

Research Article

Mathematical Modeling and Comparative Analysis of Stability in Internal Combustion Engine and Electric Vehicles

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Abstract:

With the increasing adoption of electric vehicles (EVs), understanding their stability characteristics compared to internal combustion engine (ICE) vehicles is crucial. This study investigates the impact of suspension tuning strategies on the stability of EVs, with a comparative reference to ICE vehicles. The analysis explores how mass distribution and control methods influence ride comfort and handling performance. A mathematical modeling approach is employed to examine how differences in mass distribution and suspension parameters affect vehicle stability. The results indicate that EVs experience lower ride comfort due to increased unsprung mass but exhibit improved road-holding capabilities. While EVs demonstrate greater roll stability at low frequencies due to a lower center of gravity, their yaw stability is less favorable, with increased deviations from the intended trajectory. To mitigate these drawbacks, a skyhook-damped semi-active suspension control strategy was implemented, improving ride comfort and handling stability. This study employs average parameters derived from 13 prior works to construct representative models of ICE and EV vehicles. As such, the findings highlight general trends rather than the behavior of any specific vehicle model. While the modeling approach effectively captures fundamental stability characteristics, further research incorporating nonlinear vehicle dynamics models and real-world experimental validation is recommended to enhance the applicability of these findings. Note that this study does not compare all possible ICE or EV configurations and is based on typical passenger car data; specialized ICE layouts such as boxer engines are not considered.

Keywords: Suspension, Rollover, Yaw stability, Frequency response, Vehicle control, Vehicle dynamics

1. Introduction

The rising popularity of electric vehicles is reshaping the transportation sector into a whole new landscape. As customers embrace the shift toward electric vehicles (EVs) over traditional internal combustion engine (ICE) vehicles [1], EVs are continually driving the world toward sustainability and gradually reducing environmental problems. However, EVs still have concerns in terms of stability. A major change that makes EVs different from conventional cars is the huge battery pack lying under the cabin. Since this battery is responsible for powering the vehicle for many hundreds of kilometers, it is remarkably heavy; some models can weigh as much as 25% of the entire vehicle weight [2]. Compared to the fuel tank of ICE vehicles, this significant increase in weight directly impacts vehicle stability performance.

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The changes in components in EVs are not limited to the battery but extend to the entire powertrain system. These alterations can lead to differences in mass, a lower center of mass compared to ICE vehicles, a shift in the location of the center of mass, and more. These parameters are the focus of this study.

Automotive suspension is a set of mechanical elements that link the vehicle body to the wheels. The suspension contains an elastic element (typically a coil spring) and a damping element (typically a hydraulic shock absorber) [3]. The suspension allows for relative up-and-down displacement between the vehicle body and the wheels [4], and its primary functions are to provide vertical compliance so the wheels can follow uneven road surfaces and to isolate the chassis from road roughness [5, 6]. EV suspensions come in two main types: centralized motor-driven EVs and in-wheel motor (IWM) driven EVs. Centralized motor-driven EVs are quite similar to ICE vehicles, with the main difference being that the driving power is provided by an electric motor, whereas IWM EVs relocate the motor to the wheel [7]. Previous literature has stated that the in-wheel electric motor (IWM) offers numerous advantages to the EV, such as reducing the total vehicle weight and providing more precise and accurate torque. However, this mechanism also increases the unsprung mass of the vehicle, which can negatively impact ride comfort and road holding [8, 9]. This research addresses this concern by conducting a simulation to compare the effects of increased unsprung mass in EVs.

Roll stability refers to how well a vehicle can resist rolling when cornering. If the vehicle experiences too much rolling, it can cause a rollover [10]. Among the dynamic maneuvers a vehicle can experience, rollovers are one of the most serious and threatening to the occupants [5]. There is a significant amount of research on rollover prevention, such as differential braking that applies uniform braking force to all the car's wheels [11], and active steering, which involves the use of advanced steering technology and sensors to detect potential rollover situations and adjust the steering automatically [12]. However, these studies did not compare the roll-resisting performance of ICE vehicles and EVs. Seongjin Yim [13] mentioned that the center of gravity (c.g.) is one of the most important variables that play a critical role in causing rollovers. Hence, this research will focus on comparing the effect of c.g. differences between ICE vehicles and EVs.

