Design report

Vehicle Dynamics

August 15, 2019

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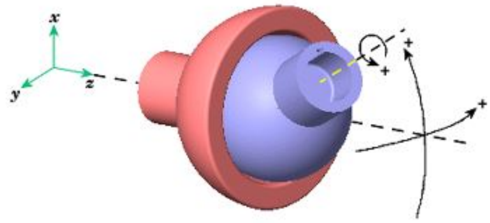


Figure 1: Spherical joint with 3 rotation degrees of freedom

1 Introduction

The purpose of a vehicle's suspension is to maintain the tires contact with the road and ensure that the tire remains in its optimal position at all times . The suspension must allow enough wheel travel to absorb bumps in the surface and to allow the chassis and the sprung mass, which are the components supported by the springs, to move under accelerations. The suspension geometry does this by controlling the path of relative motion between the wheel and the unsprung mass, which are the components of the vehicle not supported by the springs, and the sprung mass, and allows for the transfer of forces between them.

The suspension design chosen for B19 is the double wishbone (or upper and lower A-arm) suspension in the front and Five link suspension in the rear both of which are an independent design. For an independent suspension, be it front or rear, the assemblage of control arms is intended to control the wheel motion relative to the car body in a single prescribed path. In this wheel has a fixed path of motion relative to the car body. It is not allowed to move fore and aft or laterally relative to this path. The knuckle is not allowed to rotate other than as determined by this fixed path. The linkages are used to correctly position the knuckle both in static as well as dynamic conditions. In front suspensions we do have a steer rotation degree of freedom but only when it is demanded from the steering system. For solving this we need to put constraint on the transnational and rotational degree of freedom . And the suspension provides five degrees of restraint i.e., it severely limits motion in five directions. Hence only jounce and droop only.

The kinematic behaviour of a multi-link suspension follows from the connection of the upright to the chassis with six connection rods. Every connection rod has 2 joints which can be placed arbitrary in space with 3 coordinates each. This results is a total of $6{\cdot}2{\cdot}3 = 36$ degrees of freedom on one wheel (assumed that that car is left/right symmetric). The front and rear suspension have in total 72 degrees of freedom. These freedoms are restricted by several design boundary's, such as the available space. Both Front and Rear belongs to the independent suspension layouts since it has only one bump degree of freedom, where rigid-axle suspension have two bump degrees of freedom (symmetric and asymmetric bump movement).The ball-joints at each end constrain a movement with 3 translations and leave 3 rotation degrees of freedom available as shown in figure 1. This results in a kinematic chain which constraints the upright movement. Being all spatial bodies, the upright and each connection rod possess six degrees of freedom in a 3D space. This results in a total of 6 bodies: 5 fixed connection rods and 1 upright. The total degrees of freedom equals $f_{\text{tot}} = 6 * 6 = 36$. A spherical joint reduces the total degrees of freedom by $f_{\text{spherical}} = 3$. Each individual rotation of a connection rod around its own axis,r, does not contribute to the total amount of degrees of freedom. With joints, the balance of degrees of freedom of the suspension results in:

$$DOF = f_{\text{tot}} - r - g * f_{\text{spherical}} = 36 - 5 - 10 * 3 = 1 \tag{1}$$

Depending on the omitted connection rod this results in 1 degree of freedom for the bump or steer movement. This characterizes an independent suspension.

2 Tires

Bias-ply tires were the standard for many years. Bias-ply tires lay down their body cords (typically made of a fabric like nylon, rayon, or polyester) from bead to bead, running across the tire in alternating layers. Each successive ply is laid at an opposing angle, forming a cris-cross pattern. The flex allows for better gripping, better clean out, and conformity to rough and rocky terrain. The flexing also increases rolling resistance, and less control and traction at higher speeds.

Radial tires are made differently. Their body cords run across the tire radially - directly from bead to bead. Atop those first thin layers, alternating layers of body cords are stacked in the tread portion of the tire only, so the sidewalls aren't as thick as they are on a bias-ply tire.The figure 2 show the difference in construction between bias-ply and radial tires.

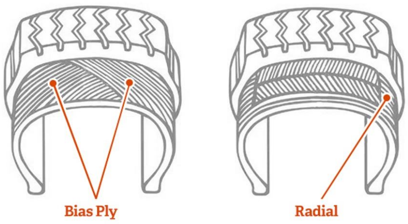


Figure 2: Difference in construction between bias-ply and radial tires.

The bias tires are Good in rough terrain and off-road and have Strong sidewalls, tough casing, Good lateral stability. Moreover they have Lower purchase price. Since high speeds are not easily in off-road competition and the car is meant to be driven for around 50 km hence Bias tires were considered to be suitable.

Another Factor which improves the performance of a an off-road vehicle is keeping the front tire smaller than rear.

