume 3.5 inches of jounce travel and use 2.5 inches to accommodate weight increase allowing 1 inch for additional travel due to braking, acceleration and localized bump on the track.

Ride rates

Ride rate front (K_{RF}) = 49.49/2.5

$\qquad$ = 19.79 lb/inch

Ride rate rear (K_{RR}) = 62.36/2.5

$\qquad$ = 24.94 lb/inch

Ride frequencies assuming side to side geometry

ω_F = 1/2π $\sqrt{}$ (K_{RF} * 12 *32.2/W1)

$\quad$ =1/2π $\sqrt{}$ (19.79 * 12 *32.2/100)

$\quad$ = 1.39 Hz or 83.88 cpm

ω_R = 1/2π $\sqrt{}$ (K_{RR} * 12 *32.2/W3)

$\quad$ = 1/2π $\sqrt{}$ (24.94 * 12 *32.2/122.22)

$\quad$ = 1.41 Hz or 84.79 cpm

Most rear drive vehicle, the front natural frequency is higher than rear, based on front roll stiffness requirements. At this point, ω_R cannot be less than 84.79 cpm without using up the 2.5 inch of bump travel. This cannot be decreased, thus ω_F will be increased to 115 cpm.

To increase ω_F square the ratio of natural frequencies.

ω_{Fnew} / ω_{Fold} = (115/88.88)^2

$\qquad$ = 1.87

Therefore new front ride rate is = 19.79*1.87

$\qquad$ = 37.19 lb/inch

Assuming left and right ride rates are same

Therefore roll rates,

$K_{\Phi F}$ =12*K_{RF}* t_f^2 /2

$\quad$ = 2733.46 lb-ft/rad.

$K_{\Phi R}$ = 12* K_{RR}* t_R^2 /2

$\qquad$ = 1659.34 lb-ft/rad.

Roll gradient = Φ/lateral acceleration= -W*H/ ($K_{\Phi F}$ + $K_{\Phi R}$)

$\qquad$ = -444.44*0.8/(2733.46+1659.34)

$\quad$ = - 0.08 rad/g or -4.63 deg/g.

3. VEHICLE PHOTOS

Fig 7- Three link suspension on quad bike

Fig 8- Vehicle during cornering

4. CONCLUSION

After numerous iterations on link design, we were able to formulate and implement a three link suspension for a quad bike. Suspension had a little more camber gain compared to double wishbone but it successfully fulfilled the requirement of less vehicle roll. As an all-terrain vehicle it had sufficient wheel travel with more vehicle stability and durability.

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