Lane departures can be a major cause of accidents, according to road crash analyses. In the United States, lane departures were identified as a factor in 66% of crashes involving a single lightweight vehicle [14, 15]. Three types of lateral systems have been developed in the automotive industry to address lane departure accidents: lane departure warning systems (LDWS), lane-keeping systems (LKS), and yaw stability control systems (YSC). This research paper focuses only on the yaw stability aspect. Yaw movements are initially generated by steering the vehicle, while changes in road friction coefficient can cause the vehicle to deviate from the original trajectory predicted by the driver, potentially contributing to lane departure accidents [10]. Many researchers have proposed that an electric vehicle with in-wheel motors can achieve more efficient yaw moment compensation [16, 17, 18]. Certainly, yaw controls can enhance the steering stability of a vehicle; however, the performance of the control approaches can alter gradually because of vehicle parameters and disturbances. A comparison of vehicle parameters is then employed in this study to determine how differently ICEs and EVs behave when the vehicle experiences unexpected yaw.

In this paper, three key aspects of vehicle dynamics and stability are analyzed: suspension characteristics, roll stability, and yaw stability. The study begins with an overview of the mathematical modeling approach, followed by the governing equations, transfer functions, and state-space representations used in the simulations. The results are presented using Bode plots to illustrate frequency-domain responses, along with discussions on their implications. A comparative analysis of EVs and ICE vehicles is conducted, highlighting key differences in stability and control. The final section summarizes the advantages and limitations of EVs in terms of stability and explores potential suspension control strategies to enhance their performance. This study provides valuable insights into vehicle dynamics and contributes to the development of more effective suspension tuning and control strategies for EVs.

While numerous studies have investigated either suspension performance, rollover risk, or yaw stability in EVs and ICE vehicles separately, few works provide a unified comparison using consistent modeling assumptions. This study addresses that gap by constructing representative models based on averaged parameters from 13 prior works, enabling a structured comparison under identical simulation conditions. The results aim to reveal key trends in ride comfort, handling, and stability between the two vehicle types. Therefore, the objective of this study is to investigate and compare three aspects of vehicle stability—suspension characteristics, roll stability, and yaw stability—between ICE and EV vehicles, using representative models constructed from averaged parameters.

2. Materials and methods

2.1 EV and ICE Vehicle Parameters

The transformation from ICE to a full EV involves significant changes in components. In ICE vehicles, the primary components include an internal combustion engine, transmission, fuel system, exhaust system, and various auxiliary components like radiators and oil coolers. EVs, on the other hand, replace these with electric motors, battery packs, power electronics, and a simplified drivetrain. The electric motor and battery pack are the most critical changes, with the battery pack often being the heaviest component. These changes can be seen in Figure 1.

This shift results in a different mass distribution and a lower center of gravity in EVs compared to ICE vehicles. The battery pack is typically placed at the bottom of the vehicle, contributing to a more evenly distributed weight. In contrast, ICE vehicles have a higher center of gravity and a front-biased weight distribution due to the engine and other heavy components being positioned toward the front chassis. While the fundamental suspension architecture remains similar, certain modifications are necessary to accommodate these differences. Adjustments to parameters such as spring stiffness, damper coefficients, and tire characteristics are often required to optimize ride comfort, handling, and stability in EVs.

Modern passenger car suspensions combine a shock absorber and a spring into a single unit connected to the wheel hub. EV suspensions have a similar setup. However, parameters like spring stiffness, damper coefficient, and tire stiffness can be tuned to different values for these two types of vehicles. Although the simulation appears to be a car-to-car comparison, EVs have key parameters that differ from ICE vehicles. For example, a lower center of gravity is unique to EVs, as the battery mounted under the chassis significantly impacts the c.g. location. Suspension parameters can be tuned to match any vehicle mass to achieve the best outcome.

