Kinematic and Structural Analysis of Independent Type Suspension System for Fsae-Vehicle Equipped with Front anti-Roll Bar Setup

Yogesh Kumar, Rehan Abad Siddiqui, Yogesh Upadhyay and Qazi Mohd Taha

June 3, 2021

KINEMATIC AND STRUCTURAL ANALYSIS OF INDEPENDENT TYPE SUSPENSION SYSTEM FOR FSAE-VEHICLE EQUIPPED WITH FRONT ANTI-ROLL BAR SETUP

Yogesh Kumar[1,2] , Rehan Abad Siddiqui[1,2] , Yogesh Upadhyay[1] , Qazi Mohd Taha[1,2]

[1]Department of Mechanical Engineering, Zakir Husain College of Engineering and Technology, Aligarh Muslim University, India

[2]ZHCET Formula Racing, SAE-ZHCET, Aligarh Muslim University, India

ABSTRACT

This study involves the development of a Suspension system for FSAE vehicles participating in the Combustion Category. The front setup is also introduced with a bar-type Anti-Roll Bar (ARB) to manipulate the Understeer and Oversteer response of the vehicle. The initial parametric study was performed with an iterative approach in Excel. Stiff ride frequency above 2 Hz are suggested for such racing vehicles; the reason being that the driver does strict manoeuvring on the race track. FSAE vehicles have ARB's for better cornering performance and to manipulate the response of the vehicle mainly understeer and oversteer suiting to the driver. Another advantage is the limitation of camber gain caused by the body roll as it improves the traction.

Further, the Kinematic study of all the components in the system was analyzed through IPG Kinematic Software. The focus was made on the variation Camber Angle, Steer Angle, Track Change, and Roll Angle with and without the ARB incorporation. The calculation for the selection of different bearing is also performed. Considering the previous vehicles and the new design goals, parametrizing of the vehicle is also performed. An overhang of the vehicle plays an important factor in the longitudinal load transfer during the braking and acceleration. Forces experienced by all the components were also extracted from the software. This data is then used as the input parameter for Structural Simulation on CAE platform ANSYS. Material properties of Aluminium 6061 T-6 are used for the Bell Cranks and carbon Steel is used for ARB setup. Static structural simulation is performed on the Front and Rear Bell-Cranks. Linear Buckling structural simulation is also performed on the Push/Pull rods. Specific torsional simulation is done on the ARB considering the axial-offset position of support bearing and the loading point on the blade.

Keywords: FSAE, Suspension Kinematics, Anti-Roll Bar, Camber Angle, Automobile, Structural Simulation.

1. INTRODUCTION

Suspension System is very critical for any automobile, as it defines the behavior of the vehicle towards all the forces and road irregularities. For a commercial vehicle, the suspension is designed accordance with the application and level of comfort desired by the recipients. This setup is primarily responsible for maintaining the proper contact with the ground and the adequate ground clearance to tackle any possible damage to the machine.

Formula-Student Competitions are annually held around the global to boost the young engineering student from every domain for implementing their theoretical knowledge into practice.[1] FSAE is a global community of student who design, build and race their own creation while fulfilling the technical requirement set by organizing committee. Usually, it is considered that Suspension designing is one of the crucial and early steps towards the formula styled racecar formulation. Mostly teams are equipped with “independent type suspension system” which are positioned outboard for the ease of adjustability, maintenance and accessibility [2].

Unequal length Double Wishbone Suspension geometry have different parts; A-arms, Actuator Rod (Push/Pull), Spring & Damper system, Additional Ant-Roll Bars (or Sway Bars), etc.[3] Anti-roll Bar (or Stabilizer/Sway Bar) can be used in the front or rear setup reduce the overall rolling of the vehicle during the excessively sharp turns in high speed conditions[4][5][6]. Push rods and Anti-Roll Bars (ARB) are also considered as tuning components in the racing community, as they can easily alter the behavior of the vehicle by varying the actuation length[7], [8]. A significant amount of work have been done over the determination of initial parameters for the suspension setup and the Finite Element Investigation of the different components[2], [9]–[13]. The suspension system is actuated either due to the minor irregularities of the road or with any obstruction. The forces generated due to the bump in the wheel travel are absorbed by the system to establish the nearby-balanced equilibrium for the vehicle [14][15].

Figure 1 ZHCET FORMULA RACING CPMBUSTION VEHCILE AT SILVERTONE, UK

The suspension setup of any vehicle is responsible to connect the body with wheels, and behavior such as pitching, rolling and yawing can be controlled by the kinematics of the suspension [12], [16]. Getting the desired output from the suspension system is a complex process and need a comparative and iterative study of different parameters to analyze the optimum parameters [2], [11]. The static determination of the various parameters like Ride Rate, Roll Rate, Ride Frequency and Roll Gradient has been performed by many researched for both FSAE and Commercial vehicles using the iterative approach [2], [11], [14], [17]. The initial kinematic analysis for double wishbone suspension is usually carries out considering the system as two-dimensional four-bar mechanisms. Authors have further used tools like ADAMS [12][18], Lotus Shark [13][19], Solidworks Motion Study[20][2], and Mathematical models[16] for optimizing the parameter for the three-dimensional setup. Graphical representation of the calibrated parameters have been showcased in the literature for their respective vehicle specification [12][16][19][20].

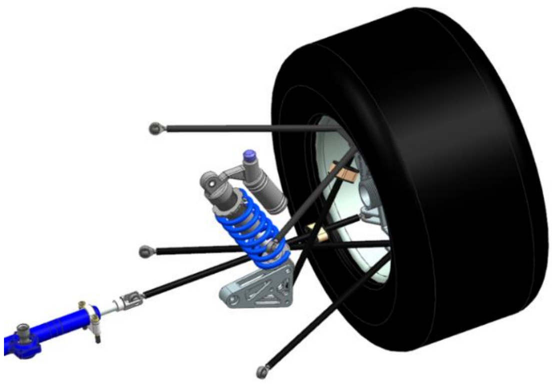


Figure 2 FRONT SUSPENSION GEOMETRY OF FSAE VEHICLE

Anti-roll bars are incorporated in the commercial vehicle to reduce the body roll especially in the heavy weight category such passenger bus, trucks, and LMV's. However, the implementation of ARB in sports category are more related to the better cornering performance, controlling the vehicle response; understeer or oversteer, and roll stiffness [21], [22]. Sway bars also accounts for limiting the camber gain caused by body roll [23]. After proper optimization and studying variation on the position of bushing in the ARB system, author have found an improvement of 31% in the handling of the vehicle [8]. Another author [23], have concluded that we can reduce the body roll of the Passenger Bus by factor of 0.8 with the implementation of the ARB system. In addition, the integration in the front geometry will provide more stable response. For the FSAE vehicles, the implementation of the Anti-roll bar is somewhat different as with the requirement of the overall vehicle and operating driver. Adjustability and accessibility are two main factor that are considered during the designing of such system.

In the literature [9], the author have successfully validated the effect on Anti-roll bar setup for the cornering situations. The setup have not only reduced the magnitude of body roll but also stabilized the vehicle to tackle with the banking and irregularities on the track. This research study aims at the kinematic investigation of the suspension system developed for the 390cc KTM engine with first-generation adjustable ARB. The force analysis on the different components is performed to evaluate the input parameters for the FEM. Only front suspension setup is considered in this study and the effect of front ARB on the vehicle is also analysed.

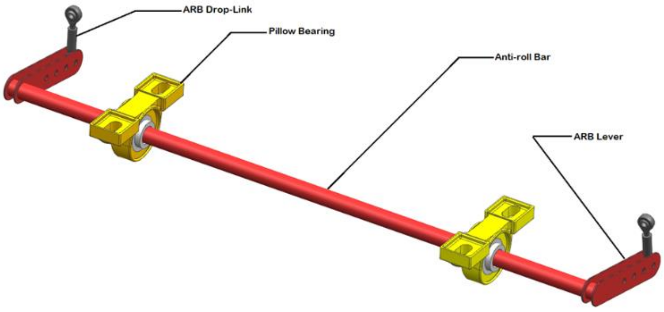


Figure 3 ANTI-ROLL BAR SETUP FOR FRONT SUSPENSION OF FSAE VEHICLE

2. METHODOLOGY

In this section, the procedure required to develop a suspension system is presented. Before, proceeding into the modelling of the system, the rules and restrictions were taken into the account[2][11]. In addition, the expectations from the system were discussed. The initial parametrization of the vehicle is perform consulting with the requirement of the space needed by the other sub-system of the vehicle. A study is conducted on the variation of longitudinal and lateral load transfer caused due to variation in the magnitude of the Wheelbase and Track-width. Other calculation containing parameters like Ride Rate, Roll Rate, Spring Rate, etc. also performed[17][11][3]. To move forward with the CAD modelling and the kinematic study, an initial preliminary design is generated symbolizing the 2D Free Body Diagram of the over-all system.

