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Design, Development & Analysis of Suspension System for All Terrain Vehicle

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***Abstract*—: An all-terrain vehicle is made to travel across any surface. This vehicle's suspension system needs to be robust to deliver a better ride, better handling, and greater comfort. For this, independent suspension systems are necessary. It is created utilizing LSSA (Lotus Shark Suspension Analysis). Following design in Lotus, CATIA is used to create the A-arms, front and rear uprights, and is then examined using ANSYS. The mechanism that attaches the wheels to the chassis via an assembly offers the rigidity required to absorb road shocks. Roll/body angle, smooth steering, camber characteristics, among many other things, are all determined by the suspension system. In order to withstand abrupt shocks brought on by drops, sudden dumps, etc., the suspension system must be rigid. The vehicle's suspension systems aid in the comfort and maneuverability of the driver. The suspension should be designed to endure rough terrain and alert driving.**

Keywords—Optimum camber, LOTUS Shark Suspension Analysis, rough terrain, Ackermann geometry variations.

Introduction

The Society of Automotive Engineers (SAE) hosts an interdisciplinary design competition called Baja SAE India. undergraduate engineering students are eligible to compete. The dynamic events include hill climbs, maneuverability competitions, suspension and traction competitions, and endurance races. The objective is to design, construct, test, and race a single-seat off-road vehicle in accordance with SAE standards.

The drive train, suspension, braking system, steering system, and chassis are all interrelated systems that make up an all-terrain vehicle. An off-road vehicle's suspension system is crucial to its performance because it keeps the wheels on the road in bouncy and droopy situations while minimizing shocks to the driver and chassis [1]. A vehicle is suspended primarily for security and performance reasons. The suspension's primary responsibility is to absorb and hold back any vertical forces that a car would encounter on an off-road track. This can range from a slight weight shift when the vehicle is loaded with people or items to a significant shift if the tyres of the vehicle continually running into a significant obstruction on the ground.[2]

The first process involves designing the suspension geometry based on the suspension parameters' initial assumptions and doing iterations to ensure that the minimum possible variation in the suspension parameters during wheel travel. The second phase involves choosing the appropriate damper after obtaining the spring rate, motion ratio, and natural frequency. The CAD model of the suspension parts is created in the third phase. The design is finalized in the fourth phase, which involves numerous simulations and optimizations using ANSYS Workbench19. The car is regularly tested and tuned to improve the design after being manufactured and put together.

In order to improve the suspension geometry, as well as subsequent design of suspension system components, the main objective of this study is to determine the suspension parameters using the selected values of camber, toe, wheel base, track width, and wheel travel.

I. DESIGN PARAMETERS

This research is founded on building an ATV suspension mechanism that complies with BAJA SAE regulations. Based on measurements of a mock chassis and considerations for the drive train, The table below lists specific parameter values required for creating the system. To ensure better track maneuverability, the width of the rear track is intentionally kept smaller than the front. For the purpose of determining the ideal value of ground clearance, the existence of rocks, bumps, and logs were taken into account.

II. ANGLES FOR SUSPENSION GEOMETRY SELECTION

A. *Camber*

To preserve traction provided by the wheel to the ground and lessen tyre wear, All-terrain vehicles should have some negative camber. To help counteract the improvement in dynamic situation, a modest negative camber is offered at static [3].

B. *Toe*

Toe in is employed in static position to counteract the tendency of the tyre to move at odds with one another while accelerating. This has an impact on the car's straight-line stability and steering reaction.

C. *Caster*

To improve the vehicle self-centering and to maintain straight-line motion, a positive castor angle is maintained between 0 and 7 degrees.

D. *Kingpin inclination or KPI*

KPI for off-road vehicles should be ranging from 4 to 11 degrees to be considered ideal. It has implications for steering effects and the wheel's vertical movement, which also adds to the self-aligning torque. The axis of steering influences the scrub radius value, KPI is therefore given a major value to reduce the scrub radius.

III. DESIGN METHODOLOGY:

A. *LOTUS analysis of geometry*

The design is finalized using the Lotus Shark suspension designing software system. The Lotus Shark Suspension Analysis is represented in figure 1. The least amount of anti-dive characteristics is used in the design to keep the ATV stable in all conditions. Before developing the car's suspension, team concentrated on a handful of the most important vehicle components.

