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Design and Manufacturing of Suspension and Steering System of a F3 Vehicle

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Abstract

The suspension and steering systems are critical subsystems of any formula-style racing vehicle, directly influencing its stability, handling, and driver safety. This paper focuses on the design, analysis, and manufacturing of suspension and steering systems for a Formula Student F3 vehicle. The primary objective is to develop a lightweight, reliable, and efficient design that complies with Formula Student competition rulebooks while ensuring optimum ride quality and performance. Using advanced computer-aided design (CAD) tools such as SolidWorks and engineering simulation software including Lotus and ANSYS, the suspension and steering geometries were developed and validated through numerical analysis.The suspension system was designed with a double-wishbone geometry combined with push-rod and pull-rod actuation mechanisms, which provide superior motion ratios, improved aerodynamics, and better cornering stability. Key design parameters such as scrub radius, caster angle, camber variation, suspension travel, and roll center height were optimized to minimize bump steer, improve handling, and maintain tire-road contact during dynamic events. The ratio of wheelbase to trackwidth was found to provide a balance between high-speed stability and maneuverability. AISI 1018 steel was selected for the spaceframe chassis due to its high strength-to-weight ratio, excellent weldability, and durability under high torsional and impact loads.The steering system was developed using Ackerman geometry to ensure precise cornering performance. Calculations for steering angles, rack-and-pinion geometry, tie-rod lengths, and turning radius were performed to achieve an Ackerman percentage close to the ideal value, thereby reducing tire slip and enhancing driver control. Simulation and analysis confirmed that the chosen suspension and steering configurations provided adequate stiffness, durability, and safety margins under braking, cornering, and impact conditions.The results of this work demonstrate that the integrated suspension and steering system not only meets competition safety and design guidelines but also enhances performance through improved handling, reduced weight, and driver comfort. This design methodology can serve as a reference framework for future Formula Student teams aiming to optimize their vehicle's performance.

Keywords: Suspension, steering, chassis, scrub, castor, acker man, toe

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INTRODUCTION

Formula Student vehicles demand suspension and steering systems that balance agility with stability while adhering to competition safety requirements [1, 2]. A double-wishbone layout is preferred for precise kinematic control, allowing independent tuning of camber gain, roll center height, and motion ratios. The steering employs an Ackermann geometry to reduce tire scrub in low-speed turns and improve predictability at the limit. The design process follows the SAE Formula Student and Supra SAE rulebooks [3–7], which strongly influence hard-point locations and packaging. Figure 1 presents the CAD overview of the front module, and Table 1 summarizes the primary geometric targets adopted at the concept stage.

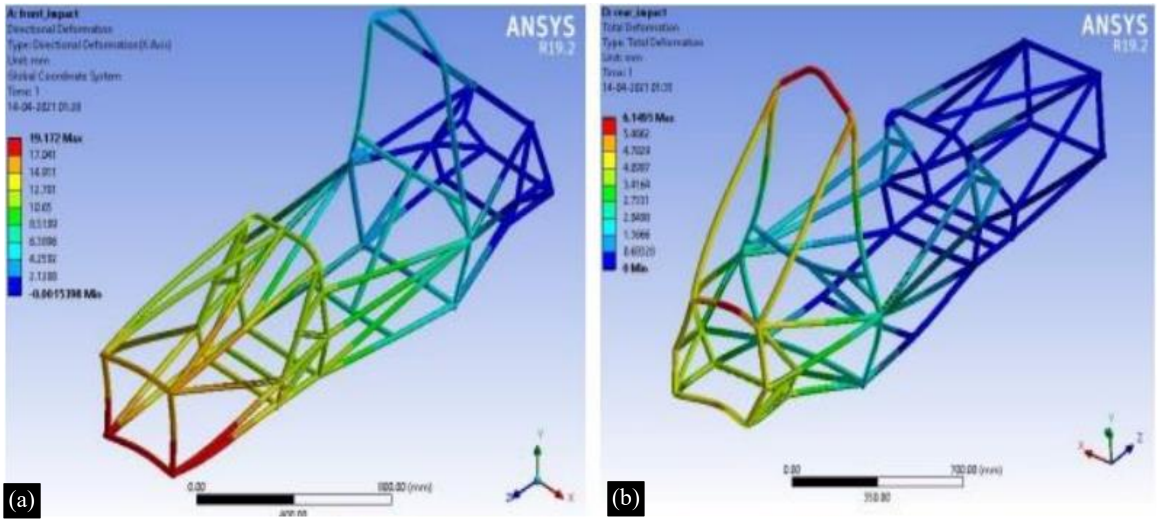

Figure 1. (a) Front impact on chassis, (b) Rear impact on chassis.