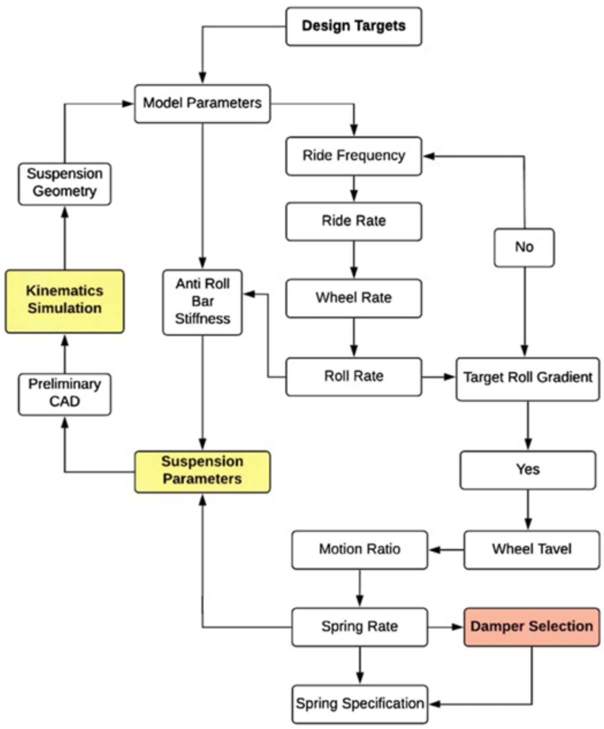


Figure 4 FLOW CHART OF DESIGN METHODOLOGY

After finalizing the suspension geometry, the data-points were extracted to be use in Kinematic Simulation using IPG. Once the data-points are finalized, the final 3D model of the suspension system is developed and processed for the structural integrity through FEM modelling using ANSYS.

2.1 REQUIREMENTS AND RESTRICTIONS

Before every session of FSAE competitions, respective organizing committees like IMechE, Formula Student Germany, SUPRA-India, Curiosum Tech, etc. releases their rulebook.

Following are the rules related to Suspension System[24][25]:

a) The vehicle should have a minimum of total operating wheel travel of 50 mm with driver seated. Minimum jounce of 25 mm should be there.
b) Vehicle should have minimum wheelbase of 1525 mm.
c) One of the track-width should not be less than 0.75 times the other.
d) A minimum of 30 mm of ground clearance is mandatory everywhere in the vehicle excluding tires with driver.
e) All the mounting should be visible during the technical inspection round, directly or by removing the covering.

Following are the design requirement and expectation:

a) Avoid generation of shearing forces in any of the components; special focused made over the phenomenon known as Rod-ends in Bending (REIB).
b) Minimum change in the parameters like Camber, Toe, etc. in dynamic conditions.
c) Low Center of Gravity and Better Rollover Stability.
d) Stiff Roll Gradient and Ride Frequency.
e) Adjustability in vehicle steering response.

Consideration of the OEM components like Dampers, Bearing, and Fasteners made during the designing process. Factors like accessibility, easy maintenance, in-house manufacturing, and cost are also considered.

2.2 INITIAL PARAMETRIZATION OF VEHICLE

Wheelbase and Track-width are standardized way to define the overall size of the vehicle. Track-width can vary in the front and rear section of the vehicle. Load transfer and concerning stability of the vehicle is highly influenced by these parameters.

Wheelbase: According to the FSAE rulebook, the shortest possible wheelbase of the vehicle can be 1525 mm. Minimizing the wheelbase helps in reducing the turning radius of the vehicle, helpful in tackling the hairpins on the track. A key factor is the overall length of the chassis to accommodate the drivetrain, engine, driver, brakes and other components. For efficient transfer of power from the engine to wheels, rear wheels and drivetrain system are aligned about the same axis. Another thing to consider is the overhang of the front section. Increased overhang will affect the turning of the vehicle, as it contribute towards the sweeping radius of the vehicle. On the other hand, overhang portion also provide room for driver or we will have increased wheelbase length.

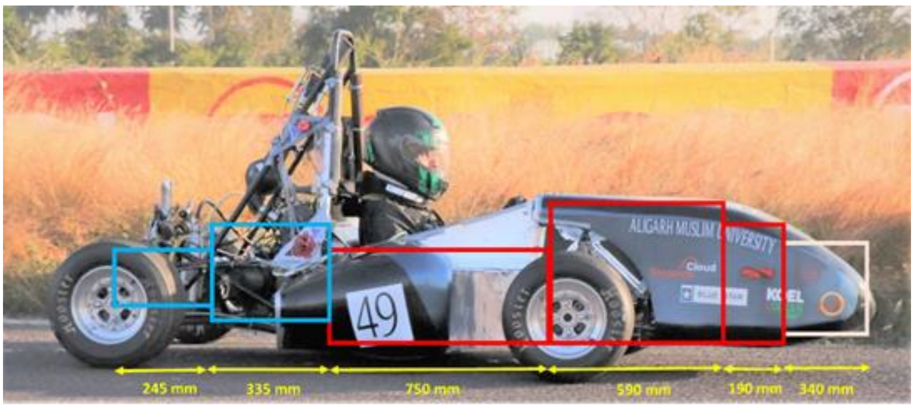


Figure 5 ESTIMATED SPACE REQUIRED BY SUB-SYSTEMS

We tried to get optimum wheelbase considering the space required by the different components and the overhang percentage of the vehicle in front. Below mentioned Table (1) shows, the estimated lengths required by different departments within the vehicle, this will give us the location of the front wheel center results into the required wheelbase of the vehicle.

Departments	Length (mm)
Drivetrain	245
Engine	335
Cockpit	750
Driver's Legs	590
Brakes	190
Crush Structure	340
Total	**2450**

Table 1 ESTIMATED SPACE REQUIRED BY THE DIFFERENT SUB-SYSTEM

From Table 1, rough estimation of the space required by different sub-system is utilized for the evaluating the wheelbase with proportion of overhang. Location of center of gravity and the wheelbase have significant effect on the axle load distribution and the resultant wheel loads. During acceleration and braking, the shorter wheelbase will provide a benefit in greater load transfer from front to rear axle or vice-versa [26].

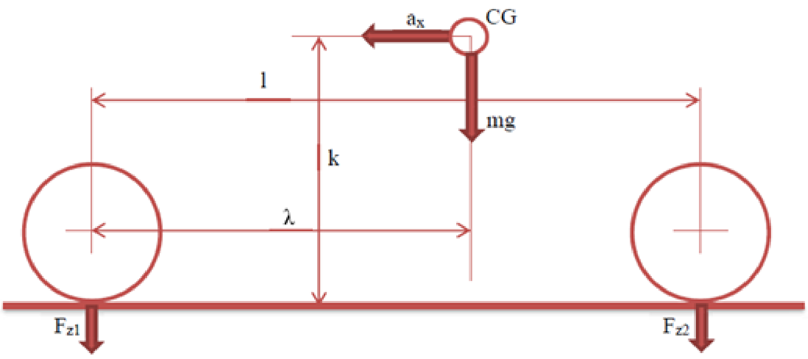


Figure 6 PARAMETERS FOR LONGITUDICAL LOAD TRANSFER

Axial Load Distribution:

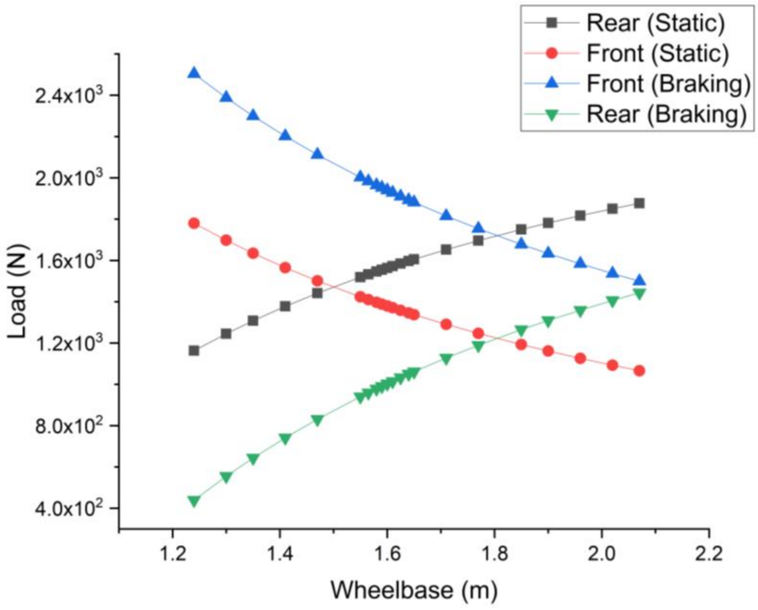
$$F_f = \frac{(l-\lambda)}{l} \times mg$$

$$F_r = \frac{\lambda}{l} \times mg$$

Longitudinal Load Transfer under Braking (1.5G):

$$F_{z1} = F_r + \frac{1.5g \times mg \times k}{l}$$

$$F_{z2} = F_r - \frac{1.5g \times mg \times k}{l}$$



Graph 1 WHEELBASE COMPUTATION FOR DIFFERENT CONDITIONS

The above wheelbase computation suggests that the larger value of wheelbase will reduce the longitudinal load transfer during braking. This will reduce the aggressive pitching of the vehicle. However, the larger wheelbase also increases the turning radius. After observing the static load distribution with the wheelbase and the steering performance consideration, we selected the wheelbase as 1.55 m or 1550 mm because of it nearly ideal 50:50 weight distribution, with approximately 45% in the front and 55% in the rear. The longitudinal load transfer is also acceptable in this range.

Track-width: For cornering performance of the vehicle, front and rear track-width plays a vital role. This directly contribute in the lateral load transfer and the rollover stability of the vehicle. Stabilizers maybe installed to reduce the lateral load transfer without affecting the parallel kinematics of the system. For FSAE vehicles, it is believed that the track-width more than 1.2 m will not leave enough space either side to maneuver on the track, considering the 3 m minimum width of the track according to the rules.

Total Lateral Load Transfer:

$$\Delta W = \frac{mg \times h \times Ay}{t}$$

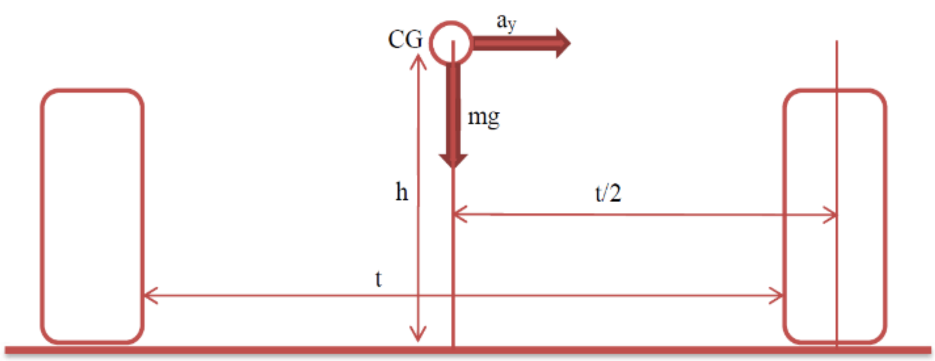


Figure 7 PARAMETERS FOR LATERAL LOAD TRANSFER

After careful consideration, a shorter track-width in the rear is selected with accumulating the 75% rule in the front. However, a square profile of the vehicle is slightly better option for the overall efficient performance.

2.3 CALCULATIONS

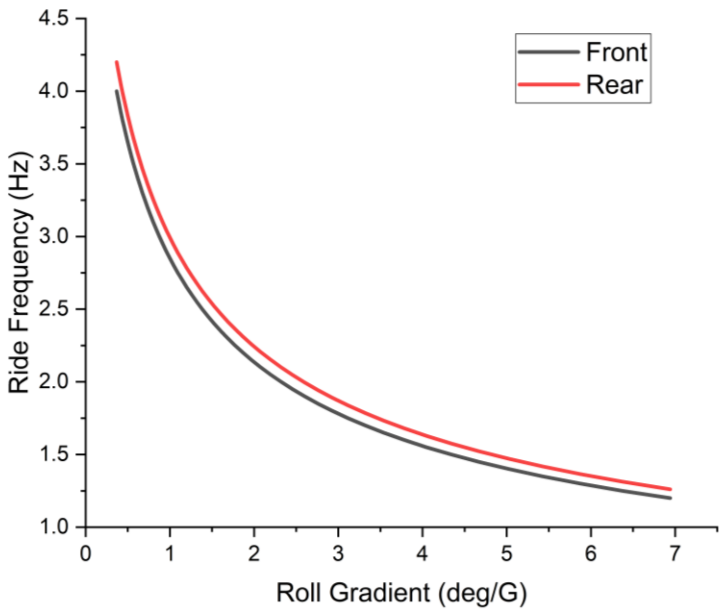
2.3.1 Suspension Parameters:

Since, the suspension parameters has to be calculated and to get the best results out of them using an iterative approach. For Ride and Roll rates, an Excel spreadsheet created with the iteration rate of 0.05. A graph between Ride Frequency and Roll Gradient is plotted to get the desired values.

Mass of the car	300		Kg
Weight at front	135	45%	Kg
Weight at rear	165	55%	Kg
Track-width (Front)	1220		mm
Track-width (Rear)	1194		mm
Wheelbase	1550		mm

Table 2 INITIAL CALCULATED PARAMETERS

Considering the application of the system, FSAE track requires stiffer roll gradient. As suggested by many authors, a range of 1.2-1.8 deg/g might be an adequate value. For the accounted tracks in the dynamics events, driver does strict maneuvering on the track. So, stiff ride frequency above 2 Hz are suggested for such racing vehicles [2], [3], [17]. Reducing the ride behavior to the softer side will cause lag for the input response. Thus, this is affect the efficiency of the driver for handling and stability of the vehicle. Being a rear-wheel drive vehicle, the concentration of mass on the rear end is more as compared to front. This will create a pitching effect during braking or bumping scenarios. Compensation can be made by increasing the rear ride frequency by a small amount, 5% in our case. The quicker response on the rear wheels will help to limit the pitching effect and have more balanced vehicle response [14].



Graph 2 ROLL GRADIENT Vs RIDE FREQUENCY

In the literature [14][27][11][16], the formulation required to calculate the various parameters have been discussed in detail. Ride rate, Lateral Weight Transfer, Wheel Rate, Wheel Travel, Roll rate, and Roll Gradient have be calculated in the literature [14]. The author have defined all the parameters and the algorithm used to compute the variables by a flow chart. An approximation assumption towards the selection of the initial Ride frequency is considered by the authors [14][19]. However, author in the literature [11] proceeded with calculating the ride frequency and then moving forward with the other parameters.

These are the final parameters obtained by:

Parameters	Front	Rear	Units
Ride Frequency	2.4	2.52	Hz
Ride Rate	15333.65	20662.09	N/m
Lateral Weight Transfer	501.86	613.38	N
Wheel Travel	32.73	29.68	Mm
Wheel Rate	17599.8	25000.2	N/m
Roll Rate	228.18	310.76	N-m/deg
Roll Gradient	1.526		Deg/g
Motion Ratio	0.825	0.925	--
Spring Rate	25.85	29.21	N/mm
ARB Stiffness	2765	--	N-mm/deg

Table 3 PARAMETERS OF THE SUSPENSION SYSTEM

2.3.2 Spring-Damper System:

From Table 3, the maximum wheel travel (either jounce or rebound) available is 33 mm on the front and 30 mm on the rear. Motion ratio is directly proportional to the spring rate. Determination of motion ratio is based on the springs that are available in the market. Unity of motion ratio describes the equivalency between the spring travel and wheel travel, which also limits the design to Direct-Actuation System. Reducing the motion ratio below unity will result into stiff springs. Pull and Push rod actuation setup with motion ratio of 0.825 and 0.925 is selected for front and rear setup respectively. Springs of 26 and 30 N/mm were finalized as per availability.

