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Optimized Suspension Design of an Off-Road Vehicle

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ABSTRACT
Suspension system is the term that defines the transmissibility of an off-road vehicle. In order to resist the bumps and jerks that usually occur in an off-road track, an integrated approach of design is developed to obtain an optimized geometry which can give the drivers a 'fun-to-drive' experience. This paper describes the development of this suspension and steering geometry design that is fast enough to be used at off-road circuit giving us appropriate camber and caster variations , toe angles , Ackermann geometry , proper flow of forces from chassis to ground and shock absorber characteristics when running on the challenges posed by a rugged off-road track. The geometry design discussed here was achieved through the thorough study of its dimensions, position of installation and application. This vehicle was a Baja off-road prototype which is used in international competitions among universities with its top speed as 45- 65km/hour and its turning radius being 10.5 ft. The car is rear wheel driven.

INDEX TERMS— Ackermann geometry; instantaneous center; off-road vehicle; optimization design; over steer; un-sprung mass.

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I. INTRODUCTION

Suspension is the most important aspect while designing an off-road vehicle. In recent years, with the development of people's travel requirements, handing stability and ride comfort are very important features of automotive driving [1]. In off-road terrain the track consists of all kinds of obstacles that could easily bind up the suspension of any road vehicle. Thus, to make the vehicle compatible to off-road conditions it is necessary to design a suspension system that can handle the roughest of bumps without affecting the vehicle stability and at the same time also provide a smooth ride to the driver. The suspension geometry should be such that it doesn't undergo drastic changes during wheel travel or heave.

There are a few design aspects that were strictly adhered to while designing this suspension system. The primary objective was to keep the suspension parameters as far from changing with wheel travel as possible. The ground clearance was to be kept enough to clear large rocks, bumps and logs. The migration of the Instantaneous Centers was to be kept as low as possible. The Roll Centre heights in the front and the rear were to be kept such that the Roll Axis slope inclined forward. This helps quicker load transfers as well as provides over steer characteristics which is very useful in off road car[3]. The other aspect which was to be considered for deciding the Roll Centre height was the Centre of Gravity height, which make a rolling arm, coupling with the roll center. The Side View Swing Arm angle was to be considered as it provides the maximum load transfer to the shocks and reducing the load being sent to pivots, thus also increasing ride quality. One other objective was to prevent the rear drive-shafts from rocks and severe bumps which may exceed the angle of the tripod joint leading to failure of power transfer[3]. The turning radius was to kept low at the same time keeping the under steer and tyre slip angles to a minimum. This led to a true Ackerman geometry.

II. FRONT SUSPENSION DESIGN CONSIDERATION

Front Suspension is very important and is designed first, in general, while designing off-road suspension[4].The front of the vehicle faces the obstacles or the jerks first and makes the base of the motion and loading characteristics. Thus, in an off-road vehicle it should provide ample wheel travel and damping effects to absorb the bumps and jerks. Moreover it should provide large amount of traction, as bearing the steering system, to maintain directional stability and reduce slip angles to prevent losses. The front of the vehicle should also have low amounts of un-sprung mass to keep optimum ride characteristics. The front suspension should also handle the load transfers optimally so that the traction on the front wheels is ample at all times to lead the vehicle as the steering system dictates, which may otherwise result in

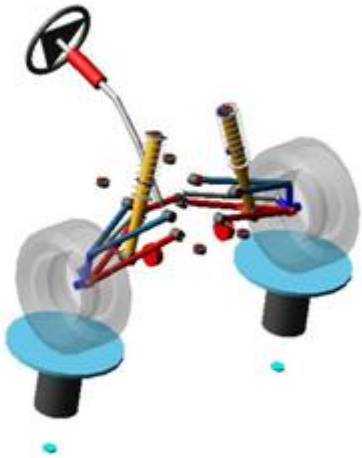

Figure 1. Front Suspension MSC Adams Assemblyunder steer characteristics and lead to loss of control and stability of the vehicle.

