Design of a suspension system and determining suspension parameters of a medium downforce small Formula type car

Sadjyot Biswal[1], Aravind Prasanth[1], Dr. R Udayakumar[2], Shobhit Deva[1], Aman Gupta

[1]*Automotive, Mechanical Department, BITS Pilani Dubai Campus, DIAC, 345055*
[2] *Mechanical Department, BITS Pilani Dubai Campus, DIAC, 345055*

Abstract. The focus of the paper is on designing a suspension system for a medium downforce small Formula type race car. The paper not only focusses on step by step design for a double wishbone type suspension but will also show the use and role of kinematics software in determining the optimized suspension of the car. The paper will also focus on the use of tire data in determining suspension parameters and the design of the double wishbone suspension. Various parameters, their design importance and the process to optimize them according to suspension design goals will be covered. The easiest and best ways to change the suspension parameters to get the best results will also be covered.

1 Introduction

The suspension system is one of the most important components of a race car because it not only connects the sprung mass to the chassis but is important for maintaining maximum tire contact patch at all times. Along with the tires and steering geometry, it plays a very crucial role in determining the vehicle dynamics of a race car. The objective of this paper is to present a step-by-step design for a good suspension system of a low downforce small formula type race car such as those used in Formula student. The paper will also talk about the parameters, which will affect suspension design; their optimization and the use of kinematic software to further improve the designs.

2 Requirements

Formula student competition like F1 uses wishbone suspension system due to its high structural strength and its ease of track tuning. The main requirements for the suspension system are:

- It should have a minimum travel of 25.4 mm in both jounce and travel.
- The system must be responsive and also stable enough to give best performance in cornering and straight line.
- The system should be such that it should help maintain maximum tire contact patch for better friction.
- It must be structurally strong but also lightweight to maintain low unsprung mass.
- It such be such that major suspension design parameters can be tuned for on track testing.

3 Design

Certain parameters need to be decided while designing a suspension geometry. The first things to be decided are the trackwidth and wheelbase. The wheelbase will affect primarily two things:

- The longitudinal weight transfers
- Steering angle required

Trackwidth can be decided also based on a few parameters. Trackwidth affects lateral weight transfer too according to the following formula:

$$F_{zr} = 0.5\rho mg + \frac{\theta_{roll}K_f}{t_f}$$

$$F_{zl} = 0.5\rho mg - \frac{\theta_{roll}K_f}{t_f}$$

According to the formula increasing track width will reduce lateral weight transfer but having a bigger track width might cause trouble in tight cornering. The difference in front and rear track width values is due to the fact to give the driver more freedom in not hitting the cones in the race events during Formula student with the rear of the car.

Other important factors to be considered are inertia and weight. Larger the wheelbase and trackwidth higher is the inertia and also weight of the car. Both higher inertia and weight are undesirable characteristics and the aim is to keep both low

Combining all the above factors, the values have been chosen accordingly:
Wb= 1665mm
Front Track= 1204mm
Rear Track= 1150mm

3.1 Outboard suspension pickup points

Once the wheelbase and trackwidth has been decided, the outboard suspension pickup points need to be decided. The main criteria for deciding the outboard pickup points is the packaging. The rim diameter, it's thickness, the hub and upright, brake rotor and calliper all affect the suspension design.

Except this the distance between the two outer ball joints on the upright also affect roll centre migration. The graph showing the trend is shown below.
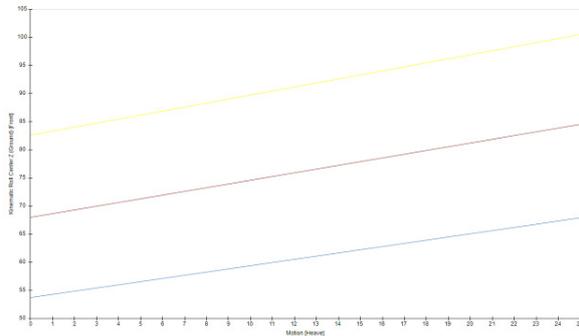

Figure 1. Roll Centre height migration in heave

From the graph, it can be inferred that as the distance between the ball joints increases the roll centre migration increases proportionally

The first step is to create the 2-d geometry both in front view and side view. The two outermost points are located keeping in mind all the above-mentioned packaging constraints. Hence the points are decided in front view. The points are kept as far away from each other to provide higher structural strength. The ratio between lower to upper wishbone length also plays an important part in determining the roll centre migration. However, if the kingpin angle and scrub radius get fixed the lengths can be altered only by changing the inboard pickup points.