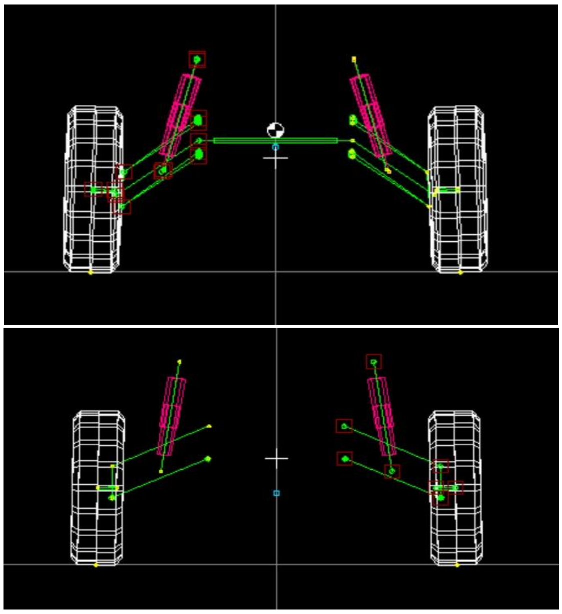

Fig 1: Lotus Shark Suspension Analysis.

Three-dimensional moving models are regularly developed and altered in the LOTUS Shark Suspension Analysis (LSSA). Using LSSA, hard points are drawn. Graphical and numerical values are calculated. This modelling strategy makes it simple for users to build their own suspension models. Diagrammatic representations of camber angle and toe angle fluctuations in connection to steering motions including roll, bump, and steering motion are possible. The damping magnitude relationship, sprung and unsprung weight, spring rate, camber angle, caster angle, roll centre, wheelbase, track size, toe angle, and ground clearance are just a few of the variables that should be considered when determining the suspension system's weak points. Therefore, prior to producing, design considerations must be made.

Here, the goal is to reduce any modification to the wheel alignment angle parameters. For effective weight transfer and to reduce tyre wear, reduced track width and wheel base modifications [4]. The final lotus iteration is displayed in accordance with the desired range of suspension angle values and the specified geometric limitations.

B. *Geometry of suspension system*

For the front, it is decided to use a double wishbone independent system. The best camber curvature throughout wheel travel is provided by a short long arm arrangement. Negative camber is incorporated onto a shorter upper arm to retain traction during tight turns. Whenever the vehicle is rolling, the inner side wheel droops and displays a favorable camber change, while the outer wheel experiences bump and exhibits a negative camber change. Turning radius and bump-steer are two steering metrics that are impacted by the front suspension geometry [5]. To reduce slip angle and understeer risk, the minimum turning radius is achieved [6].

The gearbox system's components are considered while choosing the rear suspension system. By use of a tripod joint that connects the driveshaft's inner end to the gearbox, the wheel travel is constrained due to the driveshaft's limited range of motion. In the back, a suspension system with a single upper link and an H-Arm wishbone. It exhibits toe-in and toe-out tendencies because the back wheel does not have steering, which may lead to oversteer or understeer, respectively [7]. This could lead to an imbalance and, ultimately, a loss of control. Because of this, H-arm is utilized instead of A-arm. The H-arm configuration increases ride stability and assures accurate alignment in dynamic circumstances.

Bump steer

When the road is bumpy and the car is moving, the toe angle changes. Because of the undesired wheel motion, it is minimized. Understeer and oversteer characteristics are also impacted by bump steer [4]. When there is a bump, a wheel with toe out tends to understeer, whereas a wheel with toe in tends to oversteer [9]. In order to prevent excessive tyre wear, the tyres must be realigned when the ATV hits bumps or droop. The least amount of bump steer is ideal for a desired performance. As a result, the toe change is zero because the IC point is located on the same axis as the tie rod points.

Roll center height

One of the crucial parameters, roll centre height, typically requires several cycles before reaching the ideal value. Due to the assembly's track width constraints, it is impossible to iterate the lower wishbone suspension positions. However, by altering the IC point and iterating the upper wishbone coordinates, it is possible to change the roll centre height. To boost the vehicle's stability while accounting for the 60:40 bias (Rear to Forward), It is ideal for the roll axis slope angle to be around 1 degree.