Dampers plays a crucial role in ride handling of the vehicle. Damping coefficient in the range of 0.3-0.6 is suggested for the FSAE application. Critical damping lacks the comfort to driver from no oscillations, while low damping causes the lagged response to the provided input which is undesirable for the racing vehicles. DNM Burner RCP 2S dampers are selected due to low cost and the easy availability in India.

Parameters	Front	Rear	Units
Spring Constant (Ks)	26	29	N/mm
Material	Spring Steel Grade 2		
Young's Modulus	190		GPa
Density	7.8	7.8	g/cc
Poisson's Ratio	0.3	0.3	
Outer Diameter of Spring	52	52	mm
Diameter of Spring Wire	6	6	mm
No. of Active Coil	4	5	
Free length of Spring	140	140	mm

Table 4 SPECIFICATIONS OF THE HELICL SPRINGS

2.3.3 Anti-Roll Bar:

FSAE vehicle have ARB’s for the better cornering performance and to manipulate the response of the vehicle mainly understeer and oversteer suiting to the driver [9], [26].

The bar's torsional stiffness (resistance to twist) determines its ability to reduce body roll i.e. Roll Stiffness. The increased spring rate in the front due to ARB will produce understeer effect while in the rear, it will produce oversteer effect. Another advantage is the limitation of camber gain caused by the body roll as it improves the traction [5], [7], [8].

Equations mentioned below are used to determine the roll gradient of the ride springs and thus the deficit that the anti-roll bar needs to deal with [3], [9], [26].

Roll Gradient from Spring-Damper System:

$$\frac{\phi_r}{A_y} = \frac{W \times H}{K_{\phi F} + K_{\phi R}}$$

Front Roll Rate Due to Spring-Damper Setup:

$$K_{\phi F} = \frac{\pi \times (t_f^2) \times K_{LR} \times K_{RR}}{180 \times (K_{LR} + K_{RR})}$$

Rear Roll Rate Due to Spring-Damper Setup:

$$K_{\phi R} = \frac{\pi \times (t_r^2) \times K_{LR} \times K_{RR}}{180 \times (K_{LR} + K_{RR})}$$

Desired Total Roll Rate:

$$K_{\phi Des} = \frac{W \times H}{\frac{\phi}{A_y}}$$

Total ARB Roll Rate Needed:

$$K_{\phi A} = \frac{\pi}{180} \times \left(\frac{K_{\phi Des} \times K_T \times (\frac{t^2}{2})}{K_T \times (\frac{t^2}{2}) \times \frac{\pi}{180} - K_{\phi Des}} \right) - \frac{\pi \times K_w \times (\frac{t^2}{2})}{180}$$

Front Anti-Roll bar Stiffness:

$$K_{\phi FA} = K_{\phi A} \times N_{mag} \times \frac{MR_{FA}^2}{100}$$

$$K_{\phi FA} = 2.765\ Nmm/deg$$

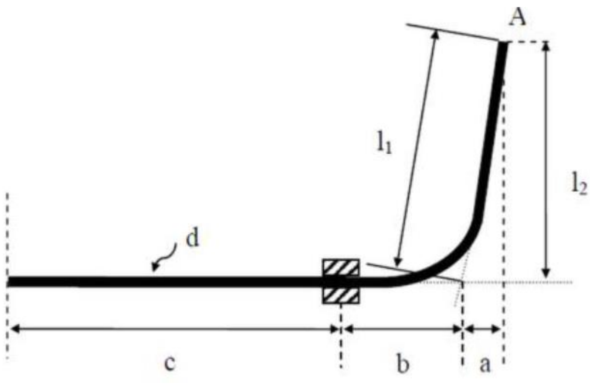


Figure 8 TYPICAL U-TYPE ARB GEOMETRY [28]

Now, we have the stiffness of the Anti-roll bar and this can be used to calculate the dimension of the bar using the concept of Designing of Torsion bars mentioned in "SAE Spring Design Manual"[28]. The nature of ARB also majorly depend on the placement of the bushing, as it the supporting point and the twisting of the bar is guided at this particular point. For the setup, OEM Plummer Block (UP004) of suitable load rating was selected with the bore diameter of 20 mm as of the bar and of suitable load rating.

Length of the Bar	718	mm
Length of the Lever	110	mm
Outer Diameter	20	mm
Inner Diameter	18	mm
Bearing Distance from Lever-end	80	mm
Max. Deflection	3.6	deg
Size of Bushing/Bearing	20	Bore Dia.
Fasteners	M6	M8.8
Length of Drop-Link	58	mm
Material of Bar	Mild Steel	

Table 5 SPECIFICATION OF ARB SETUP

2.4 KINEMATIC SIMULATION

After calculating the initial parameters and drafting, the preliminary three-dimensional free body diagram. The effect of crucial parameters influencing the vehicle behavior are analyzed using the kinematic software IPG Carmaker and Kinematics. Authors are unable to find any work based on utilization on IPG Kinematics for FSAE vehicle.

All the initial parameters, mass distribution and calculated spring rates needed to setup in the simulation control section of IPG Kinematics. Careful understanding towards the allotment of mass and bushing position is important for adequate results. Detail description of Input Data can be found in the literature [29]. Lastly, the calibration of the parameters are performed as per the required. IPG Graphs has been used to perform the comparative and iterative study of the various crucial parameters. These results are analyzed and then exported.

The objective of this study is limited to the variation in Camber Angle, Steer Angle, Track Change, and Roll Angle with and without the ARB incorporation. Reciprocal Kinematics are analyzed assumed that maximum variation can be obtained during the high-speed cornering situation. Therefore, reciprocal kinematics is preferred over the parallel kinematics, which employs the symmetrical inputs on both side of the wheels.