The kingpin inclination and scrub radius also play an integral part in designing the front view. The focus is to maintain a scrub radius as low as possible to avoid scrubbing of tires in tight corners. The kingpin causes camber variation in steering so a kingpin value of 8 degrees can be chosen after analysis in Kinematics software.

3.2 Side view geometry and caster

In side view the points are not in a straight line to give the front wheels some caster. As a rule of thumb the steer moment should not be more than 10 Nm and should be near 5 Nm caster and mechanical trail affect the steer torque along with the pneumatic trail.

The steer torque is calculated as follows:

$$\text{Steer Torque} = \frac{\Sigma(M_2, M_y, F_y, F_x, F_z)}{\text{Pitman Arm Trail}} \times \text{Steering Rack trail}$$

$$\Rightarrow \text{Steering Rack trail} = 0.025m$$

$$\Rightarrow \text{Pitman Arm trail} = 0.2m$$

$$\Rightarrow \text{Steer Torque out} = \frac{35\ N.m + 1700 \times 0.025}{0.2} \times 0.025$$

(only Fy & M2 is considered)

$$= 9.68\ N.m$$

$$\Rightarrow \text{Steer Torque in} = \frac{0.25\ N.m + 350 \times 0.025}{0.2} \times 0.025$$

$$= 1.125 N.m$$

$$\text{Total} = \text{Steer Torque out} + \text{Steer Torque in}$$
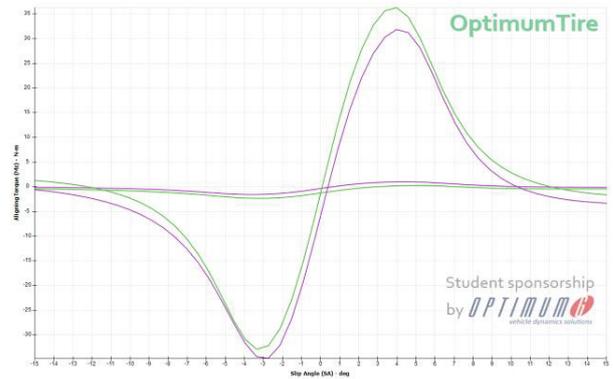
$$= 9.68 + 1.125$$

$$= 10.805\ Nm$$

Figure 2. Self-aligning torque vs slip angle

Self aligning torque is a property of the tyre and can be found out using tyre data by creating a graph between self aligning torque and slip angle from the graph it can be seen that peak sat is 35Nm.

Therefore, a caster of degrees is kept and this will help determine the side view outboard points which will be further optimized in a kinematic software to give the final set of points. The side view geometry is as follows:
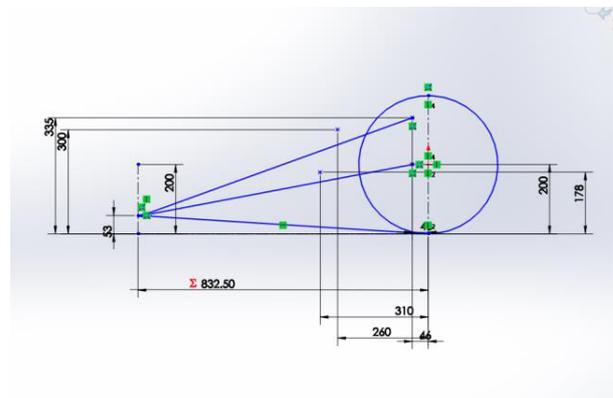

Figure 3. Side View Suspension Geometry

3.3 Inboard suspension points

These are the factors that need to be considered to decide the inboard pick up points:
- Anti-Dive
- Anti-Squat
- Roll Centre Migration
- Chassis Design
- Roll centre static height

The goal is to keep chassis as narrow as possible to make it light weight. Once the chassis width is decided according to rules, it gives a ballpark to place the inner points. The inner points can be placed till they are further optimized by kinematics software.