Table1. Dimensions of vehicle.

Overall dimension	mm	Length	2875	Width	1400	Height	1185
Wheelbase & trackwidth	mm	Wheelbase	1574.8	Front Track	1193.8	Rear Track	1193.8
Mass without driver	kg	Front	115	Rear	115	Total	230

LITERATURE REVIEW

CAD–CAE workflows have been shown to shorten development cycles and improve design quality by enabling rapid iteration. Emphasis has been placed on chassis–suspension integration, while principles of vehicle dynamics are explained as guides for kinematic tuning. Steering geometry considerations are particularly important for student formula cars. Competition rules impose strict limits on ergonomics and crash safety. Classical fundamentals on suspension components have also been provided [8–12].

Simulation has been applied for lithium-ion battery validation in EVs, demonstrating the role of CAE beyond mechanical systems. Contributions have been made to suspension kinematics, brake systems, and process parameter optimization. Torsional stiffness of BAJA chassis has been validated using CAE and gyroscopic sensors. Efforts have focused on suspension and emission system optimization. CV axle durability for BAJA has been analyzed. Research has optimized calipers and master cylinders, and there has been work on Efficycle suspension–steering design [13–20].

This review establishes CAE and CAD integration as key enablers for suspension–steering design in Formula Student.

Weight: A vehicle target weight was set at 230 Kg.

METHODOLOGY

A steel tubular space frame chassis was chosen for its better weight to rigidity ratio as compared to other chassis types. Moreover, minimum material is required for space frame. The chassis weight is 36 kg with a torsional stiffness of 3390 Nm/deg. The chassis structure features a complete triangulation which is an add-on to its structural rigidity, torsional stiffness and bending stiffness. Also, major impacts are absorbed progressively. Thus, in the case of a high-speed collision, although the car may be extensively damaged, the fact that it slows down progressively often minimizes injuries to the driver.

CAD and Kinematics

Chassis hard-points and wheel centers were modeled in SolidWorks. A kinematic template swept bump/rebound and steer inputs to extract camber, toe, and roll center trends. Figure 2 shows the front suspension hard-point layout; Figure 3 plots the bump-steer curve for ±50 mm wheel travel.

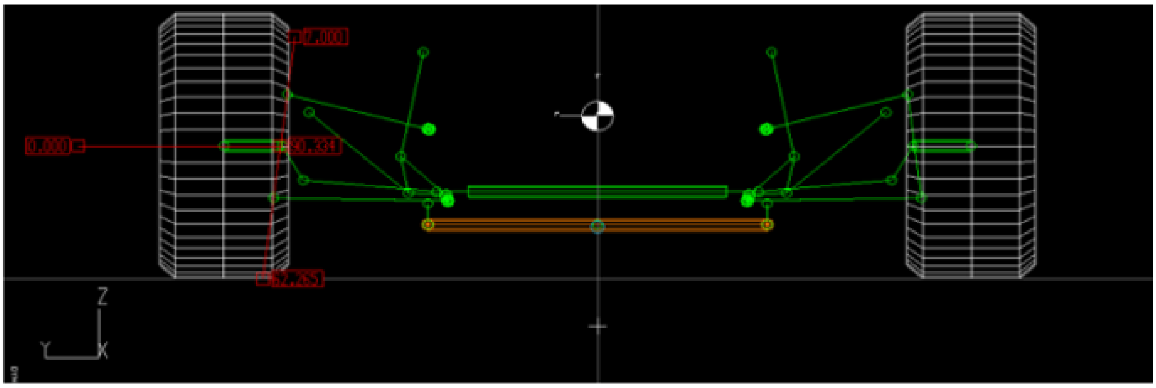

Figure 2. Front suspension pictorial representation.