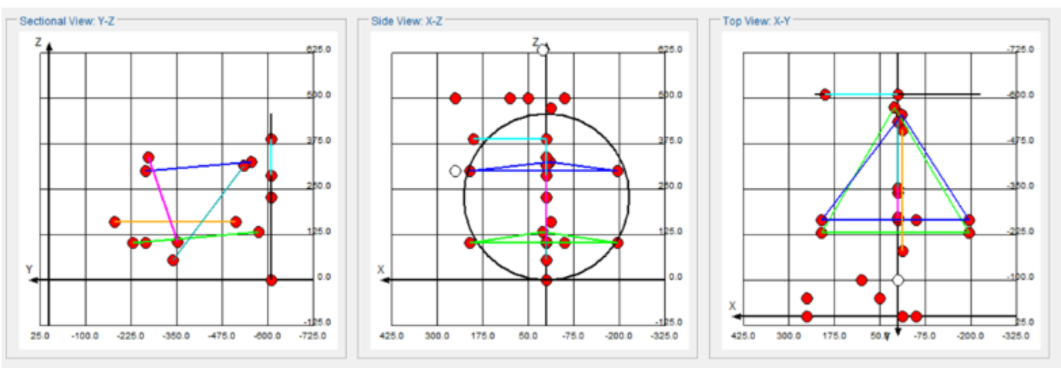


Figure 9 GEOMETRIC CONTROL OF FRONT SUSPENSION

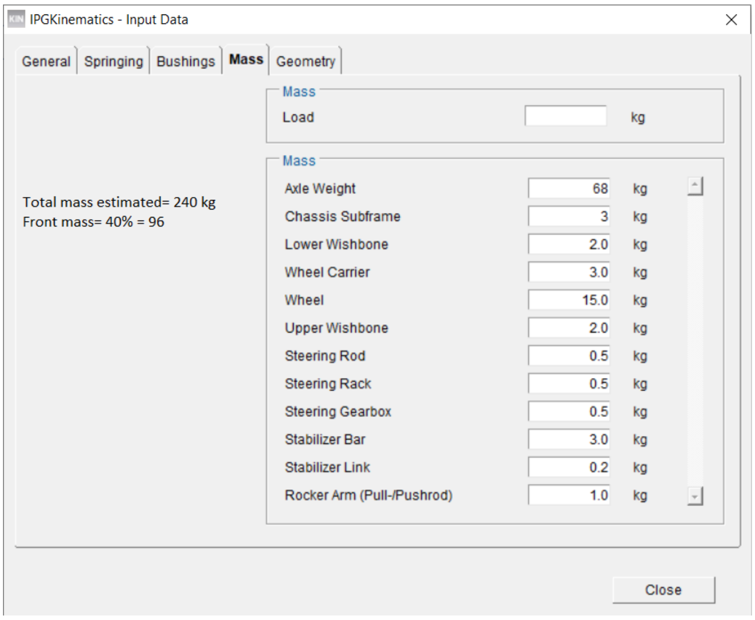
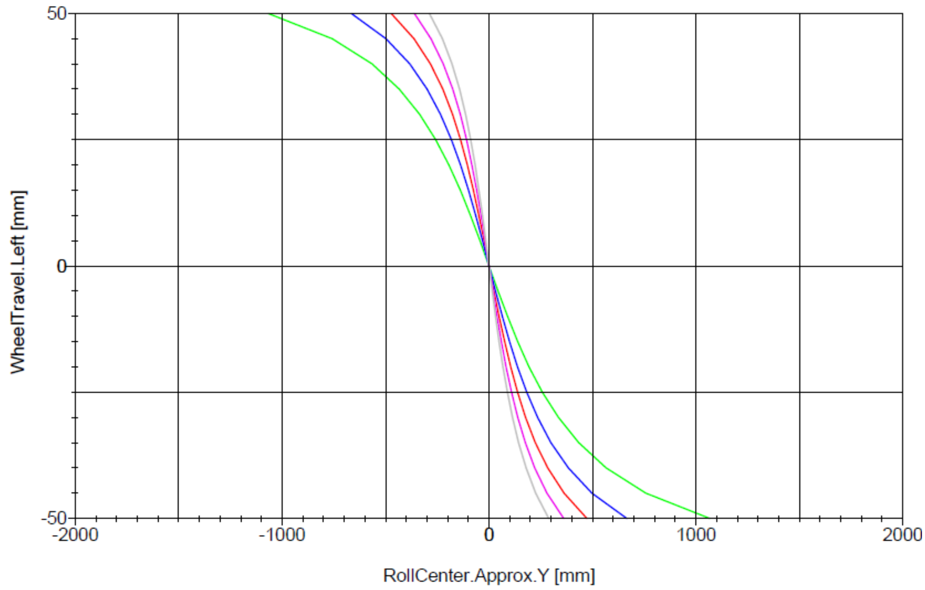


Figure 10 INPUT DATA PRESET IN IPG KINEMATICS

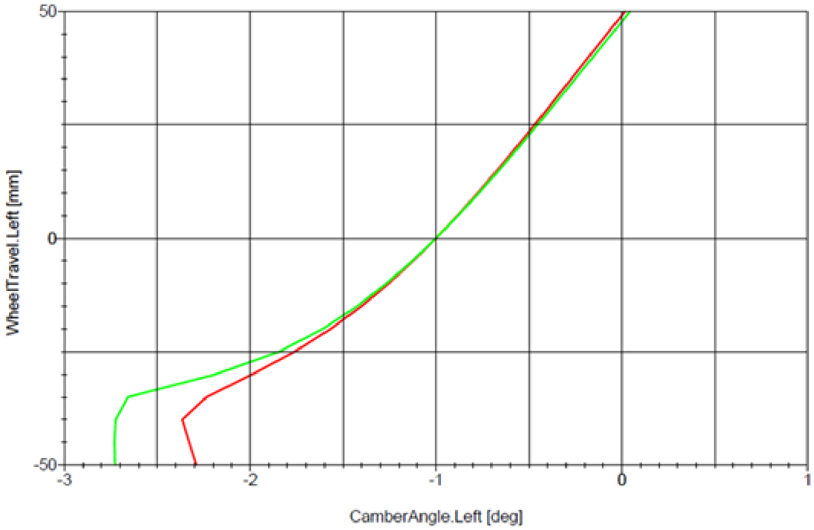
The vehicle is modelled with a wheel size of 18 inches having the lateral position of 609.6 mm from the rolling axis. The point on the contact patch is also defined with the same offset consideration. Previously developed preliminary FBD are incorporated with baseline Caster Angle for the position of the wishbones on the wheel carrier. For defining the bushing position in the preset, the official manual of IPG Carmaker has been followed [29]. Vehicle is also loaded with an axle load of 68 kg to represent the driver mass. Also, the proper distribution of front weight for the various section is calculated for the preset.



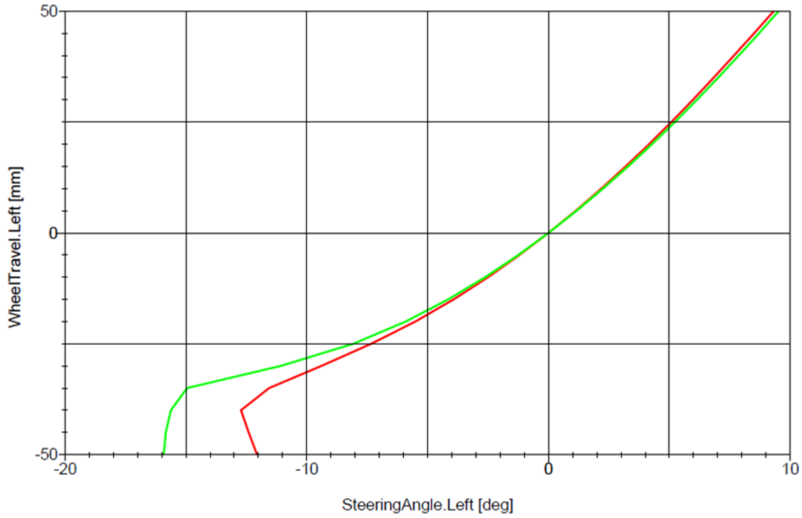
Graph 3 VARIATION OF ROLL CENTER HEIGHT IN LATERAL DIRECTION

Similarly, for the Anti-Roll Bar, the kinematic simulation performed on IPG Kinematics in Reciprocal mode. The behavior of the front suspension system with and without ARB is simulated.

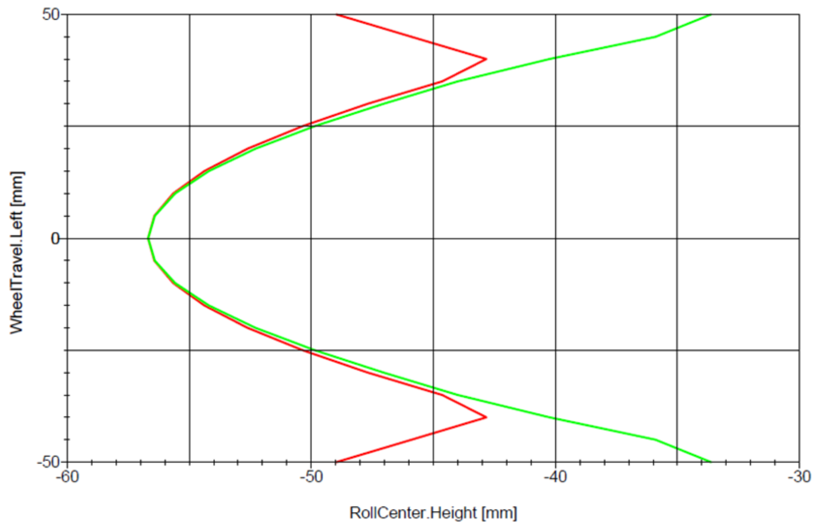
Below are the Graphs showing the change in Camber Angle, Steering Angle (suggesting the Understeer/Oversteer effect), Roll-Centre Height and the Roll Angle with corresponding to the 50 mm jounce and 50 mm rebound movement of the Wheel.