3.4 Anti-Dive

Anti-features are to be added to the suspension to stop pitching of the vehicle. More anti-percentage simply means that more load is going to the A-arms rather than through the springs. This prevents vehicle pitching. Setting Anti features is a matter of experience and is generally kept between 0-20 percent. As the inboard pickup points affect the Side View Geometry the percent anti-dive also gives a fair idea about the placement of the inboard pickup points.
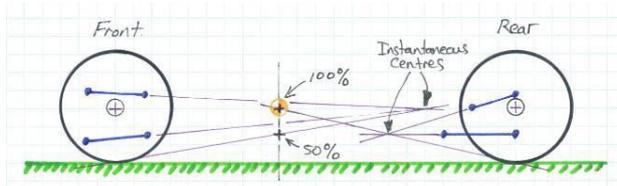

Figure 4. Anti-dive Geometry

3.5 Anti-Squat

The same logic goes for Anti Squat as Anti dive and some amount of it can be given to avoid vehicle pitching during acceleration.

The anti- squat percent affects the inboard points in the same way as anti-squat because it helps determine the side view geometry.

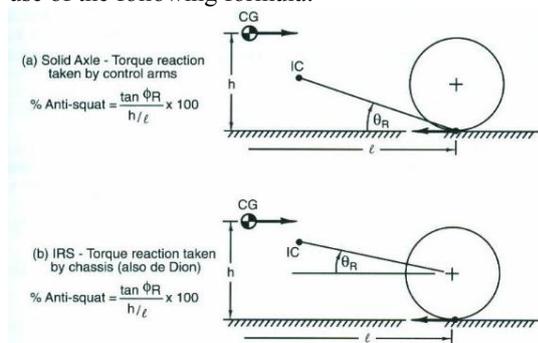The percent anti-squat can be calculated by making use of the following formula:

Figure 5. Anti-squat Geometry

3.6 Roll centre migration

Roll Centre migration also affects the inboard pickup points. The roll centre migration follows a trend where in the migration reduces drastically if the lower wishbone length is increased compared to the upper wishbone length. Thus, the inboard wishbone points can be kept under the chassis if space is available to make the lower wishbone longer while keeping the upper wishbone pick up points at their desired position relative to chassis design.

The graph showing this particular trend is shown later on.

3.7 Roll centre static height

As the front view IC is determined by the inboard pickup points, by changing the position of the inboard pick up points along the Z axis, the static roll centre height can be fixed. The static roll centre height at the front is generally kept greater than the rear to allow the rear of the car to catch up during cornering with the front. However too high a roll centre will cause jacking which is unwanted.
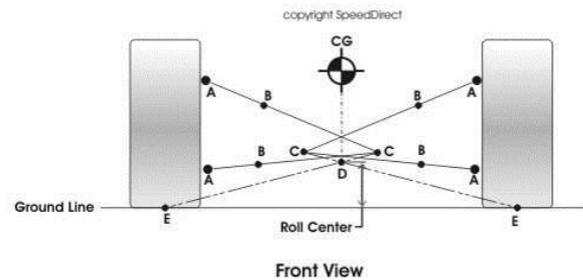

Figure 6. Static Roll centre

3.8 Camber variation in steering

The camber variation in steering is dependent on 3 variables:
- Steering Angle
- Kingpin angle
- Caster

The camber variation is directly proportional to the caster and increases with an increase in caster. Keeping Caster constant an increase in kPI will decrease camber variation with a positive steering angle and vice versa. The graph shown in Appendix shows the general trend:
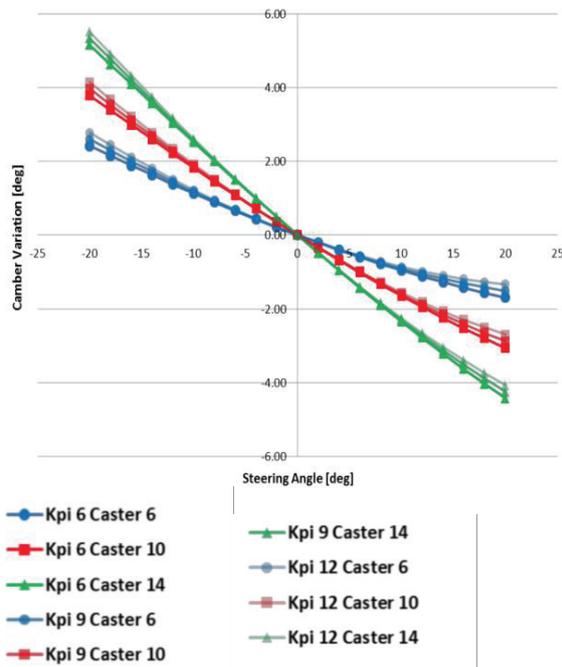
Figure 7. Camber variation vs Steering angle