A.Front Suspension Design Optimization
Suspension

Short Long Arm Double A Arm were used as they provide optimum camber curves in wheel travel and the shorter upper arm helps to induce a camber curve which maintains maximum contact patch during rolling. During rolling motion the inner wheel undergoes droop while the outer wheel goes through bump motion. This decreases the contact patches in the respective wheels which is very important in any off road vehicle. This was achieved by the Short-Long Arm geometry. This configuration helps to maintain the contact patch maximum even in rolling condition. The inner wheel, undergoing droop, suffers a positive camber change while the outer wheel undergoes bump and gains negative camber. This helps in maintaining the wheels as close to vertical as possible, thus maximizing the contact patch and in turn the traction. The Roll-Camber coefficient decides the angle of the wheel with respect to body roll and thus determines the traction during rolling.

The suspension was designed so as to accommodate all the packaging factors and other parameters and keep a roll camber coefficient at all times.

The Roll-Camber Coefficient graph below is a one with negative slope and it is close to one in bump travel while it increases with droop travel. This helps in rolling motion as the weight of the vehicle is transferred to the outer wheel and the camber angle of the outer wheel remains close to the roll angle of the vehicle. This maximizes the contact patch the outer wheel supporting most of the weight of the vehicle

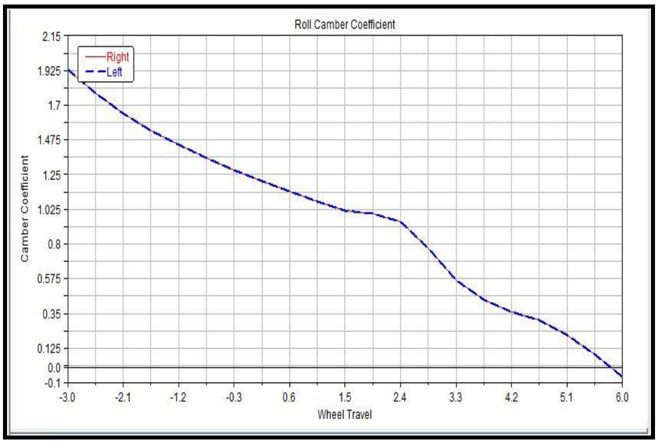

Figure 2. Roll-Camber Co-efficient

while the angle of the inner wheel, which is unloaded rapidly gains positive camber more than the roll angle thus further adding to the stability.The ICs were kept far enough to help minimize and also reduce camber change in heave motion. This also helped to keep the front RC at a height which allowed to incorporate the desired Roll Axis slope and migration maintain a rolling moment.[7] The RC was first decided to be 5.5 inches from the ground. This low RC helped to maintain an optimum stability between rolling and lateral force for the front suspension system. This enables an optimum load transfer between the two front wheels which maintains traction at the front end of the vehicle when in turns. This also helps to reduce the load transfer of the vehicle from the rear to the front which helps in inducing over steer characteristics by breaking the traction on the rear end at exit of corners.

The ground clearance was kept to be 13 inches from the ground so that after the driver's weight in the car it drops down and is 12 inches. The wheel were fixed and were of dimensions 22x7x12. The track was kept 55 inches and the