Advantage of smaller Front tires

- Better traction on the ground. You can imagine this well with a tractor, whose front wheels are too small as compared to the rear ones. Suppose your vehicle is stuck between the muds, concrete blocks, logs of wood this system will easily get out of these situations, while others with same rear and from tyre diameter will have to push the accelerator hard, in order to achieve that.
- Small tires at front are for easy maneuverability, lesser scrub, and lesser weight and steering effort.
- Large tires are wider and generally wider tires are soft. Increasing coefficient of friction and thus the max traction force your car can exert without slipping increases and thus allows u to have more reduction and a better tractive force. And thus, helps in hill climb
- On the rear end, larger tires will transfer larger torque ($Power = torque \cdot w$), which helps to accelerate and better traction ensures that the vehicle isn't stuck anywhere.

But due to larger size of front brake rotor there is lower limit to the front tires size and also smaller tires reduces ride height. To counterbalance the ride-height factor we can increase the angle the A-arms make in the front view but the problem arises with the maximum articulation angle which spacers can provide thereby limiting the maximum bump. The other option is to increase the rear tire radius but this increases effective overdrive ratio reducing maximum speed.

3 Suspension

3.1 Goals

For front suspension the following goals were targeted
Front

- To minimize the bump steer
- Small Negative camber gain on outside tire with steer.
- Preventing the camber with wheel travel and steer lock from getting too negative.
- Low anti-dive percentage.
- Roll center above the Tire center.
- Low compliance in steering :- To Minimize the compliance in steering the tie rod should not be kept angled in front view with the X-axis.

Low Camber angle are desired inorder to maximize traction with the ground and since the tires are 'Bias' the camber stiffness is small as compared to cornering stiffness and hence we cannot gain much advantage during cornering from negative camber gain.

3.2 Selecting the suspension Geometry(incomplete yet..)

3.3 Front

For Front Suspension the geometries we considered are:

MacPherson strut suspension: This geometry has a pair of Lower A-arms with damper mounted on knuckle ,(tie rod as usual) as shown in figure. This type of suspension is commonly used in passenger cars. though it makes knuckle manufacturing difficult, it can lead to weight reduction. moreover there is ample space available for steering geometry. It's major drawback is that it lead to excessive (negative) camber changes with bump which is undesirable in baja.

Figure 3: MacPherson strut suspension

Double wishbone In this geometry we have 2 Options.

- Conventional Double A arm suspension using Real Pivot points
- Five Link Suspension using Virtual Pivot point: Allows more freedom in steering geometry but difficult to manufacture as there small errors in manufacturing can lead to significant deviations.

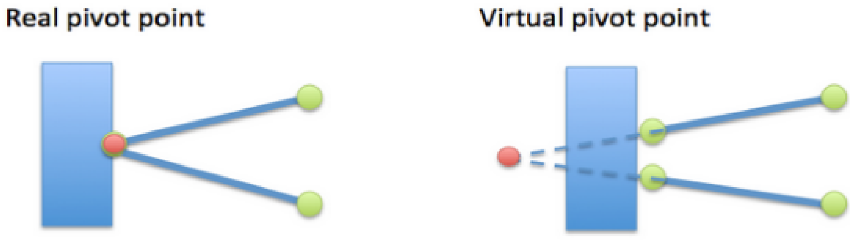


Figure 4: Real and virtual pivot

As Double wishbone suspension can be designed to meet our goals we decided to use this geometry for front.

3.4 Rear

Goals

- Minimize the camber change and Toe change with bump and droop to keep maximum traction with the ground.

Suspension selection For rear Options available are:

Pure-Trailing arm : There is no toe change, camber change at all. In terms of the number of links used there is only one, the arm. This puts heavy structural requirements on this arm which must be strong in bending in all directions to resist braking torque, camber torque, and steer direction torques. Packaging will be difficult. Less space for engine, gearbox. The roll center is at ground hence to reduce excessive roll anti-roll bar has to be used. It's common to see winning teams using this type of suspension geometry.

Semi Trailing arm suspension: This is similar to the pure trailing arm, the only difference is that the bushing axis (and "instant axis") now can run at an angle in all three views, as shown in Figure. This type of suspension can solve the problem of packaging but it has both Toe and camber changes with bump and droop. We can add camber links to control camber changes.

Double Wishbone This type of suspension geometry can be designed to meet our goals but it's major drawback is that no nodes go to roll hoop. All the nodes are on rear bracing which is generally made of secondary members. Adding nodes there would require extra support which will increase the weight of chassis. Another drawback is packaging of Drive-shaft and damper.

