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ANALYSIS OF SUSPENSION PERFORMANCE FOR BATTERY ELECTRIC VEHICLE

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Abstract

The persistent challenge of global warming and environmental pollution necessitates the adoption of sustainable and alternative mobility technologies. A significant contributor to these issues is the exhaust emissions from conventional internal combustion engine vehicles. Plug-in and hybrid electric vehicles (EVs) offer promising solutions to mitigate these concerns. This research focuses on the development of a student-designed electric car: the Battery Electric Vehicle (BEV). This urban concept car leverages compact geographical area, ideal for adopting electric cars. EVs provide advantages such as lower operating and maintenance costs and higher efficiency. By centralizing pollution to power stations, EVs offer a cleaner alternative. Despite range limitations, continuous advancements are enhancing EV capabilities. This paper details the selection and optimization of suspension parameters for the BEV's suspension system, ensuring good handling, stability, and performance while balancing optimum values with practical implementation constraints.

***Keywords:** BEV, EV, Suspension system, Wishbone, SLA. C.G., LSA*

I. INTRODUCTION

The world continues to grapple with the persistent challenge of global warming, compounded by environmental pollution which poses a significant threat to human health. A key contributor to this problem is the exhaust gases emitted by conventional internal combustion engine vehicles. Over the years, the global vehicle count is projected to reach approximately 1.8 billion, and the continuous rise in oil prices underscores the urgency of adopting sustainable and alternative mobility technologies. Plug-in and hybrid vehicles are promising alternatives that hold substantial potential for mitigating these issues. This project focuses on the research and development of a student-designed electric car. Electric vehicles (EVs) are gaining traction worldwide due to their eco-friendly nature and zero emissions. Battery Electric Vehicle (BEV) is an urban concept car that leverages compact geographical area, making it an ideal location for the adoption of electric cars. The continuous escalation of oil prices and the intensifying impact of global warming have necessitated the adoption of alternative mobility technologies. Electric vehicles (EVs) present a compelling alternative to conventional internal combustion engine vehicles [1].

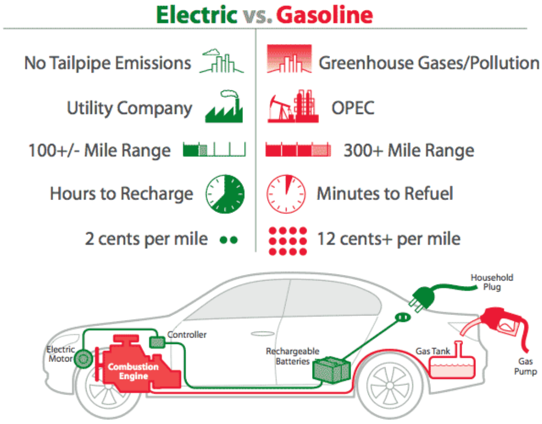

Figure 1: Electric v/s Gasoline [2]

Atmospheric pollution from exhaust gases emitted by internal combustion engines, coupled with the constant rise in oil prices, has paved the way for the adoption of electric cars globally. Electric vehicles offer significant advantages over conventional internal combustion engine vehicles, including lower operating and maintenance costs, as well as higher efficiency as shown in figure 1. Electric cars centralize pollution sources to power stations that generate electricity, whereas conventional cars distribute pollution, making them less efficient. Although the limited range of electric cars has been a discouraging factor, continuous advancements are being made to enhance their range. Numerous automakers, such as Honda, Toyota, Mercedes-Benz, BMW, and Tesla, have already commenced manufacturing electric street and sports cars as shown in figure 2 [3].

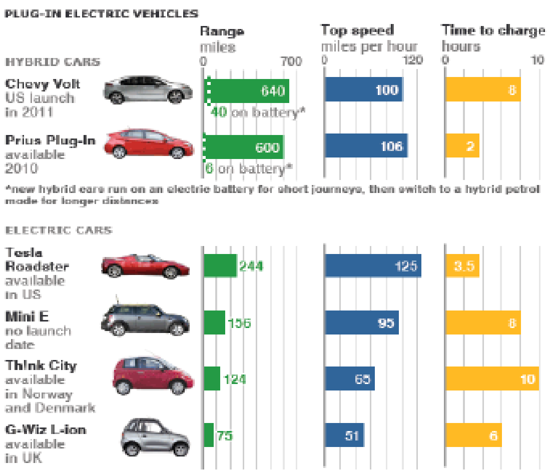

Figure 2: Different plug-in electric cars [3]