Table 2. Frame specification.

Frame construction	Steel spaceframe
Material	AISI 1018
Joining Method and Material	TIG Welding; Filler Material: Copper

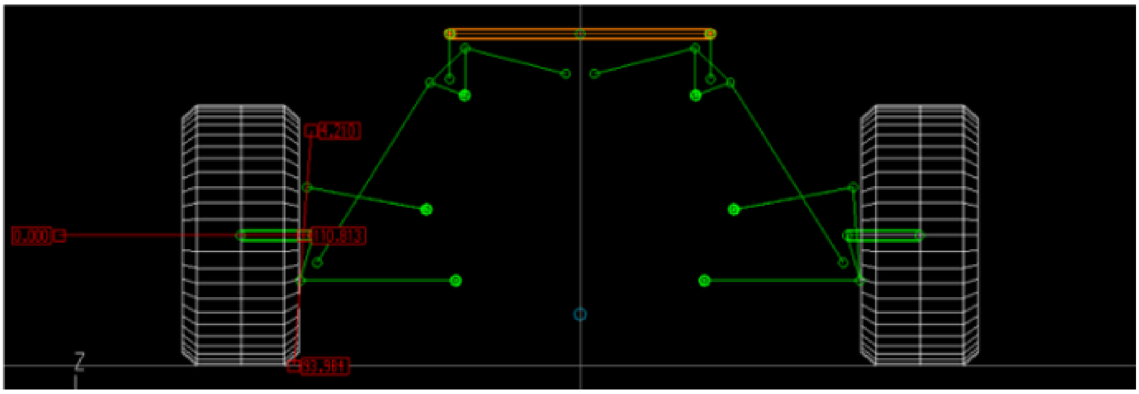

Figure 3. Rear suspension pictorial representation.

Load Cases and FEA

Static and quasi-static loads were derived for braking, cornering, and vertical impacts per competition guidelines. Component-level FEA employed refined meshing at welds and bearing interfaces. Material data for AISI 1018 steel and aluminum alloys were taken from Stockel & Duffy and Giri. Tables 2 and 3 lists the material properties used in simulations.

Steering Geometry

Rack position, tie-rod length, and steering arm offset were iterated to reach a target Ackermann percentage while controlling bump-steer. Figure 4 illustrates the Ackermann construction and intersection lines used to set steering arm angles.

MATERIAL

AISI 1018 STEEL is used as the production material for the frame. AISI 1018 (MID/LOW CARBON STEEL) is regarded as the ideal steel for carburized parts because of its superior weldability, which results in a uniform and tougher casing. The mild/low carbon steel AISI 1018 provides a good mix of strength, ductility, and toughness. The tabular dimensions of the members are 1in outer diameter and 2.5 mm thick, 1in outer diameter and 1.6 mm thick.

SUSPENSION

The mechanism that connects the body to the wheels is called the suspension system. The suspension restricts the motion. Maintaining a consistent tire contact patch during various dynamic events is the suspension's primary goal. This covers braking, accelerating, and cornering. Through the suspension, various forces and motions between the wheels and the ground are transmitted to the body. A key component of the overall vehicle design that affects the racing car's performance is the suspension system.