Five link suspension Though complicated, this geometry gives a lot of freedom in designing. Our design was inspired from an ATV *Arctic Cat Wildcat 1000*. The vehicle had a rear geometry as shown in figure. The three parallel links controlled toe and camber change. Damper was mounted on Lower 'Semi Trailing type arm' with anti roll bar attached to it. These lower 'trailing arm' bears most of the forces. Both the trailing arms along the =========.

3.5 Numerical Optimization

The kinematics of a suspension concept are normally designed according to a kinematic design target. This design target consists of an objective for the calculated suspension parameters. This can be ac-

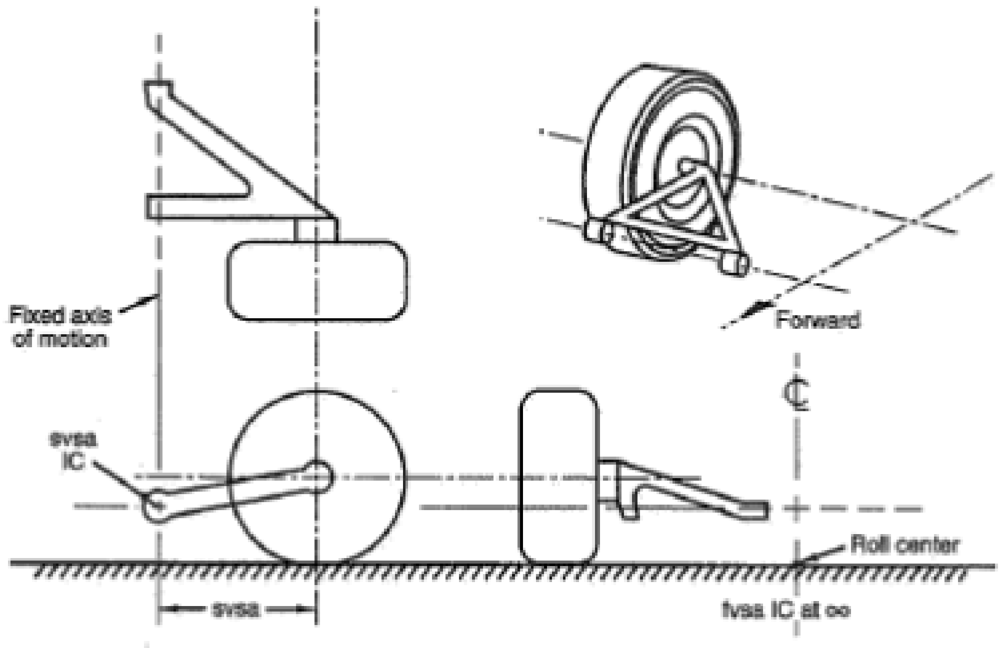


Figure 5: Pure-Trailing arm

complished by relocation the suspension coordinates. A multi-link suspension has, as discussed earlier, 36 degrees of freedom. The design target consists often of less objectives. But the available space and vehicle geometric layout restricts the range of adaptation of a degree of freedom.

3.5.1 Unconstrained minimization

The suspension design has a lot of geometrical constraints. For this reason it looks obvious to define a constraint optimization problem. A constraint optimization problem is characterized as a problem where the optimization variables (inputs) can not be freely and independently chosen. Mathematical equations restrict the choice of inputs. This can be expressed as equality or inequality constraints. An equality constraint has to be exactly fulfilled by the optimization algorithm. This could be, for example, a constraint which says that the z-coordinates of two suspension points have to be equal. An inequality constraint has to be equal or smaller than zero. An inequality constraint can, for example, be used to constrain the suspension coordinates in such a way that a connection rod would not intersect with the wheel rim. It sounds obvious to do this, but this would be mathematically very complex. Especially since the suspension is moving in 3d space regarding two states of motion (bump and steer).

It is also possible to set a boundary for the suspension coordinates (input variables) when using an unconstrained optimization proces. This simplifies the definition of the optimization proces a lot. The boundary sets a valid region for the suspension coordinates. This defines a box shaped boundary for every suspension point since a suspension point consists of 3 degrees of freedom (3 translations in x,y and z direction). This means that every variable can still be chosen independently. This could result in a design which does not satisfies the space restrictions. A new design iteration has to be done when this happens.

Two multi-variable algorithms are compared to find the most effective one. The first algorithm is called the Broyden-Fletcher-Goldfarb-Shanno (BFGS, MATLAB : fminunc) method and the second the generalized pattern search (GPS, MATLAB : patternsearch) algorithm. The last one appears to be in most cases the fastest and most robust algorithm since it does not depend on local gradients. The code in appendix 1 uses GPS algorithm and optimizes the self defined cost function for a double wishbone suspension. Future work could be done to extend it for a Five link suspension.