An electric car is a vehicle powered by an electric motor, utilizing electrical energy stored in devices such as batteries or capacitors. A plug-in electric car is recharged using an external electricity source. Electric vehicles, including hybrid and plug-in electric cars, are poised to play a crucial role in the future mobility landscape [5]. This project aims to challenge students to design and stimulate their creativity, fostering innovative ideas and solutions to bring first plug-in electric car to life. The concept of the Battery Electric Vehicle (BEV) revolves around generating clean energy from lithium-ion battery packs. The electrical power is directed to the motor through a controller, which in turn drives the rear differential. This paper includes detailed explanations on design, engineering, manufacturing and testing of the braking system for Battery Electric Vehicle.

II. SUSPENSION SYSTEM OVERVIEW

The suspension system is a three-dimensional four-bar linkage that includes shock absorbers, springs, and linkages, which connect the vehicle chassis to its wheels, allowing for relative motion between the two [4]. The suspension system fulfils three primary roles:

- **Comfort:** It provides vertical compliance, enabling the wheels to follow the road surface, while isolating the chassis from road roughness to enhance passenger comfort.
- **Safety**: It responds to control forces produced by the tires, including longitudinal and lateral forces, as well as braking and driving torques. This ensures the protection of passengers, luggage, other mechanical and electrical systems, and the vehicle itself.
- **Handling:** It maintains tire contact with the road with minimal load variations and resists chassis roll. Keeping the wheels in contact with the road surface is crucial, as all road or ground forces acting on the vehicle are transmitted through the tire contact patches.

Battery Electric Vehicle (BEV) features a double wishbone or short-long arm (SLA) suspension system at both the front and rear wheels. This system includes two unequal-length "A" or "wishbone" shaped control arms that are not parallel. These arms connect to the chassis and sub-frame at one end and the steering knuckle at the other. The upper arms are shorter than the lower ones and are mounted on the chassis, which helps maintain a constant wheel track [6][7]. Shock absorbers are attached to the lower wishbone and chassis to manage the vehicle's vertical movement. The double wishbone suspension is an independent design that allows for the independent movement of all four wheels, thereby eliminating wheel wobbling as illustrated in figure 3.

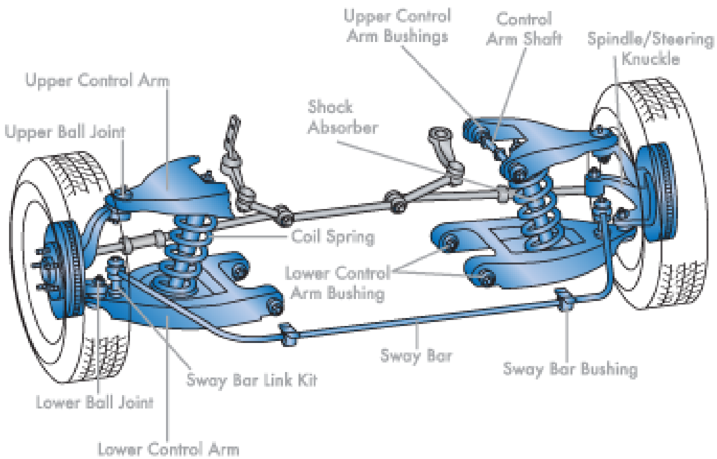

Figure 3: Double wish-bone suspension system [7]

Advantages of SLA Suspension System:

- Versatility: Allows engineers to precisely control wheel motion throughout suspension travel, managing parameters such as camber angle, caster angle, toe pattern, roll center height, scrub radius, and scuff

- Reduced Camber Angle Gain: Minimizes changes in track width
- Enhanced Handling and Safety: Improves handling, driving safety, and ride characteristics
- Lateral Stiffness: Provides good lateral stiffness to the vehicle

Disadvantages of SLA Suspension System:

- Complexity and Cost: Requires more components, increasing manufacturing costs and space requirements
- Installation Requirements: Necessitates a front and rear sub-frame, raising assembly costs, complexity, and vehicle weight [8]