IV.MULTIBODY DYNAMICS:

Multibody dynamics is used to present the dynamic examination of the driveshaft, suspension, and steering. The impacts of bump, drop, and roll were evaluated in the suspension analysis using the Lotus Shark. Analysis can be directed at 80% Ackermann by reducing the dynamic volatility of other variables. The primary objective of the investigation was to reduce dynamic variation *fig 2, fig 3*. Graphical Representation of Bump: To combat the difficult track conditions that the car is anticipated to endure during the dynamic event, a thorough investigation of previous years was undertaken. The table shows the static set values for the automobile. CG placements and ride height were investigated to establish the ideal performance and dampening qualities.

TABLE I. STATIC CONDITION VALUES.

Front Suspension	Camber: -1 Deg	Toe: 0.5mm
	Castor Angle: 6 Degrees	Kingpin Angle: 3 Degrees
	Roll Center: 294.77mm	Kingpin Offset: 51.85mm
Rear Suspension	Camber: 0	Toe: 0
	Roll Center: 271.95	Castor Angle: 0

TABLE II. DYNAMIC CONDITION VALUES.

Analysis	Variables	Dynamic Variation	
		front	Rear
Bump & rebound	Camber Angle	Rebound: -1.652 Degrees Bump: -1.229 Degrees	Rebound: -1.061 Degrees Bump: -0.6894 Degrees
	Tow Angle	Rebound: 2.499 Degrees Bump: 2.357 Degrees	NIL
	Castor Angle	Rebound: 6 Degrees Non-Adjustable	NIL
	King Pin Angle	Rebound: 3 Degrees. Non-Adjustable.	NIL
Roll	Camber Angle	Max: 1,618 Degrees Min: -4.723 Degrees	Max: 2.071 Degrees Min: -4.070 Degrees
	Tow Angle	Max: 1.818 Degrees Min: -0.312 Degrees	Max: 0.387 Degrees Min: -0.387 Degrees
	Castor Angle	Max: 6.987 degrees Min: 6.9857 Degrees	NIL
	King Pin Angle	Max: 6.228 Degrees Min: -0.373 Degrees	NIL

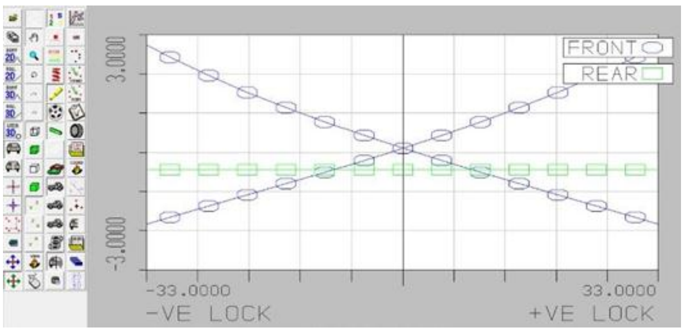

Fig 2: Steering graph results of front and rear toe changes in dynamic condition.

To get the best results for suspension parameters, multiple iterations are performed after fundamental suspension geometry hard points are entered into the Lotus software. Here are a few iterative points that can be used in the analysis: TABLE I. TABLE II.

1. By adjusting the inner pivot locations of the upper wishbone, camber change can be controlled.

2. The inner and outer tie-rod points can be adjusted to control toe change.

3. By adjusting the outer wishbone points of the higher and lower wishbone, KPI change can be controlled.

4. Track width changes slightly depending on whether it is increased or decreased.

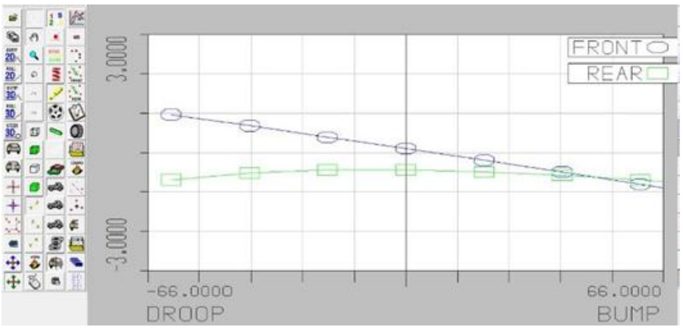

Fig 3: Bump & Droop graph results of front and rear camber changes in dynamic condition.