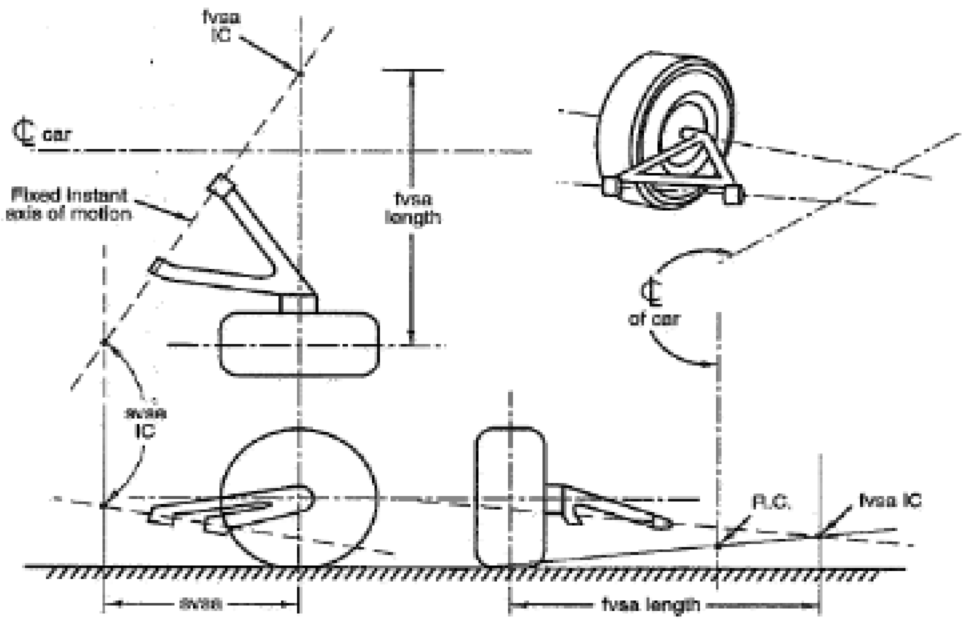


Figure 6: Semi Trailing arm suspension

Figure 7: Arctic Cat Wildcat 1000 rear suspension

3.6 Designing Front Suspension

3.6.1 Bump Steer

Bump steer is the change in Toe angle due to wheel travel in bump and droop. For good driver ergonomics, Bump steer should be zero. Imagine the case when a person is driving a the car on a uneven track, If the bump steer is not zero then the path of car will not be predictive. Although bump steer cannot be minimized at all steer angles, Hence Bump steer at zero steer angle is given utmost importance.
To achieve zero bump steer during the whole range of wheel travel the tie rod should pass through Front View instantaneous center. Only way to achieve this is to make the front view instantaneous center at infinity and tie rod parallel to the A-arms in the front view.

Figure 8: Arctic Cat Wildcat 1000 rear knuckle

3.6.2 Steering Parameters

The steering parameters include Steering axis inclination, castor, Scrub radius, mechanical trail. These parameters determine the camber change with steering, steering feedback, steering effort, overturning moment.

3.6.2.1 Scrub Radius Large scrub radii may be used so that at low speed (parking) the wheels can "roll around" the kingpin axis. This dramatically lowers the steering effort when compared to scrubbing the tire with Center Point steering, steering while standing still.
Since in rear wheel drive the force on front wheel during braking and acceleration are in backward direction so a positive scrub radius will cause a toe out and negative scrub radius will cause toe in.

Zero Scrub Radius

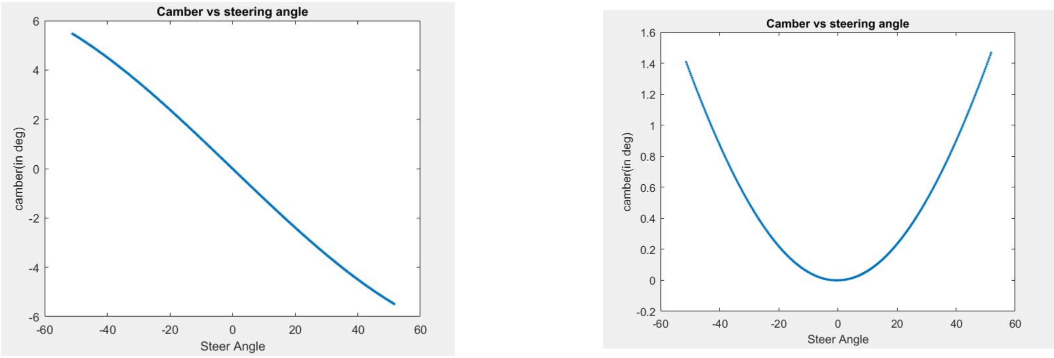
- Large steering effort needed.
- The advantage of a small scrub radius is that the steering becomes less sensitive to braking inputs. However, zero scrub radius, under hard braking, causes the suspension to be skittish because varying road conditions created varying amounts of torque (both positive and negative) around the steering axis. Therefore, some amount of scrub radius, positive or negative, is preferred.