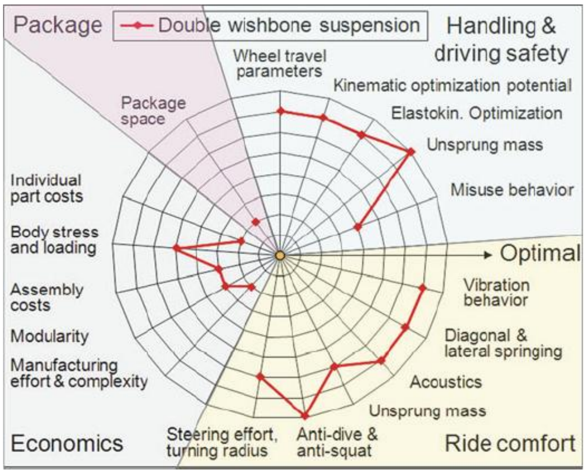

Figure 4: Performance profile-double wishbone suspension system [4]

The different components of a double wishbone suspension and their roles are as follows:

- **Spring & Shock Absorber:** Together, they support the weight of the vehicle and enable the control arms and wheels to move up and down. The primary function of the shock absorber is to dampen the body and wheel vibrations caused by uneven roads, while the coil spring cushions these vibrations to provide a comfortable ride. Without the shock absorber, the vehicle would continue to bounce after encountering an uneven road surface.
- **Anti-Roll or Stabilizer Bar:** This component limits the vehicle's body roll during cornering, maintaining constant wheel contact with the road. It connects the lower control arms on both sides of the vehicle through bar links and bushings, reducing excessive body lean or roll by resisting the centrifugal forces experienced during cornering [9].
- **Mechanism**: This specifies the kinematics of pivot points during lateral and vertical movement, controlling the suspension geometry. The mechanism includes various other components such as:

A. **Control Arms**: These define the wheel kinematics relative to the chassis. The outer end of the control arm contains a ball joint linked to the steering knuckle, while the inner end consists of a rubber bushing as shown in figure 5.

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Figure 5: Upper & lower control arms.

B. **Steering Knuckle:** As shown in figure 6, this component accommodates the installation of wheels, braking, and steering elements. It features three pivot points that connect to the upper and lower control arms and the steering rod. When the steering is turned, it rotates the knuckle, which subsequently turns the wheel assembly.

Figure 6: Upper & lower control arms.

C. **Ball Joints:** These components link the steering knuckle with the control arms, allowing freedom of movement in two translational directions and one rotational direction as illustrated in figure 7. The ball joints in the control arms have a spring mounted on them, making them load carriers.

Figure 7: Ball joint [31]

D. **Bushings:** These supplementary units absorb and isolate vibrations and noise, enhancing the wear resistance of components. Control arm bushings comprise rubber sandwiched between a metal inner sleeve and a metal outer sleeve. The inner sleeve remains stationary, while the outer sleeve moves with the control arm as displayed in figure 8. Bushings are also used in anti-roll bars, shock absorbers, and strut rods.

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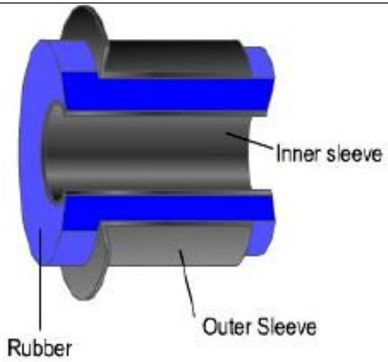

Figure 8: Bushing [31]

E. **Strut Rods:** Strut rods prevent the lower control arm from moving fore and aft, providing stability. They are connected to the frame and bolted to the outer end of the lower control arms. The installation of strut rods typically depends on the drive configuration of the vehicle, whether it is front-wheel drive (FWD) or rear-wheel drive (RWD). In Battery Electric Vehicle (BEV), strut rods are installed at the rear since it is a RWD car [10].

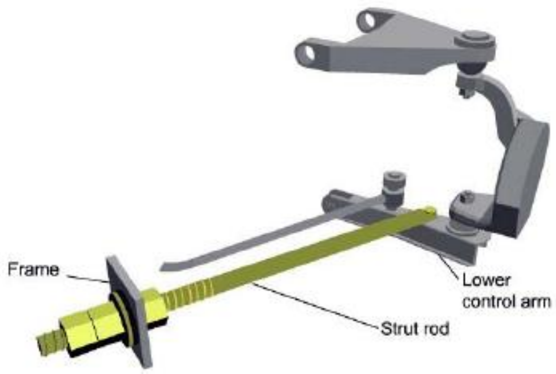

Figure 9: Strut rod [31]