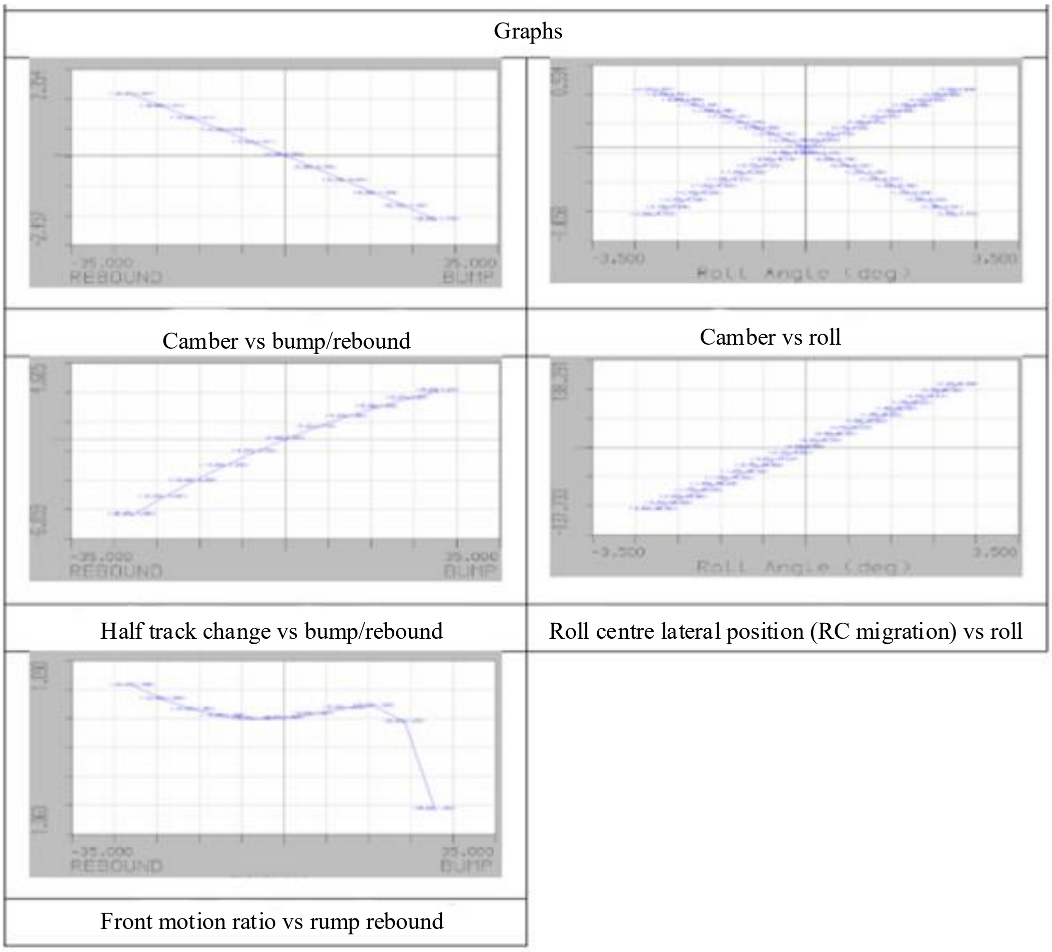

Figure 4. Front suspension characteristics.

Wheelbase-Trackwidth Determination

As the trackwidth increases the trackwidth the total lateral weight transfer decreases but having a very large trackwidth can cause trouble for the driver in narrow tracks. Also, a large trackwidth or wheelbase results increased weight. Wheelbase of an ordinary road car is observed to be more or less 1.4–1.7 times the trackwidth but in fast moving vehicles which demand more high-speed stability, the ratio seems to increase by a considerable amount. The wheelbase to trackwidth ratio for current generation Formula 1 cars is close to 2:1. It is observed that car having wheelbase in the range of 1.5–1.7 m and trackwidth around 1.2 m performs best in the Formula Student according to an article in Derek Seward's Race Car Design.

After looking at all the factors, the following values have been decided.

Trackwidth = 47 inches or 1193.8 mm (for both front and rear ends)

Wheelbase = 62 inches or 1574.8 mm

Ground Clearance = 1.356 inches or 34.5 mm

SAI, caster angle, scrub radius and castor trail:

Formula Student car design begins with suspension, so a number of terminology, values, and decisions must be taken into account during the suspension design process. Scrub radius has to be kept minimum so as to minimize the aligning torque experienced by the driver. The aligning torque can be calculated using the following equation:

M=NCOS(ΘCASTER)COS(ΘSAI)SIN(ΘSAI)[RCASTERTAN(ΘSAI)+RSCRUB]SIN(ΘSTEER)

The upright in the wheel assembly is placed in the inside portion of the hub which also includes the brake assembly. Hence, the wheel assembly should be designed in such a way that it takes minimum space for the upright to be as close as the wheel center as possible. This would mean with brake caliper taking minimum space, the lower ball joint can be positioned as outwards as possible to keep the scrub radius at its minimum.