Positive Scrub Radius

- If one drive wheel loses traction, the opposing wheel will toe in an amount. This will tend to steer the car in a straight line, even though the tractive force is not equal side-to-side and the unequal tractive force is applying a yaw moment to the vehicle. If you hit standing water at speed on one side of the car, the torque on the steering will steer you away from the puddle. Which balances the effect of drag on one side of the car.
- Causes a toe-out force

Negative Scrub Radius

- An advantage of a negative scrub radius is that the geometry naturally compensates for split braking, or failure in one of the brake circuits. If one half of the brake system fails, then the vehicle will tend to pull up in a straight line. Vehicles with a diagonal-split brake system have negative scrub radius built into the steering geometry.



(a) Zero KPI

(b) Zero Castor

Figure 9: Camber with Steer angle

- Causes a toe-in force

3.6.2.2 KPI and Castor Both the castor and kingpin inclination result in a camber change during steering. With no KPI and Castor there will be no change in camber with steering. If Positive Castor is added but no KPI then the wheel will negative camber results for outside and positive camber on inside wheel (Fig a). If the KPI is positive and zero castor, when a wheel is steered, both inside as well as outside wheel will lean out at the top, toward positive camber (Fig b). A positive KPI contributes to a positive camber change during steering on the outside wheel. The amount of this effect is small, but significant if the track includes tight turns. The reason that low kingpin angles are desirable is that although KPI subtracts from the negative camber gain due to caster on the outside wheel but it adds positive camber to the inside wheel.

To determine the Kingpin inclination angle and Castor angle the following model was designed.
Let the components of angular velocity about steering axis be $\omega_x, \omega_y, \omega_z$
Let the Kingpin Inclination be σ, Castor angle be τ and steer angle be δ
Using Relations

$$\tan\sigma = -\frac{\omega_y}{\omega_z} \tag{2}$$

$$\tan\tau = \frac{\omega_x}{\omega_z} \tag{3}$$

$$\omega^2 = \omega_x^2 + \omega_z^2 + \omega_z^2 \tag{4}$$

we get

$$\omega_z = \frac{\omega}{\sqrt{1+\tan^2\sigma+\tan^2\tau}} \tag{5}$$

$$\omega_x = \frac{\omega\tan\tau}{\sqrt{1+\tan^2\sigma+\tan^2\tau}} \tag{6}$$

$$\omega_y = \frac{-\omega\tan\sigma}{\sqrt{1+\tan^2\sigma+\tan^2\tau}} \tag{7}$$

If camber angle is γ then we have to rotate our system about X-axis such that Y axis perpendicular to plane of tire then

$$RotationMatrix = \begin{bmatrix} 1 & 0 & 0 \\ 0 & \cos\gamma & \sin\gamma \\ 0 & -\sin\gamma & \cos\gamma \end{bmatrix}$$

then we rotate ω_x and ω_y about ω_z using the below rotation matrix such that ω_x' is always perpendicular to the tire and ω_y' is always along the axis of tire.

$$RotationMatrix = \begin{bmatrix} \cos\delta & \sin\delta & 0 \\ -\sin\delta & \cos\delta & 0 \\ 0 & 0 & 1 \end{bmatrix}$$

$$FinalRotationMatrix = \begin{pmatrix} 1 & 0 & 0 \\ 0 & \cos\gamma & \sin\gamma \\ 0 & -\sin\gamma & \cos\gamma \end{pmatrix} \begin{pmatrix} \cos\delta & \sin\delta & 0 \\ -\sin\delta & \cos\delta & 0 \\ 0 & 0 & 1 \end{pmatrix}$$

Since ω_x' is always perpendicular to the tire only this component will cause the camber change.

Hence Camber change rate $\frac{\partial\gamma}{\partial t}$

$$\frac{\partial\gamma}{\partial t} = -\omega_x \cos\delta - \omega_y \sin\delta \tag{8}$$

This gives

$$\frac{\partial\gamma}{\partial t} = |\omega| \frac{-\tan\tau\cos\delta + \tan\sigma\sin\delta}{\sqrt{1+\tan^2\tau+\tan^2\sigma}} \tag{9}$$