III. LOTUS SUSPENSION ANALYSIS (LSA)

LSA serves as a design and analysis tool, employed for the initial layout of vehicle suspension hard points and the design and orientation of suspension bushes for tuning compliant behavior. Various results, such as camber angle and toe angle, were graphically displayed against bump motion, roll motion, or steering motion using this software for analysis, aiding in the confirmation of the kinematic behavior. The LSA software was employed to analyze and achieve the required kinematic behavior of the suspension. Various suspension parameters, such as camber angle and toe angle, were displayed graphically against bump, roll, and steering motion. The simulation results are approximations due to certain software limitations. Initial hard points of the front and rear suspension were obtained using a 2D CAD drawing of the Honda S2000 suspension. Using this as a reference, the coordinates of the hard points were scaled into the LSA software according to its coordinate system. Certain parameters of the front and rear suspension, such as C.G., sprung mass, bump travel, and rebound travel, were defined. Parameters like C.G. and sprung mass were

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determined collaboratively, while bump and rebound travel measurements were obtained by measuring and averaging actual bumps and potholes on the road.

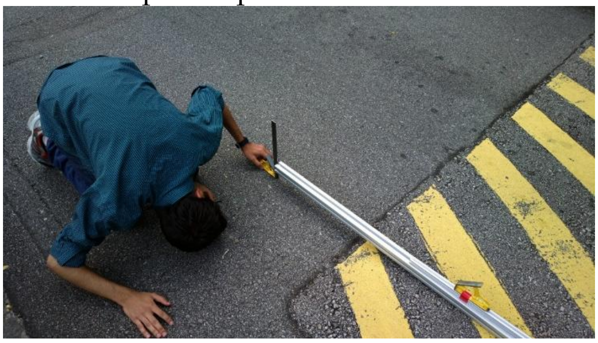

Figure 10: Measuring bump height

A. Front Suspension Analysis

The simulation for the front suspension analysis was set up by defining all the parameters as depicted in figure 11. Simulations were conducted for three different scenarios: bump, roll, and steer motion. Variations in different suspension parameters were graphically displayed on x-y plots. The points that coincide with the vertical y-axis of the graph represent the original values of those respective suspension parameters.

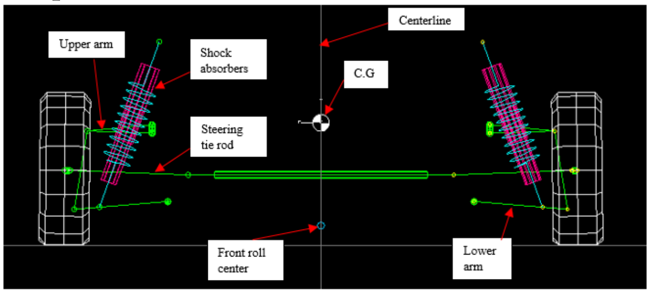

Figure 11: Front suspension setup

Simulation results of variation in front suspension parameters when the car goes over a bump are displayed graphically below in figure 12.

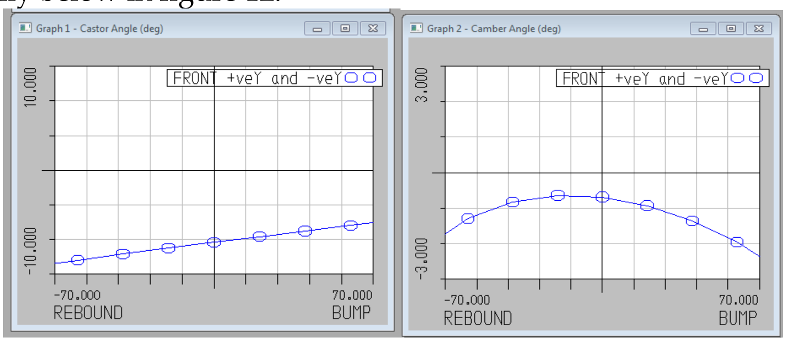

Figure 12: Castor v/s wheel travel (left) & Camber v/s wheel travel (right)

Figure 12 (left) illustrates that when the car goes over a bump, the wheel travels upwards, and the caster angle between the vertical tire line and the imaginary line through two pivots of the control arms increases and vice versa. On a flat surface or when stationary, the caster angle was found to be 6.93°. Figure 12 (right) shows that the camber angle decreases further during bump travel,

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remains constant up to a certain point during rebound travel, and then decreases. The original camber value on a flat surface was found to be -0.69°.