In addition to these fundamental locations, several additional coordinates were adjusted to determine these parameters & values inside the acceptable range. The table of data below includes the final numbers in figure *Fig 4, Fig 5, Fig 6*

FRONT SUSPENSION - BUMP TRAVEL
RHS WHEEL (+ve Y)

TYPE 1 Double Wishbone, damper to lower wishbone

INCREMENTAL GEOMETRY VALUES

BUMP TRAVEL (mm)	CAMBER ANGLE (deg)	TOE ANGLE (deg)	CASTOR ANGLE (deg)	KINGPIN ANGLE (deg)	DAMPER RATIO [-]	SPRING RATIO [-]
-20.00	-1.6528	0.4497	6.9874	3.0644	2.458	2.549
0.00	-1.5009	1.0000	6.9873	2.8453	2.482	2.583
20.00	-1.3978	1.3239	6.9871	2.7026	2.491	2.600
40.00	-1.3266	1.4996	6.9871	2.6100	2.490	2.607
60.00	-1.2781	1.5693	6.9870	2.5530	2.482	2.606
80.00	-1.2469	1.5577	6.9870	2.5233	2.469	2.599
100.00	-1.2297	1.4799	6.9870	2.5155	2.451	2.588
120.00	-1.2241	1.3447	6.9870	2.5264	2.430	2.573

INCREMENTAL SUSPENSION PARAMETER VALUES

BUMP TRAVEL (mm)	ANTI DIVE (%)	ANTI SQUAT (%)	ROLL CENTRE HEIGHT TO BODY (mm)	ROLL CENTRE HEIGHT TO GRND (mm)	HALF TRACK CHANGE (mm)	WHEELBASE CHANGE (mm)	DAMPER TRAVEL (mm)	SPRING TRAVEL (mm)
-20.00	-8.07	0.00	491.22	511.22	-16.66	1.17	8.09	7.79
0.00	-11.64	0.00	440.98	440.98	0.00	0.00	0.00	0.00
20.00	-14.25	0.00	400.36	380.36	14.00	-1.57	-8.04	-7.71
40.00	-16.40	0.00	365.91	325.91	25.77	-3.39	-16.07	-15.39
60.00	-18.34	0.00	335.68	275.68	35.62	-5.38	-24.11	-23.07
80.00	-20.21	0.00	308.43	228.43	43.74	-7.51	-32.19	-30.75
100.00	-22.09	0.00	283.34	183.34	50.29	-9.76	-40.32	-38.46
120.00	-24.07	0.00	259.81	139.81	55.36	-12.09	-48.51	-46.21

Fig 4: Bump & Droop dynamic values of front suspension.

REAR SUSPENSION - BUMP TRAVEL
LHS WHEEL (-ve Y)

TYPE 2 H-frame Lower, single upper link

INCREMENTAL GEOMETRY VALUES

BUMP TRAVEL (mm)	CAMBER ANGLE (deg)	TOE ANGLE (deg)	CASTOR ANGLE (deg)	KINGPIN ANGLE (deg)	DAMPER RATIO [-]	SPRING RATIO [-]
-20.00	-1.0612	0.0000			2.144	2.144
0.00	-1.0000	0.0000			2.155	2.155
20.00	-0.9392	0.0000			2.163	2.163
40.00	-0.8783	0.0000			2.167	2.167
60.00	-0.8167	0.0000			2.168	2.168
80.00	-0.7539	0.0000			2.166	2.166
100.00	-0.6894	0.0000			2.163	2.163

INCREMENTAL SUSPENSION PARAMETER VALUES

BUMP TRAVEL (mm)	ANTI DIVE (%)	ANTI SQUAT (%)	ROLL CENTRE HEIGHT TO BODY (mm)	ROLL CENTRE HEIGHT TO GRND (mm)	HALF TRACK CHANGE (mm)	WHEELBASE CHANGE (mm)	DAMPER TRAVEL (mm)	SPRING TRAVEL (mm)
-20.00	0.00	0.01	294.77	314.77	-9.73	0.00	9.30	9.30
0.00	0.00	0.01	271.95	271.95	0.00	0.00	0.00	0.00
20.00	0.00	0.00	251.04	231.04	8.22	0.00	-9.26	-9.26
40.00	0.00	-0.01	231.46	191.46	15.04	0.00	-18.50	-18.50
60.00	0.00	0.00	212.89	152.89	20.54	0.00	-27.73	-27.73
80.00	0.00	0.00	195.15	115.15	24.80	0.00	-36.96	-36.96
100.00	0.00	0.00	177.95	77.95	27.84	0.00	-46.20	-46.20

Fig 5: Bump & Droop dynamic values of rear suspension.