As discussed earlier in the Introduction, SAI, and Caster angle are responsible for dynamic camber angle due to rotation of wheel around steering axis. An excel is made with all the possible configurations with SAI, between 5° to 11°, and Caster angle, between 2° to 5°, varying with the steering angle from 0° to 30° in a 5° incremental fashion. The total camber angle at each steering angle is observed and the most favorable configurations of SAI and Caster angle are listed out in Tables 4–6:

Table 3. Suspension parameters.

Suspension parameters	Units	Front	Rear
Tire size, Compound and Make		205/50-10 RADIAL	205/50-10 RADIAL
Wheels (width, construction)	mm	diameter-254, material - Aluminium width-157.76, offset-0	diameter-254, material – Aluminium width-157.76, offset-0
Suspension Type		Double Wishbone Pull-Rod	Double Wishbone Push-Rod
Suspension Design Travel	mm	Jounce: 25.4 Rebound: 25.4	Jounce: 25.4 Rebound: 25.4

Table 4. SAI and Scrub radius possibilities.

	1	2	3	4	5	6	7
SAI	5	6	7	8	9	10	11
SCRUB	74.5	72.11	69.71	67.3	64.88	62.44	59.99

Table 5. SAI and caster angle possibilities.

	1	2	3	4	5	6	7
SAI	5	6	7	8	9	10	11
Caster	2	2	3	3	3	4	4

Table 6. Aligning moment outcomes.

Q-steer (in degrees)	M1	M2	M3	M4	M5	M6	M7
5	4.17	4.83	5.39	6.03	6.50	6.84	7.35
10	6.15	9.68	10.75	11.78	12.70	13.37	14.38
15	11.98	14.37	16.02	17.33	18.67	19.67	21.15
20	16.300	18.99	21.17	23.78	25.40	26.25	28.76
25	20.14	23.46	26.16	29.12	36.38	33.05	35.53
30	23.97	27.76	30.95	34.67	37.35	39.35	42.30

These are the final selected front geometric parameters based on the above analysis:
SAI = 7°
Caster Angle = 3°
Scrub Radius = 69.71 mm
Caster Trail = 10 m

Damper-Actuation Systems

Front (pull-rod actuated): Double wishbone pull rod has been used this time. Pull-rod actuation improves the shockers' motion ratio when compared to directly actuated shockers, allowing for even less stiff springs to be utilized for the same wheel stiffness. This significantly improves the vehicle's aerodynamic efficiency and lowers the COG height. The amount of positive camber that is introduced to the inner wheel during roll is reduced with this suspension setup. The inclination of A-arms provides static roll centre at 7.936 mm in front above the ground plane and provide static camber of –1°. At 1g lateral acceleration we are getting 0.53300 roll and we are getting 0.140° camber change. The static caster and KPI was decided 3° and 7°, respectively. With these values we have tried to achieve minimum camber change while steering. The suspension utilizes 54.77 N/mm linear springs. This value corelates to a wheel rate of 46.95 N/mm.

Rear (push-rod actuated): Double wishbone pushrod suspension has been used. Since the motion ratio of 1.08 offered by pushrod actuation is significantly lower than that of directly actuated actuation, even less stiff springs can be employed to achieve the same wheel rigidity. This suspension configuration minimizes the introduction of positive camber to the inner wheel during roll. The inclination of A-arms provides static roll centre at 9.026 mm in rear above the ground plane. At 1 g lateral acceleration we are getting 0.03° camber change. Rear suspension utilizes 43.89 N/mm linear springs. This value corelates to a wheel rate of 37.63 N/mm.

Wheel and Tires

Tires are one of the most important components of our vehicle as they affect our comfort, performance, and safety. We have chosen BKT 205/50–10 radial tire for our vehicle because radial tires are flexible in nature, have very less wear and tear on rough roads and very high traction with the road.