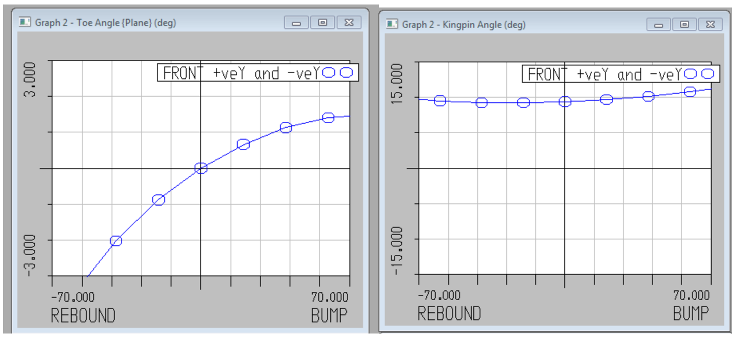

Figure: 13: Toe v/s wheel travel (left) & King-pin v/s wheel travel (right)

Figure 13 (left) shows that the toe-out angle increases with positive wheel travel, while the toe-in angle increases with negative wheel travel. The front wheels originally have a toe-in angle of 0° when the car travels on a flat road. In Figure 13 (right), it can be observed that the variation in king-pin inclination is minimal with respect to bump and rebound travel of the wheel. The original value of king-pin inclination was found to be 9.41°.

Simulation results of variation in front suspension parameters when car rolls while going around a corner are displayed graphically below in figure 14.

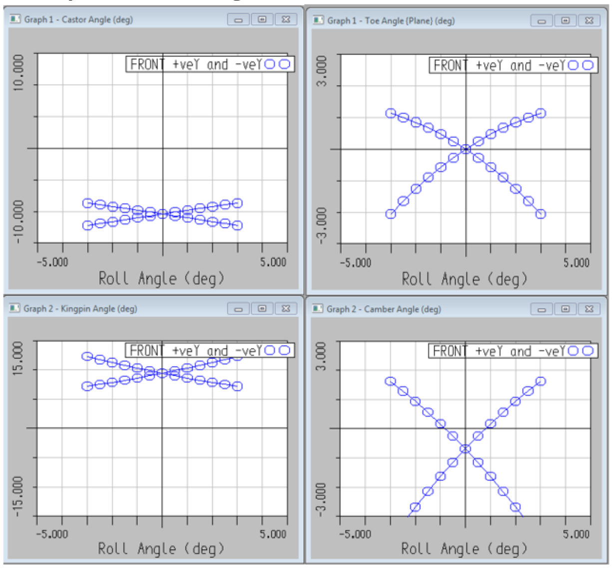

Figure 14: (Clockwise) Caster, toe, king-pin, camber angle v/s roll angle.

Variations in caster, toe, camber, and king-pin inclination were found to be symmetric about the x-y axis with respect to roll angle variations. This indicates that when the car rolled to one side, the changes in all parameters of both wheels were equal and opposite. The original values of caster, camber, toe-in, and king-pin inclination when the car is not rolling were determined to be 6.93°, -0.69°, 0°, and 9.41°, respectively.

Simulation results of variation in front suspension parameters when car is steered while being stationary are displayed graphically below in figure 15.

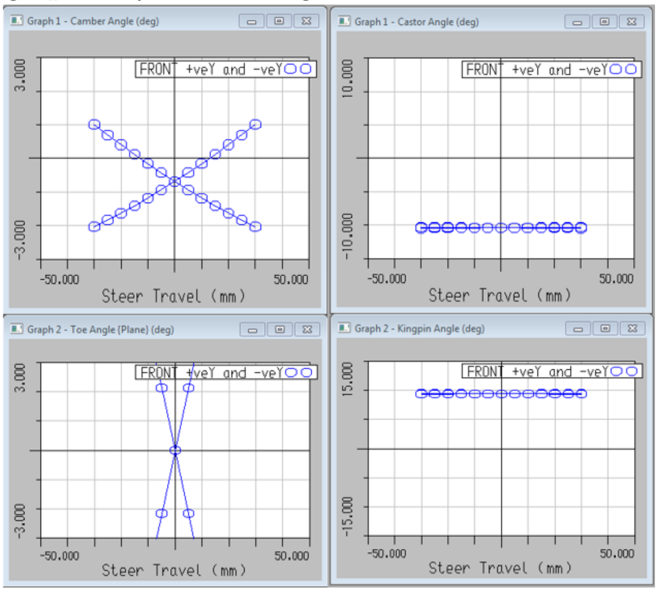

Figure 15: (Clockwise) Camber, castor, king-pin and toe angle v/s steer travel.