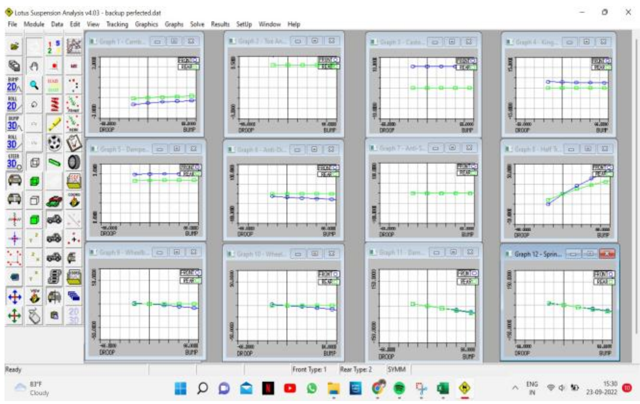

Fig 6: Graph representing Camber, toe, castor, damper, anti-dive, anti-squat, wheelbase, wheel track & halftrack.

V. FINITE ELEMENT SUSPENSION SYSTEM ANALYSIS.

Making use of FEA, it is possible to predict how a component or group will respond to different loaded boundaries and applied loads. The product is assessed and the loads additionally, there are boundary conditions to obtain the best design. A simulation program called Ansys is used to study CAD models and components under different scenarios to be able to come up with the best design that can withstand the highest applied load.

Carrying out structural analysis, thermal analysis in a steady state, fatigue study, etc, is made possible by this software. We used str structural analysis was employed, where the boundary conditions are provided, to gain and improve the results of stress, deformation, and factor of safety [8]. The components must be able to survive every circumstance without failing, which is why the worst-case scenario is used to do the static structural analysis. The goal of optimization is to create a design that is both effective and light-weight. The mass and load transfer between sprung and unsprung objects is important in terms of material and design considerations. TABLE III. TABLE IV.

TABLE III. FRONT KNUCKLE MATERIAL.

Material	7075-T6 Aluminum
Weight of the knuckle	0.3221 Kg
Volume	1.1427e+0.005 mm^3
Nodes	29510
Elements	16210

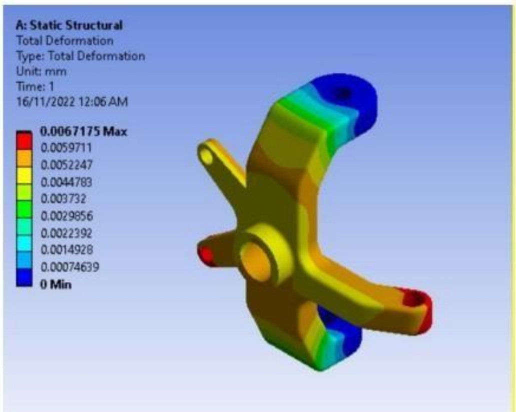

Fig 7: Front knuckle deformation.

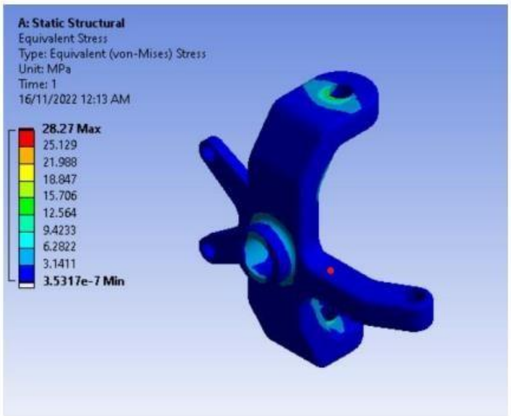

Fig 8: Front knuckle stress.

TABLE IV. REAR KNUCKLE MATERIAL.

