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Design of Double Wishbone Suspension System of BAJA Vehicle

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Abstract — *Independent motion of wheels, capable of navigating on uneven and bumpy road is important characteristic of suspension system for BAJA vehicle. The system connects the wheels with chassis by means of an assembly, which provides stiffness necessary to withstand shocks from roads. Suspension system dictates resistance to roll moment, understeer or oversteer characteristics and many other factors. In this way, it is directly responsible for handling of vehicle and safety. The suspension design must be quite stiff in order to bear sudden shocks due to potholes, steep drops, etc. The suspension characteristics of vehicle aide in manueverablity and drive comfort. Therefore, correct design of system is not only a matter of driver safety, but also a key factor in result of racing. The paper discusses various types of suspension systems suitable for BAJA vehicle, the design methodology and results achieved by simulation. It is paramount to select proper combination of spring and linkages to achieve optimum performance, with less weight. The design of suspension system is iteration-based and requires correct compromise between comfort and performance, to provide driver fatigue-less drive.*

Keywords: *Independent Suspension, Double Wishbones, Spring and Damper System, Roll Moment, ICR Diagram, Simulation, Air Spring.*

I. INTRODUCTION

The suspension system of BAJA vehicle has to withstand major shocks over uneven terrain. This calls for high stiffness springing and rigid body. However, higher the stiffness, lesser will be the comfort. Hence, the driver will feel premature fatigue and thereby the performance will be affected, on way or the other, in extreme use of springing. This is prevented by finding optimum spot between comfort and performance. Spring stiffness is one of the major parameters in design procedure. The choice of correct type and its characteristics can be of vital part in winning strategy.[1] Therefore, it is important to specify the logic behind selecting such components. It is equally important to validate the selected data using simulation software. This will ensure optimum performance characteristics and thereby enable designer to select proper input variables.

The independent suspension is major part in modern vehicle design. Allowing wheels independent motion, unhampered by relative displacement of linkages is base of comfort riding condition. The steering and suspension systems are crucial for successful operation of any variety of cars. Due to the large responsibility that the these two major components share coupled with the fact that BAJA cars are required to be capable of taking over toughest of the terrains possible, it is obvious that consequences of failure or improper setup of the suspension and/or steering could be quite catastrophic.

The suspension system, which is widely used in off-road vehicles, like BAJA and Quad, and in many commercial vehicles is independent suspension.[2] This gives freedom to wheels to oscillate about certain point, without disturbing each other's motion. This gives advantage over conventional solid axle, where both side wheels are directly coupled on same member. The independent suspension is further segregated into various types, each with unique design and its capabilities, wiz. McPherson strut, double wishbone, multilink suspension, trailing and semi-trailing arm, etc. Out of these available design possibilities, double wishbone type suspension is selected which gives adequate space for wheel assembly, better control over design characteristic and flexibility to designer in terms of performance variables.

II. Basic Concepts Related to Suspension of Vehicle

Before diving into the design, one must be familiar few concepts that play major role in design procedure, both directly and indirectly. These terms are both controlling parameters as well as design variables in design.[2]

2.1 Caster angle:

Caster angle can be defined as the angle made by the kingpin in the side view with the vertical axis passing through the wheel centre.Castor angle is very influential parameter in the dynamic behaviour of the vehicle. It is a stability oriented property and is responsible for steering centric restoring force i.e. the amount of castor affects the feel of steering and the amount of efforts required to turn the wheel. Figure below shows positive castor angle.

2.2 Caster Trail:

When the kingpin axis in the side view is extended further, it intersects ground at a particular point, the distance between that point and centre of the contact patch is called castor offset or castor trail. Castor trail affects the resisting driver effort. Some amount of castor trail is required for reducing driver effort.