3.9 Camber change in bump and roll

The camber change in bump and roll is an important factor to maintain maximum tire contact patch.

Camber change in Bump= $CB = \tan^{-1}\left(\frac{heave}{lat} * \frac{1}{VASL}\right)$

Camber change in Roll= $CR = \boldsymbol{\theta} * 1 - \left(\frac{track}{2*lat\ VASL}\right)$

Where:
- CB is camber change in Bump
- Heave is movement in z direction
- Lat VSAL is lateral Virtual Swing Arm Length
- CR is Camber change in roll
- θ is roll angle
- Track is trackwidth

Camber change in bump can also be analysed in kinematics software with varying VSAL for maximum calculated roll.

The total roll is calculated as 2.7∘.

3.10 Camber compliance

Every system has some kind of compliance and the compliance must be accounted for when going for any design. The camber compliance plays a crucial part in deciding static camber

The camber compliance for a 13-inch wheel is generally 1.7.

The following formula can be used to calculate actual camber taking compliance into account.

Camber=Static Camber + camber change-compliances

3.11 Roll centre and steering linkages

As opposed to popular belief the roll centre height static is affected by steering linkages. The effect is demonstrated by the graph.
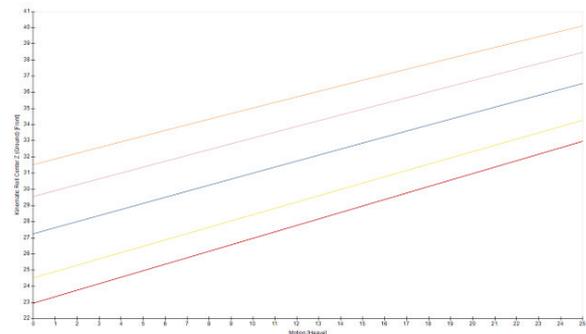

Figure 8. Roll centre height vs change in steering linkage height

as shown by the trends the roll centre height increases with decreases in z steering co ordinates and vice versa. The same trends as above can also be seen for changing the Y axis coordinates of the steering linkages.

3.12 Caster variation

Another important parameter to be discussed is caster variation. This depends on the side view swing arm length. With a long SVSAL with pitching there is a high caster change whereas the caster change is small in roll. With a short SVSAL the opposite applies.

3.13 Optimization in Kinematics software

The process to optimize in a kinematics software is tedious and involves a lot of trial and error. However certain trends were established regarding certain parameters in the software. These trends are seen later. Parameters like roll centre migration, static roll centre height, bump steer, camber variation in bump and roll can be analysed and the results of simulation can help determine final pick up points subject to vehicle testing. The bump steer is discussed in a parallel paper.

3.14 Outboard pick up points

As discussed above the outboard pickup points affect the roll centre migration and static height as they determine the A –arm lengths along with the upright length.

3.15 Inboard pick up points

The inboard pick up points also affect the same parameters and the results of analysis in Kinematics Software can be seen below.
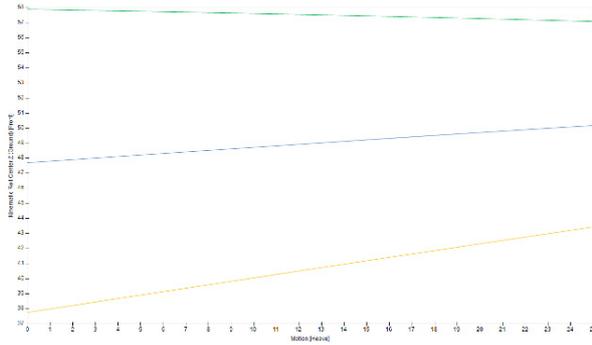
Figure 9. Roll centre migration vs inboard suspension pickup points