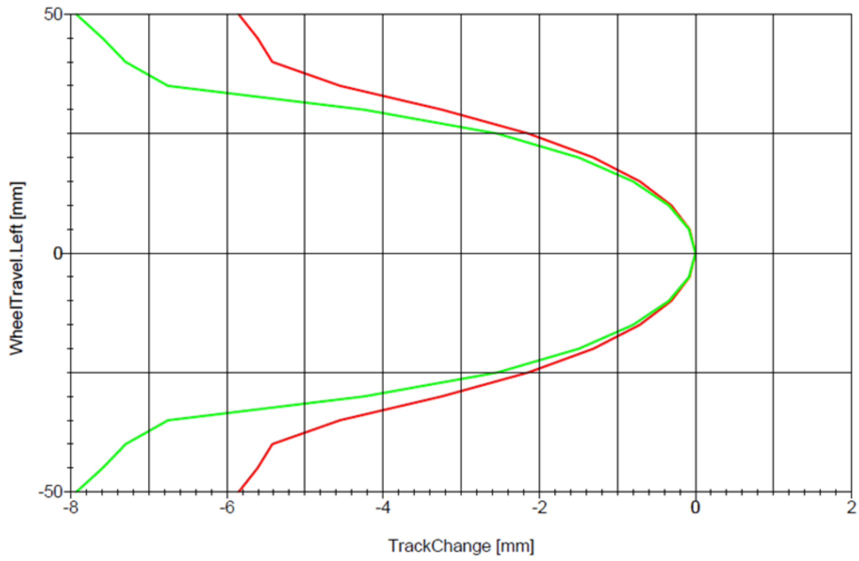
Graph 4 RECIPROCAL KINEMATICS: CAMBER ANGLE



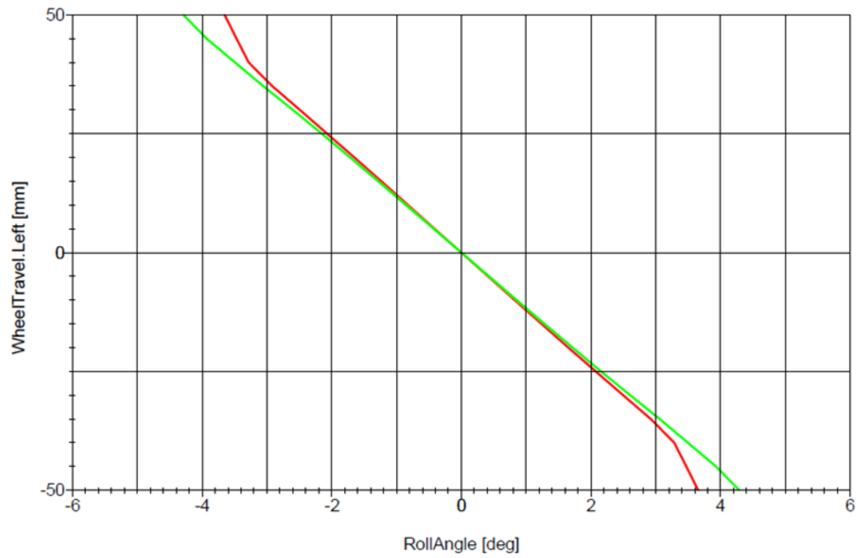
Graph 5 RECIPROCAL KINEMATICS: STEERING ANGLE



Graph 6 RECIPROCAL KINEMATICS: ROLL CENTER



Graph 7 RECIPROCAL KINEMATICS: TRACK CHANGE



Graph 8 RECIPROCAL KINEMATICS: ROLL ANGLE

ARB are designed to minimize the roll of the body, which we can clearly see in the graph of Wheel Travel Vs Roll Angle. Roll angle is now lying between ±3.8º. In addition, there is significant change in the Roll center height and the track-change as the ARB is now resisting the lateral movement/shift of the vehicle. As expected, the nature of the front suspension system has totally changed with the installation of Anti-Roll Bar. Red trend line indicates the setup with ARB and green for conventional double-wishbone setup. The Camber Angle is reduced from -3.5º to -2.5º and the reduction in Steering angle is displaying the increase of understeer effect.

This kinematic simulation also provides us the force experienced by each of the component in the system in respective direction. This data can be further use as an input for investigating the structural integrity of the components through FEM.

2.5 STRUCTURAL ANALYSIS

From the post-processing, the final hard-points are obtained for the front suspension geometry. These values are used to generate the 3D CAD models of the different components. Calculation towards the bearing load requirement was also perform to select the appropriate OEM parts. The loads for the bearing calculation as well as FEM was obtained from IPG graphs.

Since, we know the forces on the various points of the rockers one can easily calculate the loads on the pivot point, using the eccentric loading formulation [30]. The loads on the pivot point will later be used for the determining the dynamic load rating of the required bearing [31]. We have selected a deep groove ball bearing of 8x16x5, which has a dynamic load rating of 1600 N. For the Rocker (or Bell-Crank), Point A is being fixed, as it the point about which the rocker is pivoted. Point B and C are loaded with the force 1456.9 N and 300 N respectively as Fig. 10. These forces magnitudes are extracted from the IPG Kinematics in the three axis separately. Point C accounts for the forces applied by Anti-roll Bar/Stabilizer Bar, which will majorly be the force in –Z-axis whereas Points B shows the forces applied by the pull rod when the wheel is moved; the major component is the –Y-axis.

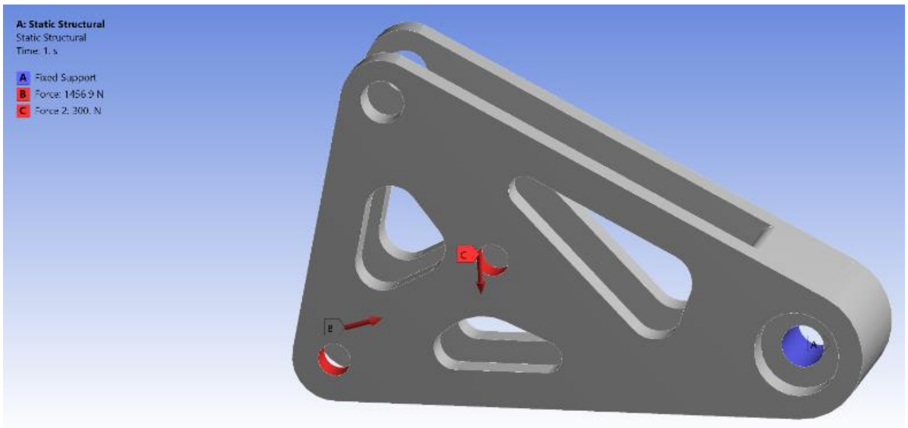


Figure 11 LOADING CONDITIONS FOR ROCKER

For the better results and considering the loading points of the geometry, the whole body was meshed ranging from 1 mm to 0.2 during the convergence process. While the separate face meshing is introduced for the loading points approximately half of the body sizing as shown in the above. Iterations were performed for removing the extra material from the geometry and reducing the thickness of the wall for min. weight optimized geometry keeping the ease of manufacturing in mind. Similarly, the other parts of the suspension setup were analyzed using the FEM. For the accomplishment of our design goals, one of the major factor was the selection of material. A comparative study towards the available materials is conducted to select the best-suited one out of that. The initial shortlisting of the composition to consider is solely based on the strength and other mechanical properties shown in Table 3.

Material	**Tensile Strength (MPa)**	**Brinell Hardness**	**Young's Modulus (GPa)**	**Fatigue Strength (MPa)**	**Strength to Weight Ratio (kNm^2/kg)**
Al 6061 T6	310	95	68.9	96.5	115
Al 7075 T6	572	150	71.7	159	196
Mild Steel	440	126	205	270	32

Table 6 MATERIAL PROPERTIES OF MATERIALS

Since, the system is being designed for students to participate in competition, several other factors like Affordability, Availability, and Ease of Machining comes into consideration. Table 4 discuss the point base system used to select the material for different components.

Material	**Mechanical Properties**	**Affordability**	**Availability**	**Machining**
Al 6061 T6	3.2	4.0	3.5	3.8
Al 7075 T6	4.8	3.8	3.4	3.7
Mild Steel	2.0	4.8	4.0	4.0

Table 7 DECISION MATRIX FOR MATERIAL SELECTION

Aluminum 7075 T6 was selected for Rocker, Upright and Hub. Anti-roll bar is made out of Mild Steel for in-house manufacturing and similar case with the Push rods. These above accumulator data was used in setting up the FEM model for different components. Mesh sensitivity study is performed on each component for better and reliable results. ANSYS software is used to perform all types of structural simulations, and here are the results of FEM performed on different components:

Component	Total Deformation (mm)	Equivalent Strain (mm/mm)	Equivalent Stress (MPa)	Factor of Safety
Rocker	0.1637	3.9E-4	79.976	3.126
Upright	0.1964	6.9E-4	138.76	1.80
Hub	0.0256	1.1E-3	138.77	1.62
Anti-Roll Bar	0.43	1.0E-3	155	1.5