As illustrated in figure 15, the caster angle and king-pin inclination remain unchanged when the car is steered to the right or left. In contrast, variations in the camber angle and toe angle are symmetric about the x-y axis, indicating that changes in the parameters of each wheel are equal and opposite when the car is steered to one side. The original values of the caster and camber angles differ slightly from the actual values due to the limitations of the LSA software.

B. Rear Suspension Analysis

The simulation for the rear suspension analysis was set up by defining all the parameters as depicted in figure 16. Simulations were conducted for two different scenarios: bump and roll. Variations in different suspension parameters were graphically displayed on x-y plots. Points that coincide with the vertical y-axis represent the static values of the respective suspension parameters.

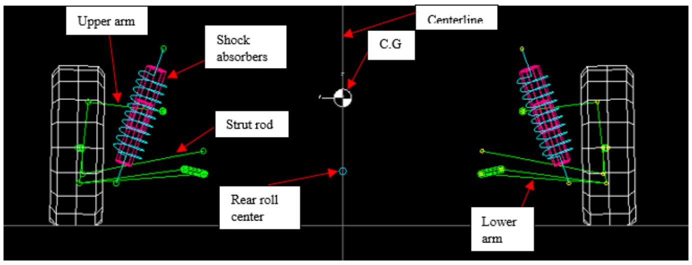

Figure 16: Rear suspension setup

Simulation results of variation in rear suspension parameters when the car goes over a bump are displayed graphically below in figure 17.

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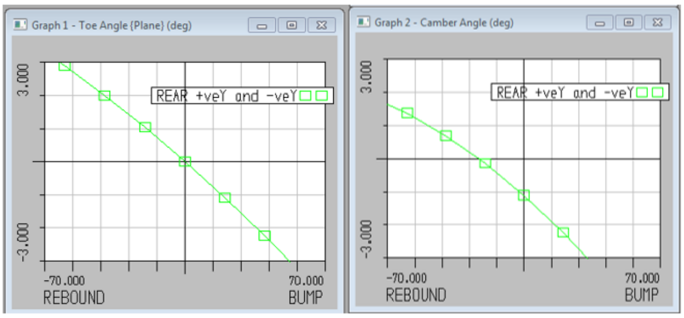

Figure 17: Toe angle v/s wheel travel (left) & camber angle v/s wheel travel (right)

Figure 17 illustrates that the toe angle and camber angle decrease with bump travel, while both increase with rebound travel of the wheels. The original values of the toe angle and camber angle were found to be 0° and -1.11°, respectively.

Simulation results of variation in rear suspension parameters when the car rolls while going over a corner are displayed graphically below in figure 18.

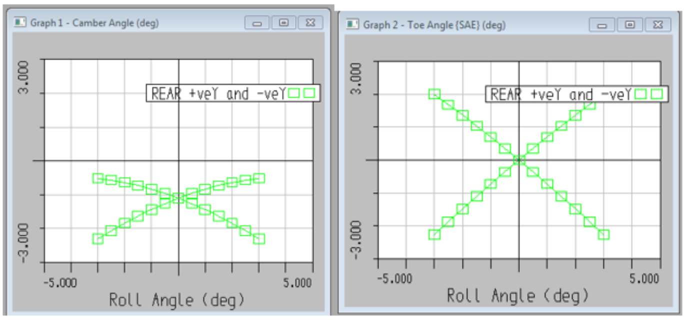

Figure 18: Camber angle v/s roll angle (left) and toe angle v/s rolls angle (right)

Variations in camber angle and toe angle with respect to roll angle were found to be symmetric when the car experiences roll, as shown in figure 18. The original values of the camber angle and toe angle were -1.11° and 0°, respectively. The significant difference between the actual camber angle and actual toe-in angle compared to LSA's original camber and toe angle was attributed to the absence of the strut rods option in the setup during the simulation of the rear suspension.

C. Suspension Force Analysis

The maximum design loads experienced by the front and rear tires in different modes of car use were evaluated. From there, the loads acting on the various suspension components were calculated, and the loads being transferred to the chassis were determined. In this analysis, the maximum loads at the tire patch during different modes of use, such as braking, cornering, rollover, and vertical bump, were first evaluated as shown in figure 19.