the graph shows that keeping the lower A arm constant and decreasing the upper A arm length the roll center migration goes down drastically
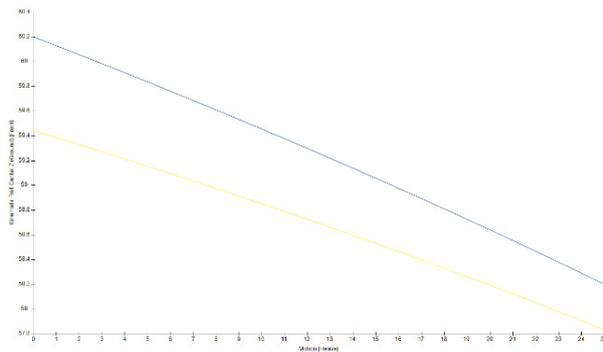

Figure 10. Roll centre migration vs inboard suspension pickup points

Similarly keeping the Upper A arm constant and increasing the lower A arm length also decreases RC migration.

3.16 Camber Variation in Roll

One of the main purpose of the suspension is to maintain maximum contact patch during all types of vehicle motion. To best use the kinematics software firstly the vehicle roll is calculated using and the roll motion is simulated. The aim is to maintain 0.4 degrees of camber at full roll to maximize contact patch. Thus, a static camber is chosen accordingly with trial and error to get 0.4 degrees of camber at full roll. Now the same static camber will be analysed in heave to see if it is close to the 0.4 degrees of camber needed. So, a static camber of -1.48 can be set.

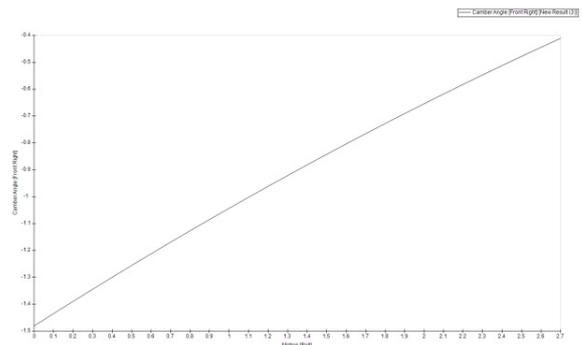The graph below shows the results of camber variation in roll.

Figure 11. Camber variation in roll

as seen from the graph the camber changenges to -.4 deg at max roll

3.17 Camber Variation in Bump

Similarly, the camber variation in bump can be seen by simulating vehicle pitch motion. The pitch motion can be specified as according to the pitch expected. We have taken the vehicle to pitch 25 mm up and down. For the following motion, the camber changes in bump from -1.48 to -0.35. As it is close to the camber required at full pitch for maximum contact, the above particular value of static camber can be finalized.
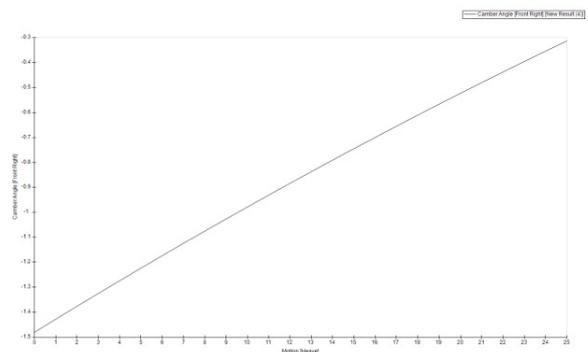

Figure 12. Camber variation in roll

as seen from the graph the camber changenges to -.35 deg at max bump

3.18 Forces on the system

The load on the suspension system can be calculated by taking moments of forces available at tire contact patch. The maximum normal load is calculated by taking into account weight transfers. The lateral maximum is known at optimum slip angle and longitudinal peak is known at optimum slip ratio by looking into the tire data.
The calculations are as follows:
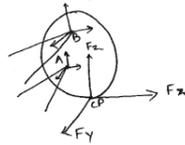
For F_z force, on lower & upper wishbone

$$\Rightarrow F_z = F_1 + F_2 \quad (F_z \text{ from Tire Data & Calculation})$$
$$1100 = F_1 + F_2$$