2.3 Kingpin Inclination:

Kingpin inclination is defined as the angle at which kingpin axis is inclined to the vertical axis passing through the wheel centre. Kingpin angle affects the performance of the car when the wheels are steered. More the kingpin angle more the car will lift when steered. When the kingpin axis is extendedup to the ground, it intersects the ground at a particular point. The distance of that point from the centre of wheel contact patch is called scrub radius. Scrub radius increases the wear of the tyre but some amount of negative scrub radius is required so that the wheel purely rolls when steered.

2.4 Camber:

Camber is another important parameter in dynamic behaviour of the vehicle. Camber is defined as the angle between a tilted wheel plane and the vertical. It is one of the important terms, which describes the suspension's alignment. Camber angle can have both negative as well as positive orientation. Camber is considered positive if the top of the wheel leans outwards and negative if the top of the wheel leans inwards. If a vehicle's wheels are properly cambered, a beneficial thrust force is produced. This thrust force, aptly named as camber thrust, contributes a lateral force in the direction of tyre's tilt. In other words, it ensures stability by pulling the bottom of the tyre in the same direction in which the top is leaning.

2.5 Toe:

Toe is final parameter used to describe a vehicle's alignment. Toe is the symmetric angle that each wheel makes with the longitudinal axis of the vehicle, as a function of static geometry, kinematic and compliant effects. Tyre wear is heavily dependent on the toe distances. Dynamic factors induce change in toe, which lead to two conditions- toe in and toe out. In rear wheel drive cars, increased front toe in provides greater straight-line stability at the cost of some sluggishness in steering response.

2.6 Roll Centre;

The Roll Centre of a suspension system is that point in the transverse plane of the axles, about which the sprung mass of that end of the vehicle will roll under the influence of that end of the vehicle will roll under the influence of centrifugal force.

III. Objectives of Suspension System

The suspension system is designed keeping track of driver input and desired behaviour response of vehicle. Following are the general objectives of double wishbone suspension system:

- To provide all wheel independence.
- Maintain ground and wheel contact for maximum amount of time throughout the ride.
- Avoid excess rolling of the chassis.
- To provide greater travel, this allows better absorption of the shocks during the changes in ground conditions.
- To reduce un-sprung mass so as to have lesser inertia loads, thus the response time of the suspension to changes in the track surface is minimized. This allows the tire to maintain constant contact with the surface as much as possible.
- To provide better handling while cornering by providing camber gain.

Roll axis inclination is directly related to the steering characteristics of the vehicle. Hence, rear roll centre has to be placed above the front roll centre to counter the effects of over-steering of the vehicle.

Validation of these objectives is done by simulation using MSC Adams car software package. This provides designer ability to modify and accommodate different dimensions of rigid links and analyse their effect on final outcome. Thus, on iteration basis, designer selects optimum values of design parameters, simulating the results in near-real-time.

IV. Components of Double Wishbone Suspension

Figure 1. Double Wishbone Suspension system

The suspension system is one constituent of a trio consisting steering system, wheel assembly and suspension system. Therefore, few of its parts are common in either of those assemblies. Followings are major components in a double wishbone suspension system:

4.1 Control Arms:

Control arms, in this case A-arms, are the rigid links that connect the wheel assembly with chassis, determine parameters like roll centre position, static camber and camber gain. These are A- shaped links made from steel. These has to withstand great bending force, partial impact loading from bumps and pots.[3] The dimensions of control arm are defied by keeping in mind the outcome required. Their design procedure include combination of loading criteria and suspension geometry as well. The material selection is done considering strength as the high preference factor; the lower control arms are designed using 2 mm thickness; 1.25-inch AISI 1018 tube, while the upper control arms are designed using 1.2 mm; thickness 1.25-inch AISI 1018 tube.

4.2 Shock Absorber:

The shocks and impact loads on wheel are prevented from being transferred to the chassis by a shock absorber. Shock Absorber is the most crucial component of the suspension system. The performance of a suspension system is directly dependent on the performance of the shock absorbers.[4] Hence, proper selection of shock absorber is very much an important criterion for better performance of the vehicle.