where

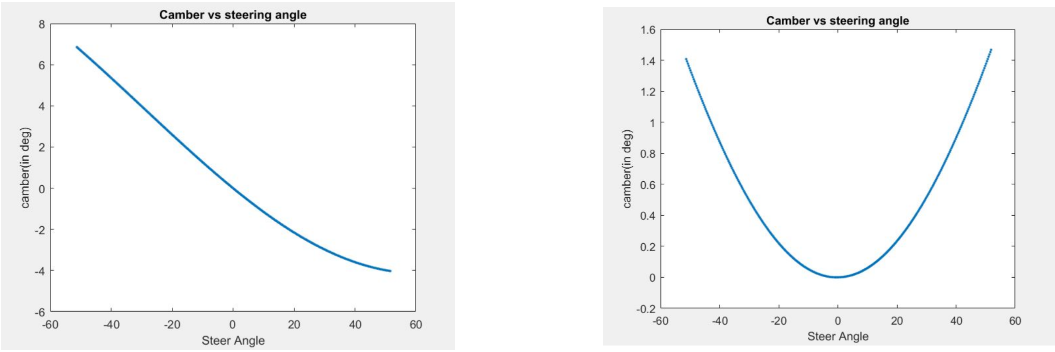
$$\frac{\partial\delta}{\partial t} = \omega_z' = -\sin\gamma(-\omega_x\sin\delta + \omega_y\cos\delta) + \omega_z'\cos\gamma \tag{10}$$

$$\frac{\partial\delta}{\partial t} = |\omega| \frac{(\tan\tau\sin\delta + \tan\sigma\cos\delta)\tan\gamma + 1}{\sqrt{1+\tan^2\tau+\tan^2\sigma}} \tag{11}$$

Combining both we get

$$\frac{\partial\gamma}{\partial\delta} = \frac{-\tan\tau\cos\delta + \tan\sigma\sin\delta}{(\tan\tau\sin\delta + \tan\sigma\cos\delta)\tan\gamma + 1} \tag{12}$$

The highest camber change rate during steering is applied to the neutral position of the vehicle. This is done to prevent large camber angles at steering lock.This is especially usable at the front suspension, since the rear suspension is not steered. The bump steer amount at the rear it that small that it does not contribute to the camber change.
Using the above theory, a code was written in MATLAB and iterating for different values of KPI and Castor. It was found that a positive camber gain on the inside tire increased if KPI is increased. Hence it was thought to keep KPI as zero but it resulted in large scrub radius around (70 mm). Hence it to keep the scrub radius minimum but still positive.



(a) 6deg KPI and 7deg Castor

(b) Camber change Rate

Figure 10: Camber with Steer angle

Keeping the decide KPI and Castor and Due to size of Tire and packaging constraints the co-ordinates of UBJ and LBJ are fixed.Now the next job is to decide Chasis Mounting points.

3.6.3 Anti-Dive

The anti-dive percentage on the front is of importance when a brake maneuver is considered. The percentage relates the amount of longitudinal brake force which is transmitted trough the suspension links and the bump springs.The braking force is totally transmitted trough the suspension links and not trough the suspension springs when 100% anti-dive is applied. However, this will never be applied to racing vehicles since it will disturb the vertical tyre load because road irregularities cause an excitation of the vertical tyre dynamics trough the relative stiff and undamped suspension links. Lowering the stiffness felt at the contact patch results in a better vertical Tyre load fluctuation and therefore more grip. Anti-dive percentages lower than 50% are common for racing applications to compromise 'grip' and the amount of dive at the front.

3.7 Steering Dynamics

To maneuver a vehicle we need a steering mechanism to turn wheels. Steering dynamics introduces new requirements and challenges

3.8 Ackermann vs Anti-ackermann Geometry

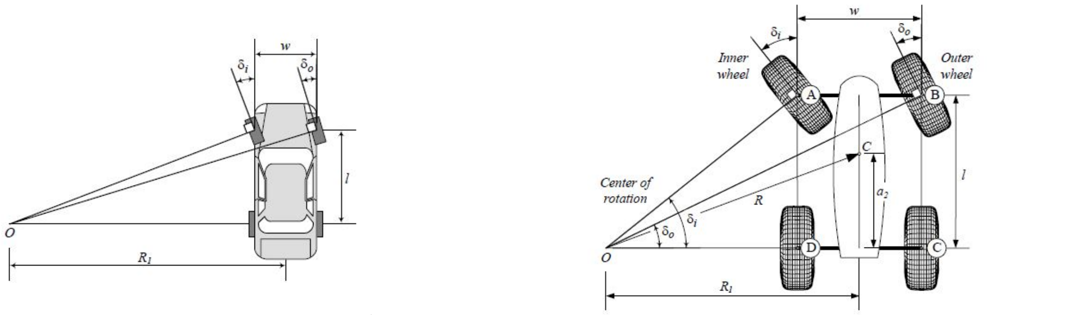
In Ackermann steering the outside wheel turns less than the inside wheel while in anti-Ackermann, the outside wheel turns more than the inside wheel during cornering. Anti-Ackermann geometry is employed in "high speed" vehicles where in during cornering a lot of weight shifts to the outer tyre, thus a higher slip angle and the inner tyre tends to lose its grip. While in "low speed" vehicles Ackermann steering geometry, there is not a significant dynamic weight transfer on the outer wheels during cornering and the inner wheel has a firm grip on the ground.