Taking Moments about A

$$F_z \cdot r_{cp} = F_2 \cdot r_B$$
$$\Rightarrow 1100 \times 25 = F_2 \times 20$$
$$\Rightarrow F_2 = 1375 \, N$$
$$F_1 = 275 N$$

For Fy Force

$$F_y = F_{y1} + F_{y2} \quad (F_y \text{ from tire Data})$$
$$\Rightarrow 1700 = F_{y1} + F_{y2}$$

Taking Moments About A

$$F_y \cdot r_{cp} = F_2 \cdot r_B$$
$$\Rightarrow 1700 \times 146 = F_2 \times 180$$
$$\Rightarrow F_2 = 1378 \cdot 8 \, N$$
$$\Rightarrow F_1 = 321 \cdot 2 \, N$$
$$F_x = F_{x1} + F_{x2}$$
$$\Rightarrow \text{Taking Moments about A}$$
$$F_x \cdot r_{cp} = F_{x2} \cdot r_B$$
$$\Rightarrow 1200 \cdot 146 = F_2 \times 180$$
$$\Rightarrow F_2 = 973 \cdot 33 \, N$$
$$\Rightarrow F_1 = 226 \cdot 7 \, N$$

Figure 13. Calculation of forces on suspension arms

3.19 Load Analysis on carbon fibre rods

One of the main goals of suspension design is to reduce the unsprung mass. So, a decision was taken to use carbon fibre rods for the A-Arms. The process to be used was to use carbon fibre rods and attach it to the rod ends with the use of Aluminium inserts which will be stuck to the carbon fibre tubes with the help of high grade epoxy. The design of the aluminium inserts is of utmost importance as weight needs to be minimum while still keeping strength intact.

3.20 Al inserts

As mentioned above Al inserts are designed in Solidworks and analysed. The results of analysis are shown below.

As the displacement is very low the above design can be considered good for the application. The weight being 70 grams also is very light and can be used as part of the suspension system.
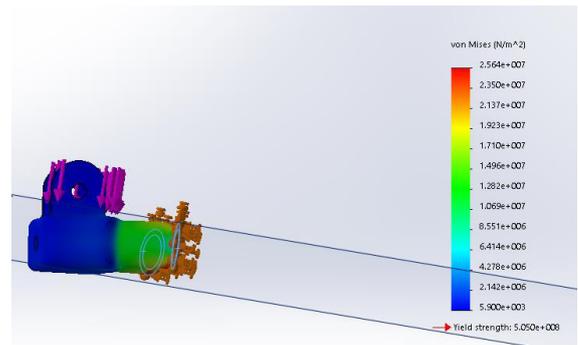

Figure 14. Stress on insert
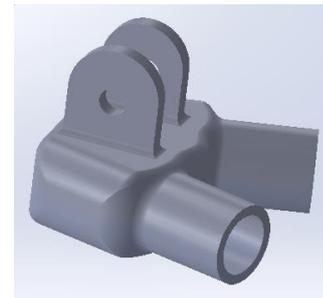
Final Design in Solidworks

Figure 15. Final Design

References

1. www.iaeng.org/publication/WCE2016/WCE2016_pp1062-1066.pdf
2. Ganesh Mohan, Claude Rouelle, Emmanuel Hugon, OptimumG LLC, Denver, Colorado *A New Method to Evaluate Bump Steer and Steering Influence on Kinematic Roll and Pitch Axes for All Independent Suspension Types*
3. http://www.optimumg.com/docs/SteeringGeometry.pdf
4. William F. Milliken and Douglas L. Milliken, 1995, "Race Car Vehicle Dynamics", Society of Automotive Engineers, Inc.
5. Gaffney, Edmund F., and Anthony R. Salinas. Introduction to Formula SAE® Suspension and Frame Design. No. 971584. SAE Technical Paper, 1997.
6. Robertson, Dennis, and George J. Delagrammatikas. The suspension system of the 2009 cooper union FSAE vehicle: A comprehensive design review. No. 2010-01-0311. SAE Technical Paper, 2010.
7. Gillespie, Thomas D., Fundamentals of Vehicle Dynamics. Society of Automotive Engineers, Inc., Pennsylvania, 1992.
8. Smith, Carroll, Engineer to Win. MBI Publishing Company, Minnesota, 1984.