To construct representative models for comparison, average parameters from 13 different studies [19–31] were used to characterize ICE and EV vehicles. These represent typical passenger cars rather than any specific model. Table 1 presents the parameters of EVs and ICE vehicles used in this study, and Figure 2 presents the schematic diagram of the vehicle's parameters. These parameters were gathered from 13 different studies on related references [19-31], each representing a passenger car with a weight range of 968 kg to 1608 kg. As predicted, the average weight parameters of EVs were found to be higher than those of ICE vehicles. This can be observed in the mean sprung and unsprung masses shown in the table. Sprung mass refers to a quarter of the vehicle weight that is supported by the suspension, and unsprung mass refers to a quarter of the suspension component weight, including tire weight. The standard deviation (SD) of the sprung and un-sprung masses for ICE vehicles is 51.4 kg and 9.1 kg, respectively, while for EVs, it is 51.1 kg and 19.2 kg, respectively. Other suspension parameters, such as suspension stiffness, suspension damping coefficient, and tire stiffness, were also affected by the mass differences. The same impact is observed in the roll and yaw motion parameters.

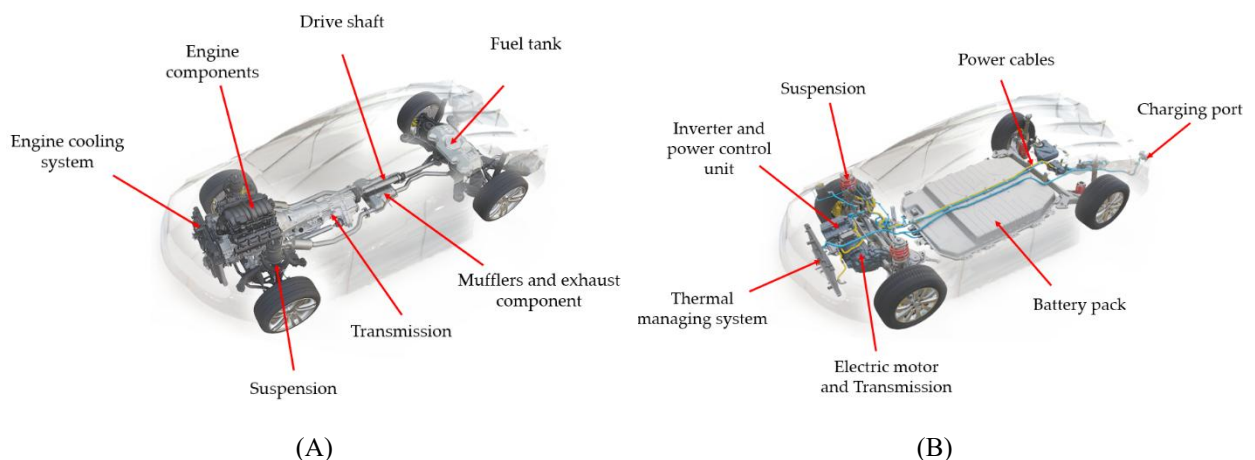
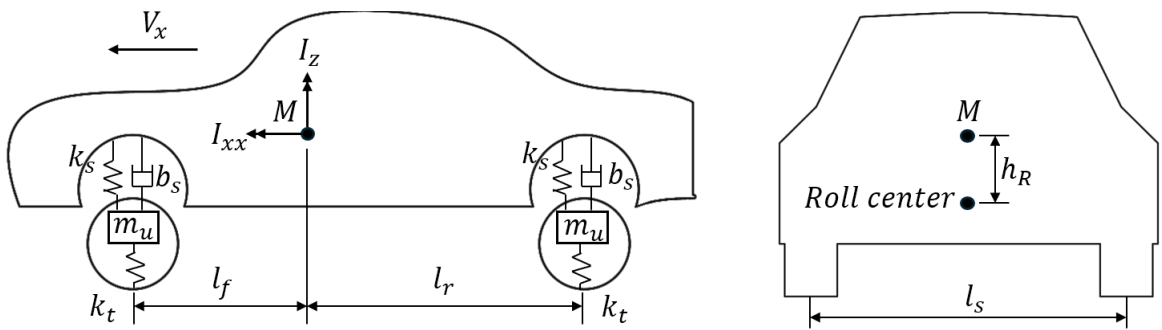


Fig. 1. Inside vehicle components of an ICE (A) and EV (B).