For selection of suitable shock absorbers, it is required to set desired natural frequency of the suspension system. According to Allan Staniforth's Competition Car Suspension,[5] the front frequency needs to be about 10 % lower than the rear. This is to avoid pitching movement caused by the front wheels rising over a bump first, followed shortly afterwards by the rears. Once started, it is at best unpleasant and at worst is capable of sending a jumping rally car into a complete somersault. However, practical experience on the track shows that on good surfaces with high frequency suspensions this "rule" is no longer necessarily valid and front frequencies may well be higher than the rear when a search for better balance is going on. In case of BAJA vehicle, with the help of stiff shock absorbers, the pitching movements are controlled greatly, but balance of the vehicle is critical. Hence, it is logical to go with more frequency at the front than the rear. Studying various books related to the topic, it was decided to keep the natural frequency in the range of 80-100 CPM (cycles per minute) for both front and rear.

Air spring is one of the most favoured type of shock absorber.Fox Float 3 shock absorbers were selected which provides easy adjustment of stiffness and progressive damping.[6] The stiffness of the shock absorber can be easily changed by changing the air pressure in the cylinder.

V. Design Considerations for Front Suspension Geometry

Following table shows values of general vehicle specifications required in design.

Table 1. General parameters required in design.

Parameters		**Values**
Wheelbase		56”
Track-width:	Front Rear	50” 50”
Ground Clearance		11.695”
King-pin Inclination		8^0
King-pin Length		150 mm
King-pin Offset		90 mm
Castor Angle		11^0
Scrub Radius		48.95 mm
Rim Dimensions:	Diameter Width Offset	10” 5” 3-2”
Tyre Dimensions:	Outer Diameter Inner Diameter Width	23” 10” 7”

The generation of geometry of font suspension includes specifying dimensions and relative positions of rigid linkages. This is done by defining the desired goal and simultaneous simulation of each iteration with the help of Instantaneous Centre of Rotation Method. Iterations for obtaining the suspension geometry are done using PTC Creo Parametric 2.0. To avoid bump steer and to have optimum distance between the front roll-centre and CG is the desirable. A geometry that yields the desired outcome was finalised. Length and angles of control arms and tie-rod was obtained from the geometry. The ICR diagram generated for suspension geometry is shown below. The dimensions of linkages, their relative angular positions and position of roll centre is also shown in the figure. This is the final iteration for specifying the dimensions.

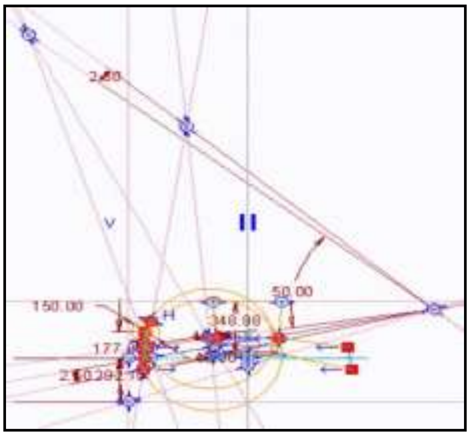

Figure 2. ICR diagram of suspension system

After performing iterations, following parameters are finalised:

Table 2. Final geometry parameters

Parameters		**Values**
Upper Control Arm	Length Angle	336.92 mm 9.45°
Lower Control Arm	Length Angle	374.8 mm 14.59°
Roll Centre Height		242.33 mm

VI. Simulation in MSC Adams Car Software Package

The geometry of the designed suspension system was imported into the simulation software. MSC Adams Car software was used for simulating the suspension system. The results of simulation gave us a clear prospect of what our suspension system was designed for. Figures showing simulation environment can be seen below.