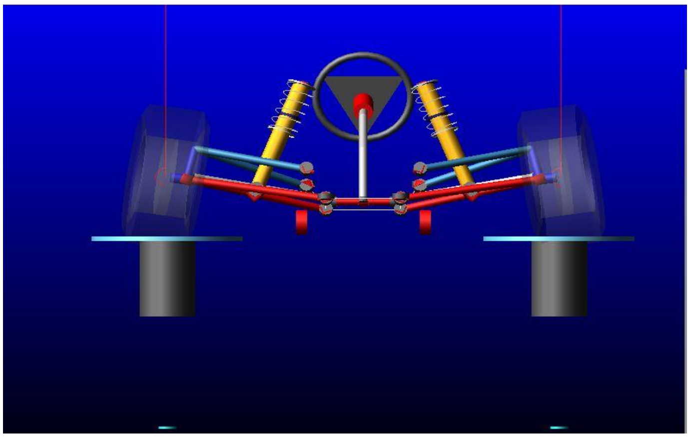

Figure 3. Front Suspension MSC Adams Simulation view

wheelbase 60 inches,to keep the arms as long as possible to minimize the variations in the geometry and also induce stability in the car along with shorter turning radius of 10 inches. The arms were inclined in the side view to keep a side view IC so as to make the shock effective in all 3 directions and also maximize load transfer to the shocks in bumps. The Kingpin Inclination was taken to be 6 degrees to reduce positive camber gain on travel. The spindle length was also taken into consideration, so as to push the lower ball joint as outward as possible, which is 4.5 inches. The lower ball joint point was kept at the center of the wheel from the side view to keep maximum ground clearance without increasing the inclination of the lower arms. The length of the upper arms was chosen such that enough negative camber was induced to get the optimum roll-camber coefficient.Toe In with bump travel was also induced in the geometry so that during sudden wheel movements. This gives directional stability as the vehicle has a very high tendency to deviate from its direction just after sudden bumps. The

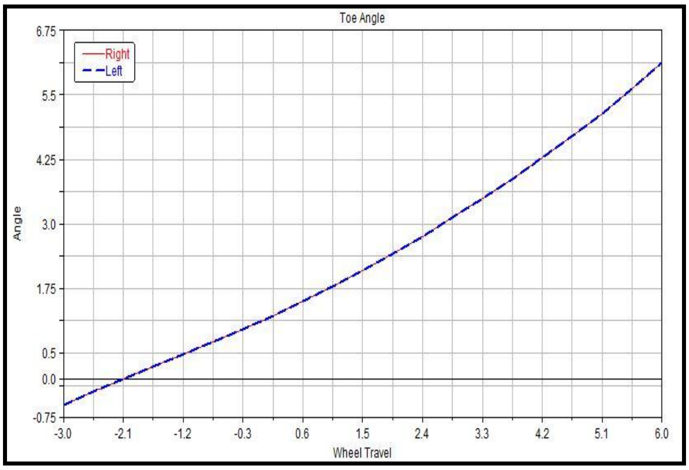

Figure 4. Toe Angle Variation with Wheel Travel

intentional toe in was incorporated such that it increases in an almost linear pattern and enhances the directional stability as the wheel travel or the bump movement increases..
The various results of the simulations were plotted in graphs, taking parameters with respect to wheel travel.

Steering Geometry

In off-road vehicle the turning radius is required to be small. This helps in maneuverability. Ackermann geometry is another aspect which helps in maneuverability at low speeds[7]. Thus 100% Ackermann was used in the geometry as the speeds dealt in off road buggies and other vehicles is generally low leading to lower unloading of the inner tyre and thus the Ackerman geometry plays an important role in the turning radius. With the true Ackerman the drag was reduced to a minimum, reducing under steer, and the turning radius was calculated out to be around 10 ft. at 55 degree max steering angle of the inner wheel and 33 degree max steering angle of the outer wheel at lock. The Ackermann angle came out to be 22.42 degree and steering knuckle arm was 2.46 inches long to minimize steering effort with this geometry.
Ackermann= tan-1(wheelbase/((wheelbase/tan(max outside tire angle))-front track))
Ackermann%=Inside tire angle/Ackermann [6]

The positioning of the rack was done such that bump steer could be minimized. Thus the tie rod were placed such that they point at the respective IC. The pivot points of the lower control arm were kept at the same distance from the center as the pivot points of the tie rods. This helped them to gain a center of rotation of the control arms and tie rods as close as possible and thus giving them an arc which almost coincides, thus reducing bump steer and roll steer, which cause instability and also may lead to failure of suspension or steering components.

Shock Absorbers

Single Direct Coil-over shocks were used from Fox. The model Podium X was selected as it gave us maximum adjustability and also long stroke length to provide large wheel travel. Motion ratio was chosen to be 0.5 so that the wheel travel becomes double that of the stroke length travel. The angle was chosen to be 55 deg so that the it was as near to vertical as possible and provided maximum effectiveness. The shock was kept perpendicular to the lower control arm in the side view so as to incline it in all 3 directions. The Spring Rate was calculated using a standard
.Calculator and as per the weight of the front end to be around 80 pounds (60 sprung and 20 un sprung) the desired value was calculated out to be around 50 lbs./inch. The length of the arm was taken to be 20 inches and the mount point was at the middle of the arm at 10 inches from the pivot point.