COMPONENT SELECTION

Dampers = DNM RCP 2S for front and rear.
Bearings = ACB 7206 BEGAY (front) and DGBB 61814 (rear)
Wheels = BKT 205/50 – 10
Upright = EN24 (front) and AL 6061T6 (rear)
Hubs = AL 6061T6 (front and rear)
Rocker = AL 6061T6 (front and rear)

STEERING

The steering calculations has been done on the basis of Ackerman geometry. As per calculations, Outside angle = 27 degrees Inner angle = 40 degrees Ackerman percentage is 103.49% for a turning radius of 2.5m. The manual rack and pinion is mounted to the base of the chassis for ease of the driver, to lower the COG and in order to satisfy bump steer. In order to satisfy the bump steer, the tie rod length has been increased which will cause us a little toe in to our front wheels. The newly fabricated steering wheel (of 130 mm radius) providing us better grip and required rack travel. The advantages of increased leg room and improved grip because of the flat bottom and hemispherical top, which are both lightweight.

NUMERICAL ANALYSIS

Suspension

a. Load Transfer

Track Width: 1194 mm, 47”
Wheelbase: 1524 mm, 60”

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Weight: 300 kg, 2940 N
COG Height: 280 mm, 11”
COG Length: 787.5 mm, 31”

Static Load Distribution
Rear axle load = Wr = W * Lm/L = 1470 N (1)
Front axle load =Wf = W–Wr = 1470 N (2)
Weight distribution Weight on front = Wf = 50% Weight on rear = Wr = 50%

Dynamic load distribution
Uring braking:
G force = 1.25g
$\Delta Wx = \pm hm * w * Ax/L$ (3)
Maximum load on front corner =1073 N
During acceleration
G force = 1.33g
$\Delta Wx = \pm hm * w * Ax/L$ (4)
Maximum load on rear corner = 1094 N

During cornering
G force = 1.8 g
$\Delta w = H * w * Ax/T$ (5)
Maximum load on front corner = 1241/2 + 1470/2 = 1355.5 N
Hub diameter Wvertical = 1355.5 N
Wlateral = 1355.5 * 1 = 1355.5 N
Rr = 228.6 mm
Maxle = WlateralRr – Wvertical * l (6)
=214982 Nmm
Yield stress,
6061 – 310 N/mm_2
7075 – 435 N/mm_2
Z = safety factor * Maxle/σy (7)
= 3 * 214982/310
= 2080.47

b. Minimum Radius of the Axel
$Z = \pi r3/4$ (8)
$R = \sqrt[3]{} 4Z/\pi$
Min R = cube root4 * 2080.47/3.14
= approx. 14 mm
spindle dia = 28 mm
Lets assume inter bearing distance to be 40 mm
Effective bearing spacing = 40 – 15 + (2 * 24) = 73 mm

Case 1 – Cornering
Fvertical = 1355 N
Flateral =1355 N
Moment about Y,
P1i = 3222 N
Moment about X,
P2i = 1790 N

Case 2 – Braking
Fvertical = 1073 N
Flongitudinal = 1180 N

Inner Bearing Vertical Load = 1073 * 55/73 = 808.4 N
Inner Bearing Longitudinal Load = 1180 *55/73 = 889 N
Resultant, P2i = 1202 N
Outer Bearing Vertical Load = 1073 * 128/73 = 1881.42 N
Outer Bearing Longitudinal Load = 1180 * 128/73 = 2069.04 N
Resultant, P2o = 2797 N
Maximum of all,
P1i = 3222 N
So = 9300/3222 = 2.88
Hence, 7206 ACBB is convincingly resistant to applied potential loads.

c. Rear Bearing Selection and Upright Calculations
Taking moments about the inner bearing centers
Case 1 - cornering
Fouter = {(Wlat – Rr) – (Wvert* (l1+l2)} / l1 (10)
= 11382 N
Finner = Fouter + Wvertical (11)
= 11382 + 1355= 12737 N

Rear Bearing Selection
Maximum acceleration
Taking moments about the inner bearing centres:
Vouter¬ = Wvert * (l1 + l2) / l1 (12)
= 8666N
Houter = Wlong * (l1 + l2) / l1 = 1202 * (111)/14 = 9530 N
Resultant = 12881 N