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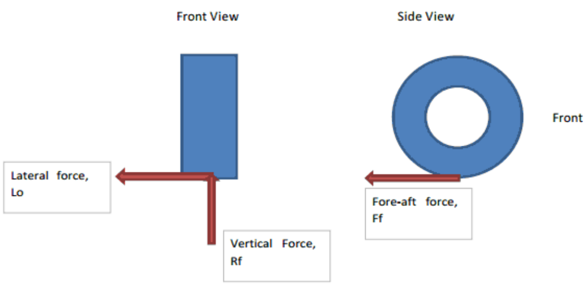

Figure 19: Different loads acting on the tire patch.

Subsequently, using equations of static equilibrium, the maximum loads at the body mounting points were evaluated. Understanding the forces acting on the various suspension system components was crucial for identifying hard-points and designing the main chassis and roll-cage of the BEV.

The different modes of use are as follows:

1. **Vertical bump load**: The vertical dynamic load experienced by the tires when the car traverses over a bump.

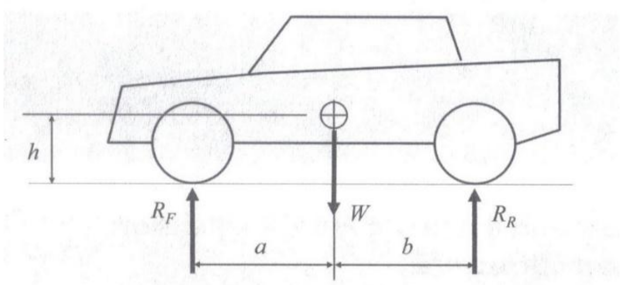

Figure 20: Vertical loads [22]

As depicted in figure 20, the vehicle is at rest, and the front and rear patch tire loads (Rf and Rr, respectively) were determined using static equilibrium and equations 1 and 2 [22].

$$R_f = \frac{b}{(a+b)}W \quad (Eq.1) \qquad R_r = \frac{a}{(a+b)}W \quad (Eq.2)$$

where:

- W is the weight of the car
- Rf & Rr are the vertical loads at the front and rear tires, respectively
- a and b are the distances of the center of gravity (C.G) from the front and rear axles, respectively

These are the static loads, to take dynamic loading effects of bump into account static loads were multiplied by design load factor r =3. [i.e. (Dynamic design load) = 3(Static design load)]. This would give us the vertical bump loads acting at the front & rear tire patches. Thus, the dynamic design load is three times the static load, giving us the vertical bump loads acting on the front and rear tire patches.

2. **Braking:** The fore-aft load at the front and rear tire patches, resulting from steady-state braking

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deceleration of n times the acceleration due to gravity, was calculated using equations 3 and 4 [22].

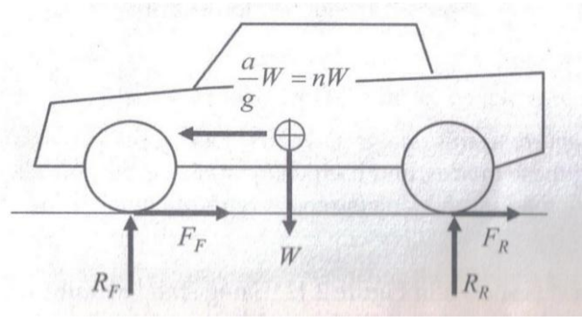

Figure 21: Fore-aft loads [22]

$$F_f = \mu \frac{(b+nh)}{(a+b)} \frac{W}{2} \quad (Eq.3) \qquad F_r = \mu \frac{(a+nh)}{(a+b)} \frac{W}{2} \quad (Eq.4)$$

Where, μ: coefficient of friction between tire and road
W: weight of the car
h: height of C.G above the ground
n: braking acceleration in g's
Ff & Fr: Fore-aft forces at front and rear tire patches
a,b: distance of C.G from front and rear axle

3. **Cornering:** The lateral load acting on the outer tire patch during steady-state lateral cornering acceleration was calculated. The lateral load on the front and rear outer tire patches was determined using formulas 5 and 6, respectively [22].