Table 1: ICE and EV parameters. [19-31]

Parameters	Value (ICE)	Value (EV)	Unit
Sprung mass (m_s)	288	295.8	kg
Unsprung mass (m_u)	43.8	57.5	kg
Vehicle mass (M)	1327.2	1413.2	kg
Suspension stiffness (k_s)	20,000	24,000	N/m
Suspension damping coefficient (b_s)	1,550	2,000	Ns/m
Tire stiffness (k_t)	180,000	220,000	N/m
Distance between left and right suspension (l_s)	1.5	1.5	m
Height of the center of gravity from the roll center (h_R)	1	0.504	m
Principal roll moment of inertia (I_{xx})	600	728.5	kg/m ²
Moment of inertia of the vehicle in the z-axis (I_z)	1,550	1,550	kg/m ²
Distance between center of mass and front axle (ℓ_f)	1.04	1.41	m
Distance between center of mass and rear axle (ℓ_r)	1.56	1.20	m
Front cornering stiffness ($C_{\alpha f}$)	35,000	35,000	N/rad
Rear cornering stiffness ($C_{\alpha r}$)	35,000	35,000	N/rad
Longitudinal velocity of the vehicle (V_x)	30	30	m/s

**Fig. 2.** Schematic diagram showing vehicle's parameters.

2.2 Vehicle suspension

As mentioned in Section 1, the suspension system in a vehicle is a critical component designed to maximize the friction between the tires and the road surface, ensuring stability, road holding, and passenger comfort. It consists of a combination of springs, dampers, and linkages that connect the vehicle to its wheels. Vehicle suspension can be categorized into three main types: passive suspension, semi-active suspension, and active suspension systems. In this paper, the uses of passive and semi-active suspension will be further discussed in this section.

A passive suspension system features components like springs and dampers with fixed characteristics. These characteristics are determined by the suspension designer based on the design goals and the intended application. Typically, the parameters of a passive suspension are set to achieve a balance between minimizing vibrations and enhancing road grip [6].

The first goal of this study was to investigate the effects of the weight differences between ICEs and EVs on passive suspension. In this case, a two-degree-of-freedom quarter-car model, shown in Figure 3, will represent the movement of a car with a suspension system in the vertical direction. As seen in the figure, the sprung mass m_s represents a quarter of the total vehicle mass supported by the suspension. The unsprung mass m_u represents the mass not supported by the suspension, such as the axle and tire. The suspension itself is shown to consist of a spring k_s , a damper b_s and an active force actuator F_a . The active force F_a can be canceled out in the case of passive suspension. The vertical stiffness of the tire is represented by the spring k_t . The variable z_s , z_u and z_r represent the vertical displacements from static equilibrium of the sprung mass, unsprung mass, and the road, respectively. The equations of motion for the quarter-car model shown in Figure 3 are given by:

$$m_s \ddot{z}'_s + b_s (\dot{z}'_s - \dot{z}'_u) + k_s (z_s - z_u) = F_a \quad (1)$$

$$m_u \ddot{z}_u + k_t (z_u - z_r) - b_s (\dot{z}_s - \dot{z}_u) - k_s (z_s - z_u) = -F_a \quad (2)$$

To qualify the overall performance, three aspects of the suspension will be analyzed in terms of transfer functions (TF): sprung mass acceleration TF, suspension deflection TF, and tire deflection TF. Sprung mass acceleration TF refers to the acceleration experienced by the part of a vehicle that is supported by the suspension system, which typically includes the vehicle's body, cabin, and any cargo or passengers it carries. This transfer function is used to quantify ride quality. Suspension deflection TF determines the rattle space in the vehicle; a small value of suspension deflection provides good support for the vehicle's static weight and can quantify suspension lifetime. The road-holding performance of a vehicle depends on vertical tire deflection TF [10].