Table 8 CONCLUSION FROM FEM INVESTIGATION

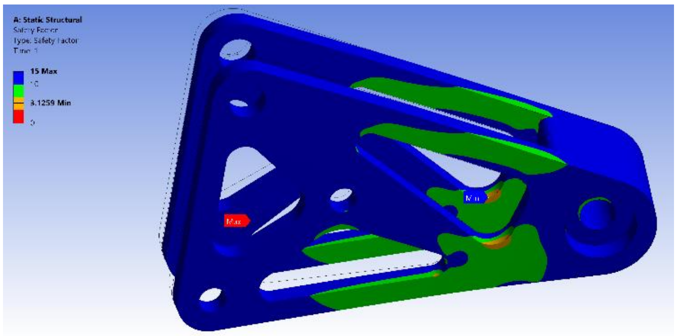


Figure 12 FEM SIMULATION OF ROCKER: FACTOR OF SAFETY

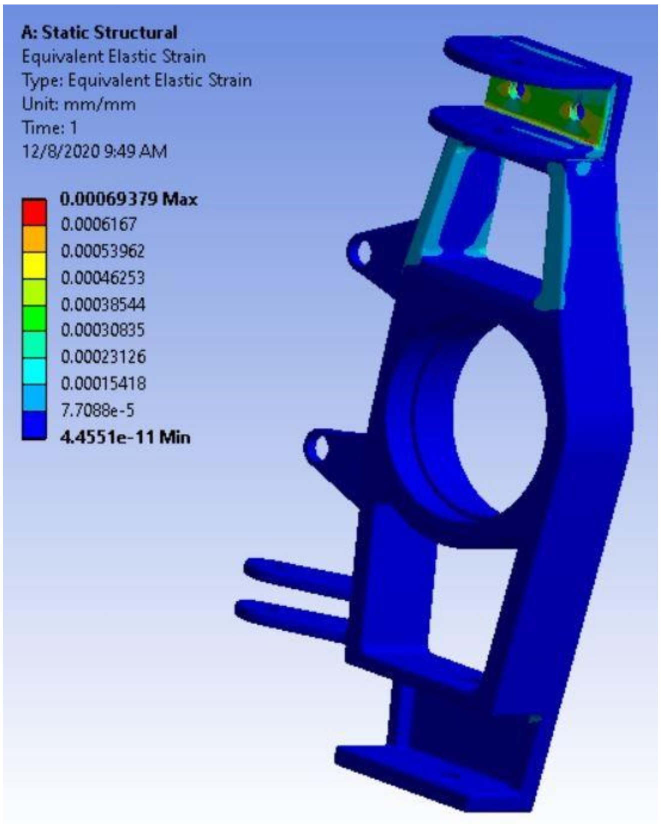


Figure 13 FEM SIMULATION OF UPRIGHT: EQUIVALENT ELASTIC STRAIN

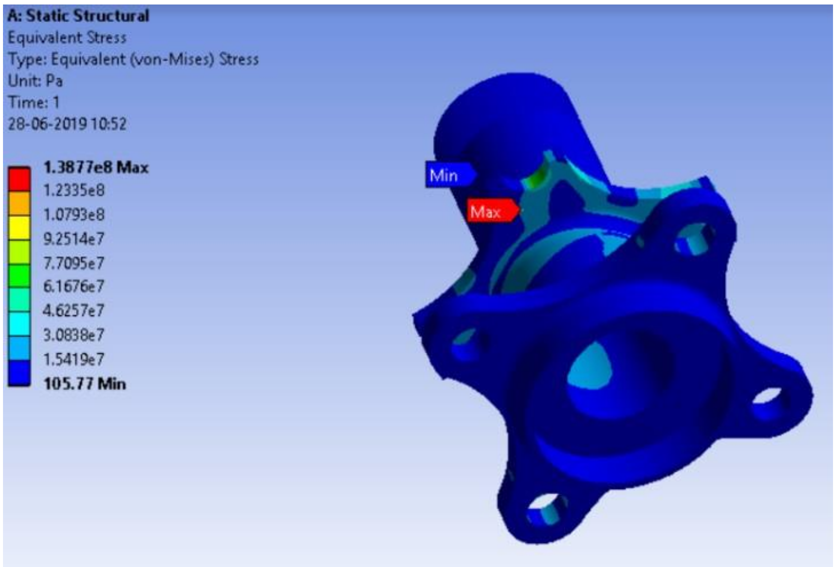


Figure 14 FEM SIMULATION OF HUB: EQUIVALENT (VON-MISES) STRESS

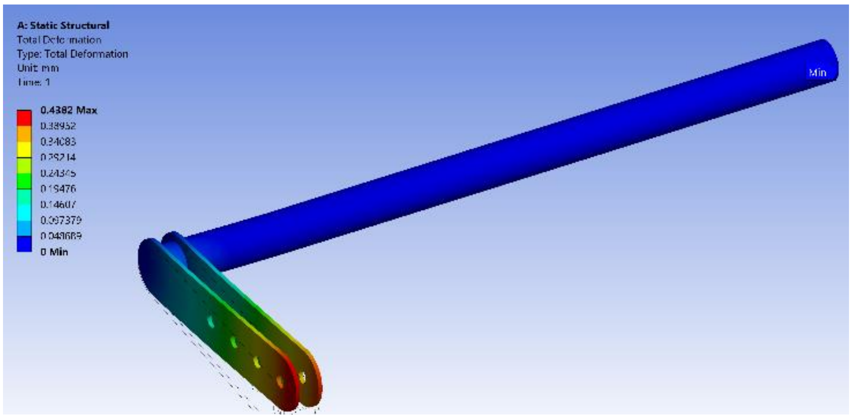


Figure 15 FEM OF ANTI-ROLL BAR: TOTAL DEFORMATION

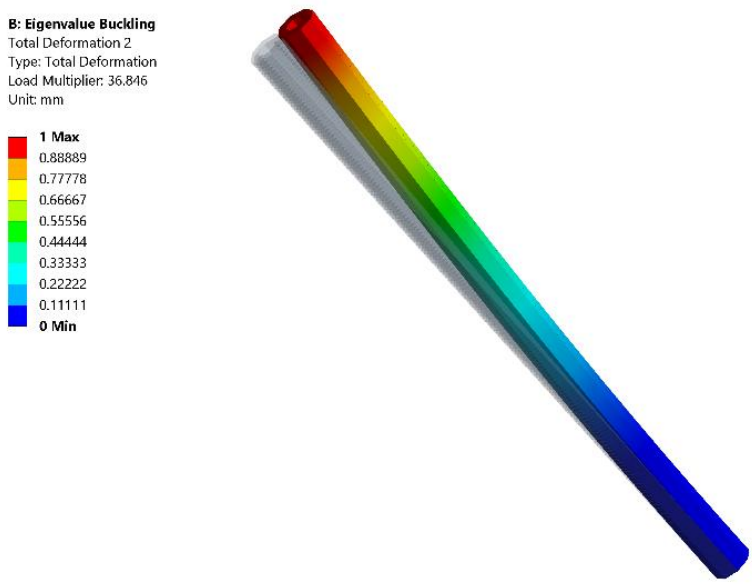


Figure 16 EIGENVALUE BUCKLING SIMULATION OF PUSH ROD

3. RESULTS AND DISCUSSION

Reciprocation Kinematics is performed to check the variation of the different critical angles contributing towards vehicle behavior. The study indicates that incorporating an anti-roll bar will reduce the Roll Center Height and Track Change variation. The curving back of Roll Center Height indicates towards the additional stiffness provided by the ARB setup. Bringing down the roll center at one end will reduce the roll moment of the vehicle, which is counted by the other end. This will develop a better spread-out loading during the cornering condition. The camber angle variation trend lines shows the reduction of 0.5° which highly influence the gripping nature of the vehicle. It is always preferred to have minimal variation in the Steering Angle and Camber Angle of the vehicle. The slightest variation will cause the drastic change at high speed, and also problematic for proper maneuvering of the vehicle. An initial negative camber of 1° is provided in the model, to suite the high-speed cornering conditions on the track. The actuation of the Anti-roll bar is mostly in the reciprocal wheel travel situation, so this improve the cornering behavior without affecting the straight path response. Two different sets stiffness values can be obtained from same setup according to the input condition.

4. CONCLUSION

The purpose of this research is to quantify the results published previously, involving the determination of the suspension system parameters and the FEM modelling of the components involved in the typical FSAE vehicle. Further, the incorporation of an Anti-roll Bar in the Front Suspension System was analyzed for different parameters. A comparative study towards the effect on the handling response with and without the ARB is made.