Material	7075-T6 Aluminum
Weight of the knuckle	0.4231 Kg
Volume	1.5057e+005 mm^3
Nodes	30489
Elements	16424

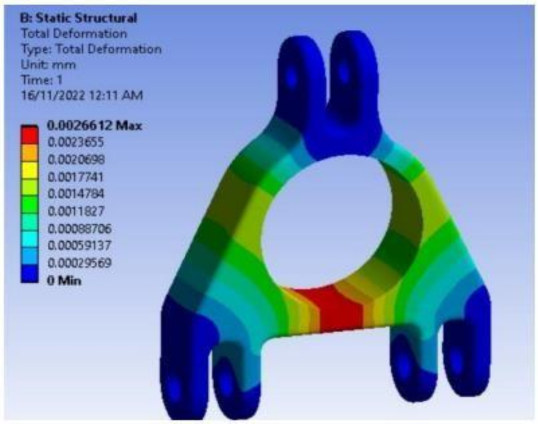

Fig 9: Rear knuckle deformation.

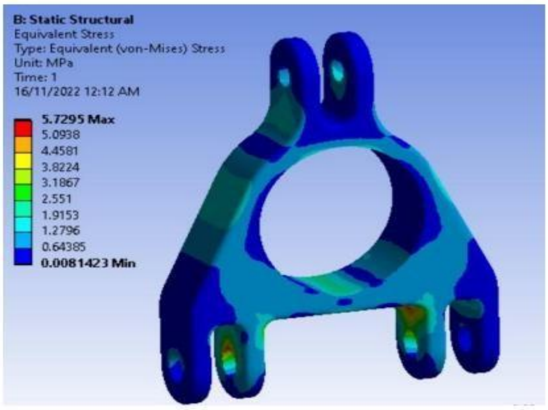

Fig 10: Rear knuckle stress.

VI. CALCULATIONS.

Total Sprung mass: 123 kg

Total Unsprung mass: 72 kg

Total Weight: 195 kg

Front track width: 45 inches

Rear track width: 47 inches

Static ride height: 9.8 inches

Tyre diameter: 22 inches

Ride frequency front: 2 Hz

Ride frequency rear: 2.5 Hz

Spring Constant:

$w = \sqrt{\left(\frac{k}{m}\right)}$

where,

w = amplitude

k = spring constant

m= sprung mass

We also know that,

w= 2πf

where f = ride frequency

on equating both the equations we get, k = 4π2mf2 N/m

using the above formula, the spring constants for the system of front and rear suspension are calculated. Quarter body analysis of the sprung mass is done so that sprung mass acting on each wheel is determined. The weight distribution according to the calculation is front: rear = 40:60.

Front spring constant:

Entire sprung mass (S_m)= 116 kg

Front sprung mass = 48.72 kg

Mutual mass (M_f): 48.72 x 1.7= 82.824 kg

The load acting on each wheel at front (m_1) = 41.412 kg

k = 4π2m1f2

k = 4 x π2 x 41.412 x 22

k = 6532.89 N/m or 6.532 N/mm

Rear spring constant:

Entire sprung mass = 116 kg

Rear sprung mass = 116 x 0.58 = 67.28 kg

Mutual mass = 114.37 kg

The load acting on each wheel at front (m_2) = 57.18 kg

k = 4π2m2f2

k = 4 x π2 x 57.18 x 2.52

k = 14094.29 N/m or 14.094 N/mm

Wheel rate:

Wheel rate = spring rate x (motion ratio) ^{2}

Front wheel rate = 6.532 x (0.5) ^{2} = 1.63 N/mm

Rear wheel rate = 14.094 x (0.7) ^{2} = 6.90 N/mm

VII. CONCLUSION

- Due to the nearly vertical tyre and largest feasible contact patch, negative camber in a static position has increased our lateral load while cornering.

- A design with a larger ground clearance in the front suspension has been made possible by a higher castor angle, allowing for easier maneuverability and flexibility from track impediments.

- Given that the anti-Ackermann mechanisms oversteer is less noticeable at slower race speeds, we picked Ackermann over it.

- The purpose of the suspension system, which aims to provide "comfort," "contact," and "control," is achieved.

- Considering the effectiveness and power characteristics, less unsprung mass was produced. Consequently, the front and rear double wishbone an H-arm in the back were successfully designed and examined.

- The investigation's findings indicated that the track and camber changes made during dynamic analysis were minimal, resulting in good stability and less bump steer.

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