The state equations of a quarter-car suspension can be written as:

$$\dot{x} = Ax + B_1 F_a + B_2 \dot{z}_r,$$

$$\begin{bmatrix} \dot{x}_1 \\ \dot{x}_2 \\ \dot{x}_3 \\ \dot{x}_4 \end{bmatrix} = \begin{bmatrix} 0 & 1 & 0 & -1 \\ k_s & -b_s & 0 & b_s \\ -\frac{1}{m_s} & -\frac{1}{m_s} & 0 & \frac{1}{m_s} \\ 0 & 0 & 1 & 1 \\ k_s & -b_s & -k_s + k_t & -b_s \\ \frac{1}{m_u} & -\frac{1}{m_u} & -\frac{1}{m_u} & -\frac{1}{m_u} \end{bmatrix} \begin{bmatrix} x_1 \\ x_2 \\ x_3 \\ x_4 \end{bmatrix} + \begin{bmatrix} 0 \\ 1 \\ 0 \\ -1 \\ 0 \\ -1 \end{bmatrix} F_a + \begin{bmatrix} 0 \\ 0 \\ -1 \\ \frac{k_t}{m_u} \end{bmatrix} \dot{z}_r \quad (3)$$

$$y = Cx + D_1 F_a + D_2 \dot{z}_r,$$

$$y = \begin{bmatrix} x_1 \\ x_2 \\ x_3 \\ x_4 \end{bmatrix} = \begin{bmatrix} 1 & 0 & 0 & 0 \\ 0 & 1 & 0 & 0 \\ 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 1 \end{bmatrix} \begin{bmatrix} x_1 \\ x_2 \\ x_3 \\ x_4 \end{bmatrix} + \begin{bmatrix} 0 \\ 0 \\ 0 \\ 0 \end{bmatrix} F_a + \begin{bmatrix} 0 \\ 0 \\ 0 \\ 0 \end{bmatrix} \dot{z}_r \quad (4)$$

where:

$x_1 = z_s - z_u$ is the suspension deflection

$x_2 = \dot{z}_s$ is the absolute velocity of sprung mass

$x_3 = z_u - z_r$ is the tire deflection

$x_4 = \dot{z}_u$ is the absolute velocity of unsprung mass.

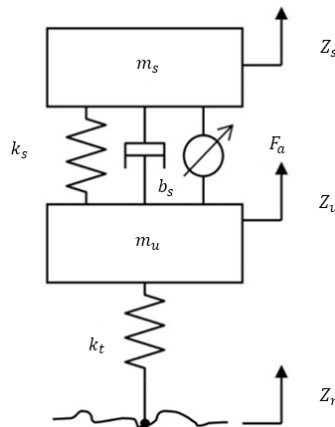


Fig. 3. Quarter car model.

The following three transfer functions are then derived from state equations (3-4) and will be used to determine suspension performances through bode plot.

a) Acceleration transfer function (Effect on ride comfort)

$$H_A(s) = \frac{\ddot{z}_s(s)}{\dot{z}_r(s)} = \frac{s \dot{z}_s(s)}{\dot{z}_r(s)} \quad (5)$$

b) Suspension deflection transfer function (Effect on suspension lifetime)

$$H_{SD}(s) = \frac{z_s(s) - z_u(s)}{\dot{z}_r(s)} \quad (6)$$

c) Tire deflection transfer function (Effect on road grip)

$$H_{TD}(s) = \frac{z_u(s) - z_r(s)}{\dot{z}_r(s)} \quad (7)$$

For a system described by state equations, the dynamic response can be analyzed by converting the system's state equations into a transfer function. This can be achieved using the Laplace Transform. Hence, the Laplace Transform of a state equation given by $\dot{x} = Ax + Bu$ and $y = Cx + Du$ become:

$$G(s) = \frac{Y(s)}{U(s)} = C(sI - A)^{-1}B + D \quad (8)$$

For simulation purposes, by substituting vehicle parameters in equation (8) the transfer functions derived in equations (5, 6, 7) can be further developed into the following forms:

a) For sprung mass acceleration transfer function. Let $C = [0 \ 1 \ 0 \ 0]$ and $D = 0$. Substituting the matrices A, B, C, D into equation (8), we obtain the relationship $G(s) = \frac{s\dot{z}_s(s)}{\dot{z}_r(s)}$. Therefore, it will be necessary to differentiate one more time to get $G(s) = H_A(s) = \frac{s\ddot{z}_s(s)}{\dot{z}_r(s)}$

$$H_A(s) = \frac{b_s k_t s^2 + k_s k_t s}{m_s m_u s^4 + (m_s b_s + m_u b_s) s^3 + (m_s k_s + m_s k_t + m_u k_s) s^2 + b_s k_t s + k_s k_t} \quad (9)$$

b) For suspension deflection transfer function. Let $C = [1 \ 0 \ 0 \ 0]$ and $D = 0$. Substituting the matrices A, B, C, D into equation (8), we obtain the relationship $H_{SD}(s) = \frac{z_s(s) - z_u(s)}{\dot{z}_r(s)}$

$$H_{SD}(s) = \frac{-m_s k_t s}{m_s m_u s^4 + (m_s b_s + m_u b_s) s^3 + (m_s k_s + m_s k_t + m_u k_s) s^2 + b_s k_t s + k_s k_t} \quad (10)$$

c) For tire deflection transfer function. Let $C = [0 \ 0 \ 1 \ 0]$ and $D = 0$. Substituting the matrices A, B, C, D into equation (8), we obtain the relationship $H_{TD}(s) = \frac{z_u(s) - z_r(s)}{\dot{z}_r(s)}$

$$H_{TD}(s) = \frac{-m_s m_u s^3 - (m_s b_s + m_u b_s) s^2 - (m_s k_s + m_u k_s) s}{m_s m_u s^4 + (m_s b_s + m_u b_s) s^3 + (m_s k_s + m_s k_t + m_u k_s) s^2 + b_s k_t s + k_s k_t} \quad (11)$$

2.2.1 Semi-Active suspension

Automotive suspension can be further improved by active suspension, which is a suspension system that uses electronic sensors and hydraulic or pneumatic actuators to actively control the vehicle's suspension movements. It adjusts the damping and spring rates in real time to optimize ride comfort, stability, and handling. Ref. [10] also qualifies the performance of active suspension against the regular passive suspension and between ICEs and EVs. However, an active suspension setup has major drawbacks, including high cost, complexity, and frequent maintenance requirements. For this reason, semi-active suspension, a system that can adjust its damping characteristics in real time without the need for fully active control, offers a balanced solution. Semi-active suspension systems use adjustable dampers and an electronic control unit to provide many of the benefits of active suspension, such as improved ride comfort and handling, while being more cost-effective and requiring less maintenance. The quarter-car model representative of a semi-active suspension is similar to Figure 3, except that actuator F_a will change to semi-active damper b_{semi} as shown in Figure 4.

There are several control methodologies that can be implemented in a semi-active suspension. In this research, the focus is on a method called 'skyhook damped suspension.' It uses a control strategy where the vehicle body is considered to be 'hooked' to the sky, meaning it moves independently of the wheels' motions. In practical terms, this

system employs sensors to detect body movement and adjust the damping force in the suspension to minimize body motion relative to the road surface. To achieve a skyhook damped system, we constructed a feedback controller.

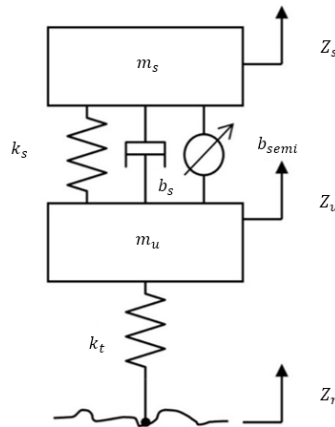


Fig. 4. Quarter-car semi-active suspension

Similar to the equations (1) and (2), semi-active suspension equations of motions can be expressed as:

$$m_s \ddot{z}_s + b_s(\dot{z}_s - \dot{z}_u) + k_s(z_s - z_u) = -b_{semi}(t)(\dot{z}_s - \dot{z}_u) \quad (12)$$