Summing vertical forces:
Vinner = Vouter – Wvert (13)
= 8666 – 1093 = 7573 N

Summing horizontal forces:
Hinner = Houter – Wlong (14)
= 9530 – 1202
= 8328 N

For this, we have considered DGBB 61814 bearing.
A-arms calculation
Front

Case – Cornering
Wvertical = 1355.5 N
Wlateral = 1355.5 N
Fpushrod = Wvertical/ sin(41.61) (15)
= 1355/0.664
= 2040.66 N
Hpushrod = 1525.76 N
Ftop = Wlateral * H1/H2 (16)
= 1355*189.83/129.45
= 1987.01N
Fbottom = Wlateral *(H1 + H2)/H2 (17)

= 1355*319.34/129.45
= 3342.647 N

Case Braking
Wvertical = 1073 N
Wlateral = 1073 N
Fpushrod = Wvertical/ sin(41.61) (18)
= 1073/0.664
= 1616 N
Hpushrod = 1209 N
Ftop = Wlateral * H1/H2
= 1073 * 189.83/129.45
= 1573.481 N
Fbottom = Wlateral * (H1 + H2)/H2
= 1073 * 319.34/129.45
= 2647 N

Rear
Case – cornering
Wvertical = 1355.5 N
Wlateral = 1355.5 N
Fpushrod = Wvertical/ sin(57.85) (19)
= 1355/0.847
= 1600 N
Hpushrod = 851.42N
Ftop = Wlateral * H1/H2
= 1355 * 119.71/150
= 1081.38 N
Fbottom = Wlateral * (H1 + H2)/H2
= 1355 * 269.71/150
= 2436.38 N
Case Acceleration
Wvertical = 1094 N
Wlateral = 1094 N
Fpushrod = Wvertical/ sin(57.85) (20)
= 1094/0.847
= 1291.6 N
Hpushrod = 687.30 N
Ftop = Wlateral*H1/H2
= 1073*119.71/150 = 856.32 N
Fbottom = Wlateral * (H1 + H2)/H2
= 1073 * 269.71/150
= 1929.32 N

d. Dampers
Front Mass Natural frequency = 3.15
Rear Mass Natural frequency = 2.82
These values are assumed according to past results and research papers

SPRING RATE
Front Ks = 4 π^2 * (Ff)^2 * Msm * MR^2 (21)
Ks = 54,773 N/m or 54.77 N/mm
Rear Ks = 4 π^2 * (Fr)2 * Msm * MR^2 (22)
Ks = 43,959 N/m or 43.89 N/mm

e. Wheel Rate
Wheel Rate = Ks/MR^2 (23)
Front Wheel Rate = 46,959 N/m or 46.95 N/mm
Rear Wheel Rate = 37,635 N/m or 37.63 N/mm
ROLL RATE
KΦF = (π * T^2* KRF * KLF)/{180 * (KRF+KLF)} (24)
KΦF = 291.47Nm/deg
KΦR = (π * T^2 * KRR * KLR)/{180 * (KRR+KLR)} (25)
KΦR = 233.60 Nm/deg

Roll Gradient
Front wheel rate = 46959 N/m
Front Chassis roll stiffness = (t2 * kwheel)/2 * (360/2pie) (26)
= 582.94 Nm/deg
Rear wheel rate = 37635 N/m
Rear Chassis roll stiffness = (t2 * kwheel)/2 * (360/2pie) (27)
= 467.19 Nm/deg
Total Chassis roll stiffness = 582.94 + 467.19 = 1050.93 Nm/deg
Torque = (vehicle mass * lateral acc * vd) (28)
Chassis roll angle = 560.93/1050.93
= 0.533 deg
At 1g lateral acc. we are getting 0.533 deg roll.

Steering
Wheelbase (L) = 1574.8 mm
Trackwidth (T) = 1193.8 mm
Turning Radius (R) = 2500mm (Considered according to autocross event)

First we need to obtain tyre inside and outside angles in order to calculate Ackerman percentage. Therefore,
Tyre outside angle (Ao) = tan(Ao) = L / (R + T/2) (29)
= 27 degrees
Tyre inside angle (Ai) = tan(Ai) = L / (R – T/2) (30)
= 40 degrees