3.9 Ackermann steering

Consider a front-wheel-steering 4W S vehicle that is turning to the left, as shown in Figure 7.1. When the vehicle is moving very slowly, there is a kinematic condition between the inner and outer wheels that allows them to turn slip-free. The condition is called the Ackerman condition and is expressed by

$$\cot \delta_o - \cot \delta_i = \frac{w}{l} \tag{13}$$

where, δ_i is the steer angle of the inner wheel, and δ_o is the steer angle of the outer wheel. The inner and outer wheels are defined based on the turning center O.Track-width w and wheelbase l are considered



(a) A front-wheel-steering vehicle and the Ackermann condition.

(b) A front-wheel-steering vehicle and steer angles of the inner and outer wheels.

Figure 11: Ackermann Geometry

as kinematic width and length of the vehicle. The mass center of a steered vehicle will turn on a circle with radius R,

$$R = \sqrt{a_2^2 + l^2 \cot^2 \delta} \tag{14}$$

where δ is the cot-average of the inner and outer steer angles.

$$\cot \delta = \frac{\cot \delta_o + \cot \delta_i}{2} \tag{15}$$

The angle δ is the equivalent steer angle of a bicycle having the same wheelbase l and radius of rotation R.

3.10 Rack and pinon steering

Rack and pinon steering geometry is one of the way to design steering geometry which is based on ackerman principle. Here is a list of various steering geometry parameters in case of rack and pinion geometry.

- x= steering arm length
- y= tie-rod length (in top view)
- p= rack casing length
- p + 2r= rack ball joint center to center length
- q= travel of rack
- d= distance between front axis and rack center axis
- β = Ackermann angle

From Zero Toe condition

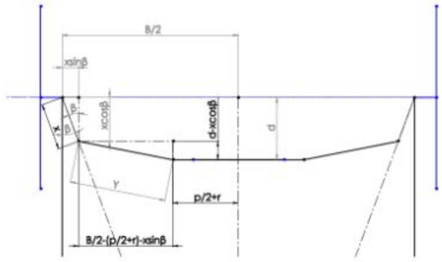
$$y^2 = [\frac{w - (p + 2r)}{2} - x \sin \beta]^2 + [d - x \cos \beta]^2 \tag{16}$$

From inner wheel geometry

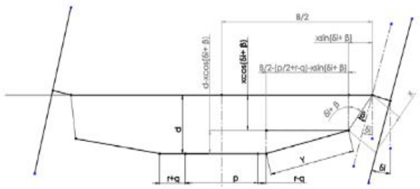
$$y^2 = [\frac{w - (p + 2r - 2q)}{2} - x \sin(\delta_i + \beta)]^2 + [d - x \cos(\delta_i + \beta)]^2 \tag{17}$$

From outer wheel geometry

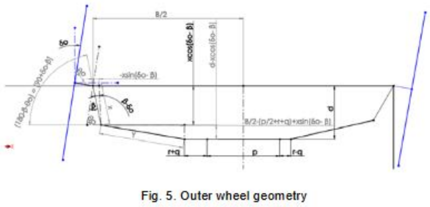
$$y^2 = [\frac{w - (p + 2r + 2q)}{2} + x \sin(\delta_o - \beta)]^2 + [d - x \cos(\delta_o - \beta)]^2 \tag{18}$$



(a) Toe zero condition



(b) Inner Wheel Geometry



(c) Outer Wheel Geometry

Figure 12: Rack and pinion Geometry

Solving these three equations we get

$$\delta_o = \arcsin(c(A-q) + \sqrt{c^2(A-q)^2 - (A-q)^2 + d^2c^2 - d^2}) + \beta \tag{19}$$

where
$A = \frac{w-(p+2r)}{2}$ and $c = \frac{y^2-d^2-x^2-(A-q)^2}{2x}$

3.11 Steering Mechanism Optimization

Optimization means steering mechanism is the design of a system that works as closely as possible to a desired function. Assume the Ackerman kinematic condition is the desired function for a steering system. Comparing the function of the designed steering mechanism to the Ackerman condition, we may define an error function e to compare the two functions. An example for the e function can be the root mean square (RMS) of the difference between the outer steer angles of the designed mechanism $\delta_D o$ and the Ackerman $\delta_A o$ for the same inner angle δ_i.

$$e = \sqrt{\int (\delta_D o - \delta_A o)^2 d\delta_i} \tag{20}$$

Data:-

- l, wheelbase = 1.47m
- x= steering arm length = 0.10m
- p + 2r = rack ball joint center to center length = 0.452m
- d= distance between front axis and rack center axis = 0.075m
- w = Track width = 1.117m