III. REAR SUSPENSION

In rear suspension design the basic goals are the same apart from the fact that the rear suspension should have less travel to incorporate the limited drive shaft travel. The angle of the tripod joint used in the inner end of the driveshaft doesn't allow a very large angular travel and the shaft may come out of the socket of the tripod joint when it exceeds its maximum angle. The driveshaft angle at static position is also an important factor that had to be considered. The clearances for the length of the links were considered as per the braking system used(outboard braking).Since most of the weight of the body is in the rear the suspension had to be designed to keep camber curve to provide max traction at rolling situations. The load transfers also had to be

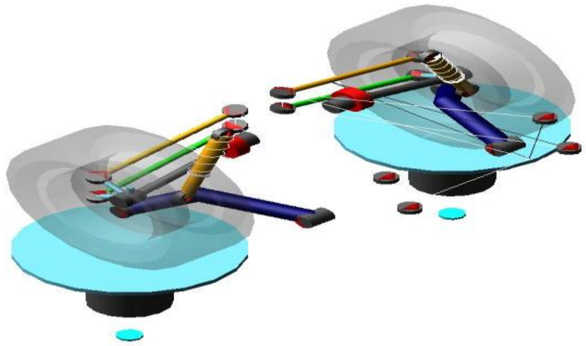

Figure 5. Rear Suspension MSC Adams Assembly

considered to evaluate the load on the outer wheel in turns. There are basically 3 forces acting on the center of gravity[2].

- Momentum on the rear and frontal axle due to load transference from the inner to the external wheel during the curve.
- Centripetal force due to suspended mass.
- Centripetal force due to non-suspended mass.

The load being very high at the rear the load transfer is quick and massive giving a lot of rolling motion to the rear part of the car which may lead to loss of stability and swaying away of the vehicle.

3-link suspension geometry consists of 2 camber links at the rear and one trailing link coming from the chassis. This type of suspension is basically a multi link suspension which combines the benefits of both trailing arm and Short Long arm suspension, thus giving ideal camber curve on rolling for off-road conditions and also providing the benefits of the trailing arms like large displacement course, easy to design and manufacture,high structural resistance and can also help in the movements of the wheel as per the desired[9]. Toe characteristics of the rear wheels is controlled by the two toe links which maintain the toe angle of the wheels and help reduction of wastage of power[9].

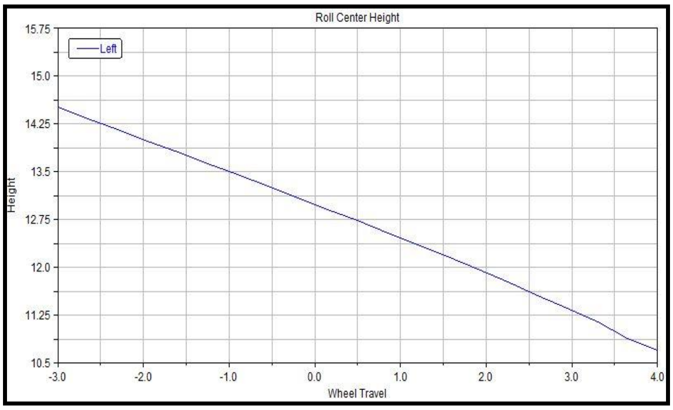

Figure 6.Roll-Center Height Variation with Wheel travel.

Thus the roll centre of the rear end was kept high enough so as to gain a balance between the lateral force at tyres and the rolling moment. This helps in inducing over steer and prevents heavy rolling which again helps in taking tighter turns by reducing the turning radius. The suspension geometry was setup in way such that the migration of the roll center was minimum at the rear thus inducing stability in the heavy rear part of the vehicle.