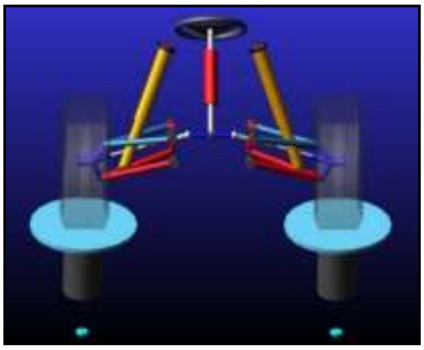

Figure 3. Simulation of front suspension on MSC Adams

Suspension System was simulated for camber change with wheel travel, castor change with wheel travel and body roll with respect to wheel travel. The graphs generated are attached below:

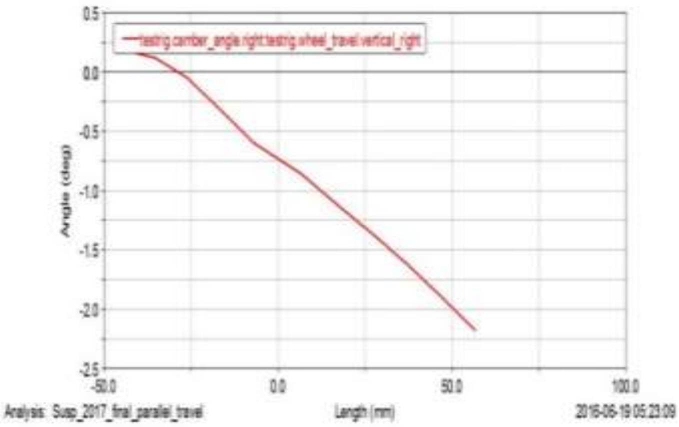

Figure 4. Camber Vs Wheel travel graph.

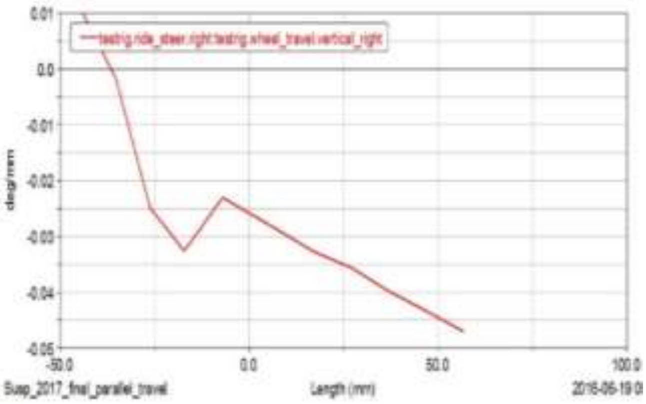

Figure 5. Graph of bump steer.

VII. FEA of Control Arms

FEA of A-arm was performed to get results on stresses and deformation of linkages. ANSYS software was used to analyse the linkages.

Boundary conditions considered:

1. The A-arm chassis mounting points were considered frictionless support.
2. The point where shock absorber is mounted is considered a fixed support.
3. The A-Arm knuckle point is subjected to the bump force.

Followings are the results of FEA performed on model of A-Arm:

Figure 6. Boundary condition for A-arm

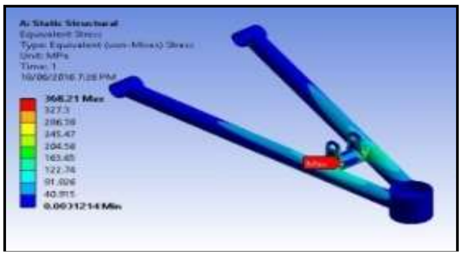

Figure 7. Von mises stress in A-arm

The maximum stress occurring in A-arm is 368.21N/mm^2

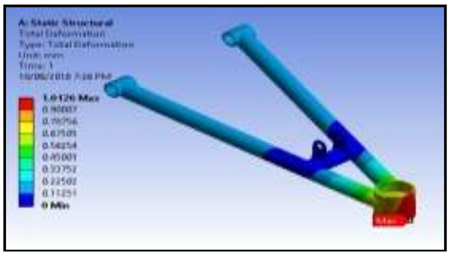

Figure 8. Deformation of A-arm

The maximum deformation in A-arm is 1.01 mm.