Using the given values for different parameters, we get the following graphs: Appendix 1 [language = Matlab]

Appendix 2

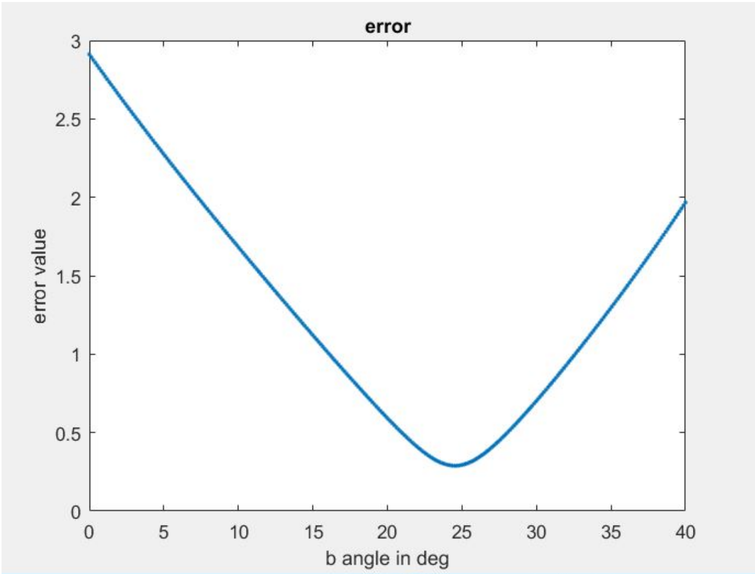


Figure 13: Error vs β angle

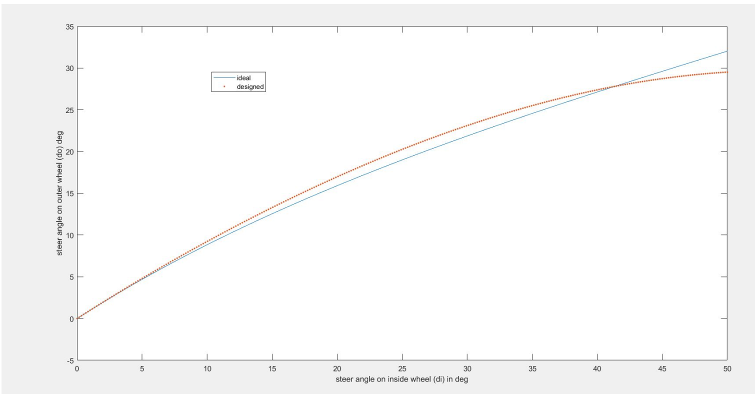


Figure 14: ideal and designed ackermann geometry

3.12 Conclusion

Suspension

Parallel equal length double-wishbone suspension, with damper attached to the lower A-arm, was selected for the front. Tie rod is kept parallel to the A-arm in order to minimize the bump-steer for all values of wheel travel making the drive more predictable in rough terrain. Equal length A-arms were selected to prevent too much of negative camber gain during bump with steer-lock, which is a common affair on BAJA race track during cornering. Too much of negative camber will reduce traction and will cause tires to wear out with concentrated heating effect.

For rear suspension geometry, a Five-link suspension was selected constraining the wheel path so as to achieve zero-toe change and zero-camber change during bump simultaneously. This geometry had an additional advantage of having an appropriate rear roll-center height which is not too low to cause large rolling moments and not too high to induce jacking forces. Without making any compromises, the suspension geometry also allowed to keep the roll axis below CG giving a positive feedback to the driver during cornering and This also ruled out the use of Anti-roll bars at the rear as the roll center height was controllable.

Design was initiated by selecting tires and determining the camber change at steer lock for different sets of values of Castor and KPI (Kingpin Inclination). The graphs for Camber-gain vs steer angle for different values of KPI and Castor were generated through a MATLAB code implementing a rotation matrix to obtain camber gain as function of steer-angle. The results favoured low KPI angles but it resulted in large scrub radius. Hence appropriate values for KPI and castor were selected which gave Scrub radius of (30mm @soumik) which is neither too large to give excessive forces at steering and also not too small to increase steering effort and negligible steering feedback. For Side view geometry the anti-dive percentage was also kept zero to direct the forces towards CG during braking. By keeping in mind, these parameters, suspension geometry was designed. After initializing the geometry, simulations were done for stress and deformation analysis in A-arms, tie rod, knucle and mounting points on the chassis so as to establish properly effective force distribution in the suspension arms against yield and buckling failure. A MATLAB code was generated and the forces on front suspension components were calculated considering cornering and braking Drop from height of 1.5m and front impact due to bump (such as log of wood), and for the rear these were calculated considering cornering and acceleration and impact due to bump.