The following conclusions has been summarized below:

1. Camber Angle is reduced from -3.5° to -2.5° and the reduction in Steering angle is displaying the increase of understeer effect.
2. ARB are designed to minimize the roll of the body which we can clearly see in the graph of Wheel Travel Vs Roll Angle, as it is now lying between ±3.8°. This will improve the cornering stability of the vehicle without affect the harshness of the ride response.
3. There is significant change in the Roll center height and the track-change as the ARB is now resisting the lateral movement/shift of the vehicle.
4. From the FEM study, the obtained Factor is Safety is sensible for the critical parts.
5. Safety factor of more than 3.5 indicates that there is a possibility of some weight reduction in the Rocker.

ACKNOWLEDGEMENTS

The authors gratefully acknowledge team ZHCET Formula Racing, SAE-ZHCET Collegiate Club, Department of Mechanical Engineering, A.M.U., India, for providing access to the software resources and laboratory support.

DECLARATION OF CONFLICTING INTERESTS

The author(s) declares no potential conflicts of interest with respect to the research, authorship and/or publication of this article.

REFERENCES

[1] S. F. S. C. Competition, "SAE Student Event - Formula Student Combustion." https://www.sae.org/attend/student-events (accessed Apr. 18, 2021).

[2] Y. S. Saurabh *et al.*, "Design of Suspension System for Formula Student Race Car," *Procedia Eng.*, vol. 144, pp. 1138–1149, 2016, doi: 10.1016/j.proeng.2016.05.081.

[3] M. D. Milliken WF, *Race Car Vehicle Dynamics*, R-146. SAE International, 1995.

[4] L. Zhang, H. Liu, Y. Xu, S. Wu, and L. Gu, "Study on modeling method of anti-roll bar using one dimensional beam element," *SAE Tech. Pap.*, 2009, doi: 10.4271/2009-01-1454.

[5] K. Hubert and A. Kumar, "Anti-roll stability suspension technology," *SAE Tech. Pap.*, no. 724, 2005, doi: 10.4271/2005-01-3522.

[6] L. Wang, G. Xu, N. Zhang, and H. Roser, "Experimental comparison of anti-roll bar with hydraulically interconnected suspension in articulation mode," *SAE Tech. Pap.*, vol. 2, 2013, doi: 10.4271/2013-01-0710.

[7] S. Y. Ribeiro and M. E. Silveira, "Application of finite element method in the study of variables that influence the stiffness of the anti-roll bar and the body roll," *SAE Tech. Pap.*, vol. 13, 2013, doi: 10.4271/2013-36-0643.

[8] W. Shi, C. Wang, and Z. Li, "Improving Light Bus Handling and Stability by Anti-roll Bar and Bushing Adjustment," *SAE Tech. Pap.*, vol. 2015-March, no. March, 2015, doi: 10.4271/2015-01-0026.

[9] S. S. Kelkar, P. Gautam, S. Sahai, P. S. Agrawal, and R. Manoharan, "A detailed study on design, fabrication, analysis, and testing of the anti-roll bar system for formula student cars," *SN Appl. Sci.*, vol. 3, no. 3, 2021, doi: 10.1007/s42452-021-04279-z.

[10] J. W. Wirawan, Ubaidillah, R. Aditra, R. Alnursyah, R. A. Rahman, and S. I. Cahyono, "Design analysis of formula student race car suspension system," *AIP Conf. Proc.*, vol. 1931, no. February, 2018, doi: 10.1063/1.5024110.

[11] L. Sun, Z. Deng, and Q. Zhang, "Design and strength analysis of FSAE suspension," *Open Mech. Eng. J.*, vol. 8, no. 1, pp. 414–418, 2014, doi: 10.2174/1874155X01408010414.

[12] B. Zhu and N. Sun, "Design and Optimization of FSAE Race Car suspension system," *Proc. 3rd Int. Conf. Mech. Eng. Intell. Syst. (ICMEIS 2015)*, vol. 26, no. Icmeis, pp. 1019–1022, 2015, doi: 10.2991/icmeis-15.2015.193.

[13] A. A. Vadhe, "Design and Optimization of Formula SAE Suspension system," *Int. J. Curr. Eng. Technol.*, vol. 8, no. 3, pp. 0–2, 2018, doi: 10.14741/ijcet/v.8.3.17.

[14] M. T. Yogesh Kumar, "Selection of Spring For Spring Coil Dampers Using An Iterative Approach," *Int. J. Sci. Adv. Res. Technol.*, vol. 6, no. 5, p. 4, 2020, [Online]. Available: http://ijsart.com/Home/IssueDetail/37706.

[15] J. C. Dixon, *Suspension Geometry and Computation*. John Wiley & Sons Ltd, 2009.

[16] S. Chepkasov, G. Markin, and A. Akulova, "Suspension Kinematics Study of the " Formula SAE " Sports Car," *Procedia Eng.*, vol. 150, pp. 1280–1286, 2016, doi: 10.1016/j.proeng.2016.07.288.

[17] A. Mittal, M. Faraz, and Kashif, "Study of Vehicle Dynamics of a Formula Student Car," Aligarh Muslim University, 2016.

[18] D. B. Ren, X. M. Fan, J. F. Wang, and Y. Q. Zhang, "Simulation and optimization of FSAE racing suspension," *Adv. Mater. Res.*, vol. 765–767, pp. 366–369, 2013, doi: 10.4028/www.scientific.net/AMR.765-767.366.

[19] C. Arévalo, A. Medina, and J. Valladolid, “Estudio cinemático y dinámico del sistema de suspensión de un monoplaza de competencia eléctrico Formula Student Kinematic and dinamic study of the suspension system of an electric single seater competition Formula Student,” pp. 96–107, 2018.

[20] M. L. Junior, J. Greaves, C. Smout, and M. Gimeno-fabra, “Kinematic Design and Finite Element Analysis of a Suspension System for a Four Wheel Drive Electric Formula Student Vehicle Kinematic Design and Finite Element Analysis of a Suspension System for a Four Wheel Drive Electric Formula Student Vehicle,” no. April, pp. 0–16, 2020, doi: 10.13140/RG.2.2.16653.49129.

[21] P. Bharane, K. Tanpure, A. Patil, and G. Kerkal, “Design, Analysis and Optimization of Anti-Roll Bar,” *J. Eng. Res. Appl. www.ijera.com*, vol. 4, no. 9, pp. 137–140, 2014, [Online]. Available: www.ijera.com.

[22] B. Harshal, K.Rushikesh, P.Baskar, “Finite Element Analysis of Anti-Roll Bar to Optimize the Stiffness of the Anti-Roll Bar and the Body Roll,” *Ijmer*, vol. 4, no. 5, pp. 11–23, 2014, [Online]. Available: http://www.ijmer.com/papers/Vol4_Issue5/Version-3/IJMER-45031123.pdf.

[23] A. Maske, S. Mane, A. Maske, M. Truck, S. Academy, and O. Engineering, “Ride and Handling Improvement of Passenger Bus #1,” no. June, pp. 466–472, 2016.

[24] C. Tech, “2021 Formula Bharat Rules Booklet,” vol. V1.2, p. 125, 2021, [Online]. Available: https://www.formulabharat.com/downloads/.

[25] F. Germany, *Formula Student Germany Rulebook 2020*, 1st ed. 2020.

[26] O. De Garston, “Formula Student Car Suspension Design Motorsport Engineering Formula Student Car Suspension Design.”

[27] V. Sakthi Murugan *et al.*, “A numerical approach to suspension kinetics analysis of fsae CAR,” *Int. J. Mech. Eng. Technol.*, vol. 8, no. 10, pp. 910–917, 2017.

[28] S. C. Society of Automotive Engineers, *SAE Spring Design Manual*, 2nd ed. Society of Automotive Engineers, 1996, 1996.

[29] IPG Automotive GmbH, “Formula CarMaker Tutorial 1.0,” pp. 1–104, 2016, [Online]. Available: https://ipg-automotive.com/fileadmin/user_upload/content/Specials/FormulaCarMaker_Tutorial_2017.pdf.

[30] V. B. Bhandari, *Design of Machine Elements*. Tata Mcgraw-Hill Publishing Company Limited, 1994, 2017.

[31] SKF Group 2018, “SKFs Bearing and Mounting Products,” 04/2018., S. M. Group, Ed. SKF Group, 2018, p. 587.