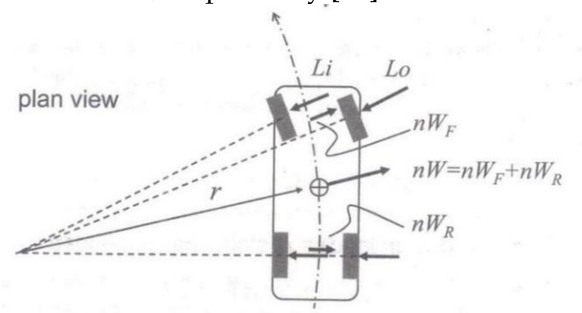

Figure 22: Lateral loads [22]

$$L_o = \mu\left(\frac{1}{2} + \frac{nh}{t}\right) R_f \quad (Eq.5) \qquad L_o = \mu\left(\frac{1}{2} + \frac{nh}{t}\right) R_r \quad (Eq.6)$$

Where, t: track width
h: height of C.G above the ground
Lo: lateral load on outer tire patch
n: braking acceleration in g's
μ: coefficient of friction between tire and road
Rf & Rr: Vertical load at front and rear tires

4. **Rollover:** Rollover occurs during high-speed cornering when the steady-state resultant acceleration at the center of gravity shifts outside the vehicle's base. This phenomenon occurs when the lateral acceleration in g's reaches n = t/(2h). The lateral loads at the front and rear tire patches during rollover were calculated using equations 7 and 8 [22].

$$L_o = \frac{2h}{t} R_f \quad (Eq. 7) \qquad L_o = \frac{2h}{t} R_r \quad (Eq. 8)$$

Where, Lo: lateral load on outer tire patch
t: track width
h: height of C.G above the ground
Rf & Rr: Vertical load at front and rear tires

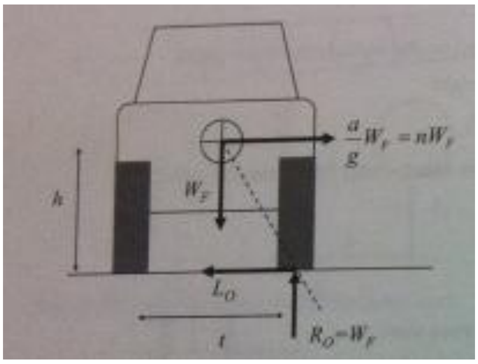

Figure 23: Rollover loads [22]

The maximum tire design loads for these modes were summarized using tables 1 and 2.

Table 1: Front tire patch loads.

Mode	**Lateral (N)**	**Fore-aft (N)**	**Vertical (N)**
Static	-	-	3531.6
Braking	-	4734.96	4734.96
Cornering	5276.06	-	5276.06
Incipient Rollover	3488.93	-	7063.2
Maximum Design load per tire	**5276.06**	**4734.96**	**10594.8**

Table 2: Rear tire patch loads.

Mode	**Lateral (N)**	**Fore-aft (N)**	**Vertical (N)**
Static	-	-	4316.4
Braking	-	5519.76	5519.76

Cornering	5289.62	-	6366
Rollover	4099.18	-	8632.8
Maximum design load per tire	**5289.62**	**5519.76**	**12949.2**

IV. FUTURE WORK

This research highlights the importance of using advanced design tools and simulations to optimize vehicle suspension systems, which has significant implications for the future of transportation. As electric vehicles (EVs) become more prevalent, understanding and refining suspension parameters will be crucial for enhancing their performance, stability, and handling.

By adopting EVs, we can significantly reduce environmental pollution and combat global warming, as EVs centralize pollution to power stations rather than distributing it across individual vehicles. This research supports the development of more efficient and reliable EVs, which can lead to broader acceptance and adoption of sustainable transportation solutions.

Furthermore, the research conducted showcases the potential for innovation in the field of electric vehicles, encouraging more institutions and automakers to invest in similar projects. As advancements in EV technology continue, we can expect improved range, lower operating costs, and an eco-friendlier transportation system for the future.

V. CONCLUSIONS

This research demonstrates the pivotal role of advanced design and analysis tools like LSA in optimizing vehicle suspension systems. Through detailed simulations and analysis, key parameters such as camber angle, toe angle, caster angle, and king-pin inclination were meticulously evaluated for various scenarios including bump, roll, steering, braking, cornering, and rollover. The findings highlight the dynamic interactions of suspension components and their impact on vehicle performance. The challenges of environmental pollution and global warming underscore the urgency for alternative mobility solutions like electric vehicles (EVs). The Battery Electric Vehicle (BEV) exemplifies innovation in sustainable transportation, leveraging compact geography. This research's in-depth analysis of suspension parameters contributes to the broader goal of advancing EV technology, ensuring enhanced handling, stability, and performance while aligning with ecological imperatives.

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