$$m_u \ddot{z}_u + b_s(\dot{z}_s - \dot{z}_u) + k_s(z_s - z_u) = -b_{semi}(t)(\dot{z}_s - \dot{z}_u) \quad (13)$$

where:

$$x_1 = z_s - z_u$$

$$x_2 = \dot{z}_s$$

$$x_3 = z_u - z_r$$

$$x_4 = \dot{z}_u$$

$$F_{semi} = -b_{semi}(\dot{z}_s - \dot{z}_u) = -b_{semi}(x_2 - x_4)$$

To validate the performance of a semi-active suspension, the desired semi-active force was defined as $F_a = -4000\dot{z}_s$. The semi-active damping coefficient is determined using:

$$b_{semi} = 0 \quad \text{If} \quad -F_a(x_2 - x_4) \leq 0$$

$$b_{semi} = -\frac{F_a}{x_2 - x_4} \quad \text{If} \quad 0 < -\frac{F_a}{x_2 - x_4} \leq b_{max}$$

$$b_{semi} = b_{max} \quad \text{If} \quad -\frac{F_a}{x_2 - x_4} > b_{max}$$

Feedback control is a control strategy used in control systems where the state variables of a system are measured and fed back to adjust the control input. This technique is particularly useful in achieving desired system behavior. In mathematical terms, consider a linear time-invariant (LTI) system described by the state-space equation $\dot{x} = Ax + B_1 F_a + B_2 \dot{z}_r$. In feedback control, the control input u , in this case F_a , is designed to be a linear combination of the state variables. This can be represented as $u = F_a = -Kx$ where K is the state feedback gain matrix. When this control law is applied to the system, the closed-loop dynamics become:

$$\dot{x} = (A - B_1 K)x + B_2 \dot{z}_r, \quad (14)$$

Here A and B_1 represent state matrix and input matrix derived from equation (3) where the gain matrix K is set to such $K = [0 \ 4000 \ 0 \ 0]$. Based on the closed-loop system in equation (14), the transfer functions for sprung mass acceleration, suspension deflection, and tire deflection can be computed using equation (8).

2.4 Yaw stability

The final goal of this research was to investigate the parameter that affects the yaw motion of a vehicle, in this case, the location of the center of gravity. ICE vehicles are known for having their engines located in either the front or rear engine bay, which makes the center of gravity lean toward the front or rear. Conversely, EV vehicles usually have their battery located under the cockpit, resulting in a center of gravity that might appear around the middle to rear of the vehicle. A 'bicycle' model of the vehicle with two degrees of freedom is considered, as shown in Figure 6. The two degrees of freedom are represented by the vehicle's lateral position y and the vehicle yaw angle Ψ . The vehicle's lateral position is measured along the lateral axis of the vehicle to point O, which is the center of rotation of the vehicle. The vehicle yaw angle Ψ is measured with respect to the global X-axis. The longitudinal velocity of the vehicle at the c.g. is denoted by V_x [10]. The front and rear cornering stiffness are represented by C_{af} and C_{ar} respectively. The variable ℓ_f is the distance between front wheel and the c.g. and the variable ℓ_r is the distance between rear wheel and the c.g. I_z is a moment of inertia of the vehicle in the z-axis. Lateral dynamic can be described as: [10]

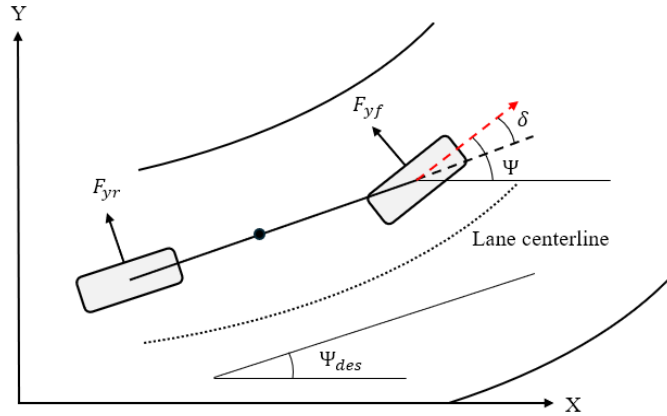


Fig. 6. Bicycle model of a vehicle lateral dynamic on a road.