VIII. Calculations for Roll Rate

8.1 Abbreviations and input values:

G = 9.81m/s²

Mass on front wheels = 96 kg (mf)

Mass on rear wheels= 144 kg (mr)

C.G front, hf = 355.6 mm

C.G rear, hr = 405.6 mm

RC height, RC_F = 242.33mm

RC height, RC_R = 250mm

Front track, T_F=1270mm

Rear track, T_R=1270mm

Front spring roll rate = $K_{\varphi fr}$lbs.ft/°

Rear spring roll rate = $K_{\varphi sr}$lbs.ft/°

Front ride rate = K_{RF} lbs./in.

Rear ride rate = K_{RR} lbs./in

Front ride rate frequency = w_f = K_{RR} lbs./in

Roll moment front $=M_{1gf}$ N/m

Roll moment rear= $M_{1g}r$ N/m

8.2Front Ride Rate:

From Race Car Vehicle Dynamics written by William F. Milliken and Douglas L. Milliken,[1]
Front spring roll rate:

$$K_{\varphi SF} = \frac{K_{RF} * TF^2}{1375}$$

$$= \frac{29.4 \times (\frac{1270}{25.4})^2}{1375}$$

$K_{\varphi SF} = 53.45$ Nm/°

$$K_{\varphi SF} = \frac{53.45 \times 4.4482}{(\frac{1000}{12 \times 25.4})}$$

$K_{\varphi SF}$=72.5 Nm/°

8.3 Roll moment for the front at 1g acceleration:

$M_{1gf} = (h_f - RC_f)\ m_f g$

$M_{1gf} = (0.556-0.24233)\times 96\times 9.81$

$M_{1gf} = 106.67$ Nm

$$\text{Roll rate at front} = \frac{\textit{Roll moment}}{K_{\varphi}SF}$$

$$= \frac{106.67}{72.5}$$

Roll rate at front= 1.47°/g

8.4 Calculations for Coil Rate of shock absorber:

Desired Natural Frequency = 100 CPM or 1.667 Hz

Sprung Mass at the front = 80.75 kg

Wheel Rate (lbs./in.):

$$WR = \left(\frac{\text{Wheel Frequency (CPM)}}{187.8}\right)^2 * \text{Sprung Weight (lbs)}$$

$$= \left(\frac{100}{187.8}\right)^2 * \frac{80.75}{2} * 9.81 * 0.2248$$

= 25.24 lbs./in.

Assume,

Motion Ratio (Suspension leverage) = 1:1

Coil rate = Wheel rate * (Suspension leverage) ²

= 25.24 * (1) ²

Coil Rate = 25.24 lbs./in.

8.5Final Iteration:

For Motion Ratio = 0.7

Coil Rate = Wheel Rate * (Suspension leverage) ²

Coil Rate = $25.24 * \left(\frac{1}{0.7}\right)^2$

Coil Rate = 49.61 lbs./in.

IX. Results

Following table consists of values of final design parameters:

Table 3. Result of simulation and calculations

Parameter	Value
Front Spring roll rate	72.5 Nm/°
Roll rate at front	1.47°/g
Wheel Rate	25.24 lbs./in.
Motion ratio	0.7
Coil Rate	49.61 lbs./in.
Wheel Frequency	99 CPM or 1.65 Hz

X. Conclusion

After initial testing of the sub-systems designed, it was observed that the systems fulfilled their individual performance expectations. The suspension system has near about optimum steer characteristics. Thus, aiding the driver handle the vehicle better. The design goals are validated through simulation, as the system yields required output. The calculations describe values of roll rate at front, wheel rate and coil rate along with wheel frequency. In this way, the design of suspension system is successfully completed and serves as ground for further research on said topic

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