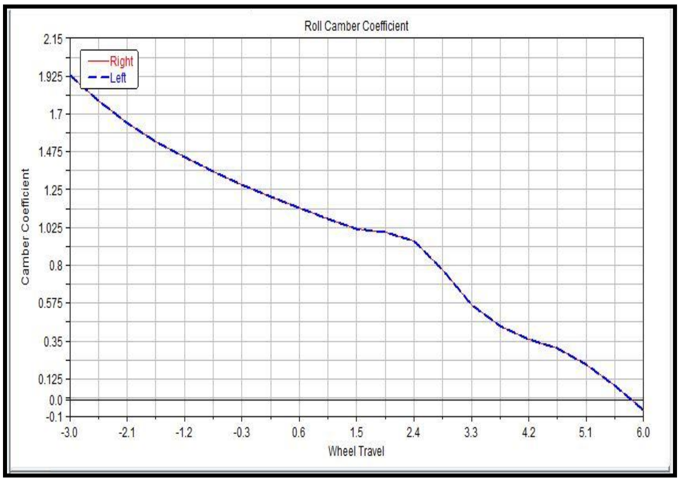

Figure 7.Roll Camber Co-efficient Variation with Wheel Travel

The camber angle of the rear suspension system was kept as low as possible to help gain maximum traction at all times by keeping the tyres as close to vertical as possible.

The figure is shown below -

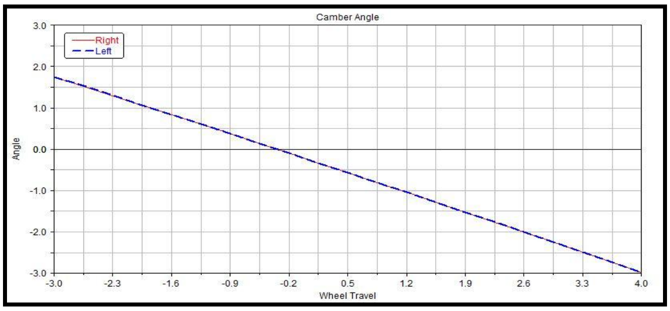

Figure 8. Camber Angle variation with Wheel travel.

A. Rear Suspension Design Optimization

Shock Absorbers

The Positioning of the shocks in the rear is done such that the shocks are inclined in the direction of the vector where the forces act thus making the shocks more efficient. The shocks in the rear were kept stiffer than the front. 3-link suspensions helps in providing the exact same. Its arm running from hub to the RRH gives the desired mount positions to support and maintain a proper ground clearance even during bumps.

Damping Ratio

The damping ratio, usually designated as ζ, is defined as the ratio of actual damping coefficient to the critical damping coefficient. The reason why we work with damping ratios instead of actual damping coefficients is so that we can normalize the discussion for all dampers[5].Choosing a damping ratio is a generally a tradeoff between response time and overshoot (you want to minimize both). Typically, passenger cars will use a damping ratio of around 0.25 to maximize ride comfort. For an off-road car, the damping must be considerably higher for road holding and control of the un-sprung mass motion. Data has shown that for off road cars, a good baseline for damping ratio is between 0.65 and 0.70.

IV. CONCLUSION

Designing and optimization of a off-road Buggy car was done using simulation and analysis with softwares. In this project, a suspension was designed in a way which would help in making a vehicle provide resistance to all impact loads. The compatibility of the A-arms and 3-link suspension with the detailed parameters has produced exceptional results in the graphs of camber and caster variations , toe angles , Ackermann geometry , proper flow of forces from chassis to ground, bump steer and shock absorber characteristics. This ensures that the off-road vehicle would improve its perceived quality of its dynamic performance and would provide good driver satisfaction in concert with excellent vehicle packaging. The same design could be used in other off-road vehicles like Forest Rangers, Military Cars and Trucks, even in many Passenger Sports Utility Vehicles with slight variations as per the vehicle specifications and usage. The results obtained from the design are convincing enough to call it successful. Thus the Double A Arms can be used for most of the off-road vehicles.

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