Now,
Ackerman value = tan (Q) = L / (L/tan(Ao) – T) (31)
= 38.69
Ackerman Percentage = Inside Angle / Ackerman Value (32)
= 40 / 38.69
= 103.3% ~ (103% approx.)
Steering Wheel Angle = 120 degrees
Lock to Lock Angle = 240 degrees
Lock to Lock Value = 240/360 = 0.666
Steering wheel travel = 2 * pi * r / 3 = 2 * 3.14 * 130 mm / 3 (33)
= 272.1 mm
Steering Ratio = Maximum steering wheel angle / Max angle turned (34)
= 240 / (27 + 40)
= 3.58 ~ 3.6 approx
Rack Travel = Steering Wheel travel / Steering ratio (35)
= 272.1 / 3.6
= 75.6 mm
C-factor = Linear distance travelled by rack in one complete rotation of pinion
= Rack travel / lock to lock value (36)
= 113.41.
Steering Arm Length (SAL) =

(1/steering ratio)= {sin^-1(C-factor/SAL)}/360 (37)
=115 mm
Steering Arm Angle = 20.75degrees

RESULTS

Kinematic Results

Camber gain remained within the target window across jounce/rebound, limiting outer-wheel load loss in roll [2]. Bump-steer remained below the design threshold across ±50 mm travel, as shown in Figure 3. The front roll center migration was constrained within the design band, and the motion ratios produced acceptable wheel rates (Table 3).

Structural Results

FEA results for wishbones and rockers satisfied yield and stiffness criteria with safety factors above targets under braking and cornering. Upright stress concentrations were reduced by fillet optimization and wall thickening near bearing seats. Figure 5 shows the stress contour for the front lower wishbone.

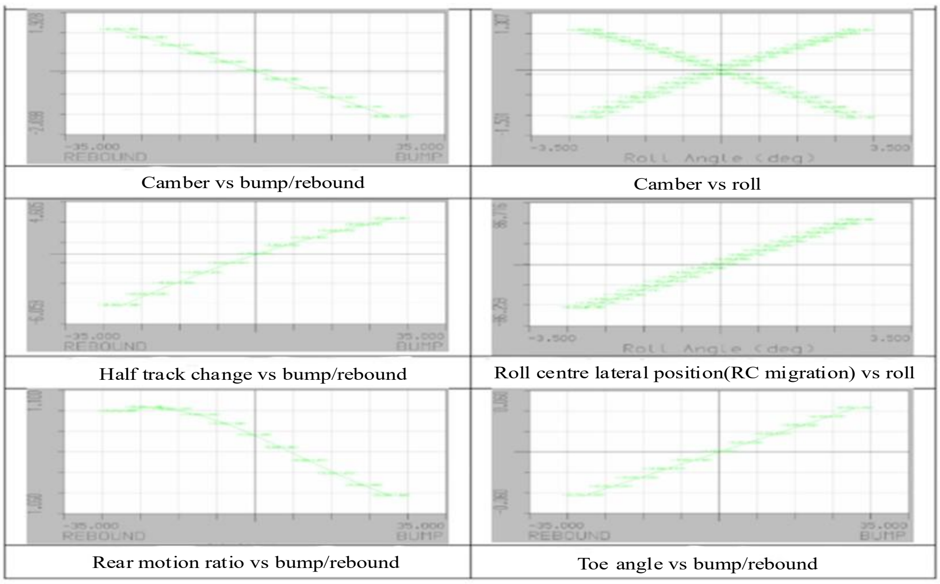

Figure 5. Rear suspension characteristics.

CONCLUSION

The developed suspension and steering systems for the F3 vehicle achieved performance and safety goals while meeting competition standards. Kinematic targets – low bump-steer, stable roll center, and appropriate camber gain – were satisfied alongside structural safety factors under braking and cornering. The CAD–kinematics–FEA workflow enabled efficient iteration and manufacturable solutions. The approach provides a practical framework for future Formula Student teams seeking predictable handling, reduced mass, and robust reliability.

Nomenclature

Wr	Rear Axle Load
W	Weight of Vehicle
lm	Length of COG from front Axle
L	Wheelbase
T	Trackwidth
hm	Height Of COG
Wf	Front Axle Load
T	Trackwidth

Ax	Axle Load
Rr	Rolling radius
Ks	Spring rate
KΦ	Roll rate
R	Radius of Spindle
H	Vertical Height
P	Moment
T	Turning Angle

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