$$\ddot{y} = \left(-\frac{2C_{af} + 2\ell_r C_{ar}}{mV_x} \right) \dot{y} + \left(-V_x - \frac{2\ell_f C_{af} - 2\ell_r C_{ar}}{mV_x} \right) \dot{\Psi} + \left(\frac{2C_{af}}{m} \right) \delta \quad (18)$$

$$\ddot{\Psi} = \left(-\frac{2\ell_f C_{af} - 2\ell_r C_{ar}}{I_z V_x} \right) \dot{y} + \left(-\frac{2\ell_f^2 C_{af} + 2\ell_r^2 C_{ar}}{I_z V_x} \right) \dot{\Psi} + \left(\frac{2\ell_f C_{af}}{I_z} \right) \delta \quad (19)$$

Hence, equation (18) and (19) can also be written in state space model as [10]:

$$\begin{bmatrix} \dot{y} \\ \ddot{y} \\ \dot{\Psi} \\ \ddot{\Psi} \end{bmatrix} = \begin{bmatrix} 0 & 1 & 0 & 0 \\ 0 & -\frac{2C_{af} + 2\ell_r C_{ar}}{mV_x} & 0 & -V_x - \frac{2C_{af}\ell_f - 2C_{ar}\ell_r}{mV_x} \\ 0 & 0 & 0 & 1 \\ 0 & -\frac{2\ell_f C_{af} - 2\ell_r C_{ar}}{I_z V_x} & 0 & -\frac{2\ell_f^2 C_{af} + 2\ell_r^2 C_{ar}}{I_z V_x} \end{bmatrix} \begin{bmatrix} y \\ \dot{y} \\ \Psi \\ \dot{\Psi} \end{bmatrix} + \begin{bmatrix} 0 \\ \frac{2C_{af}}{m} \\ 0 \\ \frac{2\ell_f C_{af}}{I_z} \end{bmatrix} \delta \quad (20)$$

The comparison of yaw stability in this paper was divided into two parts. The first part was the comparison of yaw rate. A yaw rate refers to how much the vehicle tends to yaw when cornering.

From equation (18) and (19), if $C_{af} = C_{ar} = C$, then:

$$\ddot{y} = \left(-\frac{4C}{mV_x} \right) \dot{y} + \left(-V_x - \frac{2\ell C(a-b)}{mV_x} \right) \dot{\Psi} + \left(\frac{2C}{m} \right) \delta \quad (21)$$

$$\ddot{\Psi} = \left(-\frac{2\ell C(a-b)}{I_z V_x} \right) \dot{y} + \left(-\frac{2\ell^2 C(a^2 + b^2)}{I_z V_x} \right) \dot{\Psi} + \left(\frac{2\ell C a}{I_z} \right) \delta \quad (22)$$

Use final value theorem or solve equation (21)-(22) with steady state yield:

$$\psi = \frac{2V_x C \delta}{-mV_x^2(a-b) + 2\ell C} \quad (23)$$

By using equation (23), we can derive a relationship of:

$$\frac{\psi_1}{\psi_2} = \frac{-mV_x^2(a_2 - b_2) + 2\ell C}{-mV_x^2(a_1 - b_1) + 2\ell C} \quad (24)$$

The second part of the yaw stability comparison is to assess the efficiency of yaw stability. The aim is to develop a lateral control system. It is suitable to employ a dynamic model that describes the state variables concerning the errors in position and orientation relative to the road. Consequently, the lateral model depicted in equation (20) will be redefined to incorporate the following error variables: e_1 , the distance of the c.g. of the vehicle from the center line of the lane and e_2 , the orientation error of the vehicle with respect to the road.

Consider a vehicle traveling with constant longitudinal velocity V_x on a road of constant radius R . The radius R is assumed to be large, allowing the rate of change of the vehicle's desired orientation to be defined as equation (25) [10]:

$$\dot{\psi}_{des} = \frac{V_x}{R} \quad (25)$$

Then, the desired acceleration of the vehicle can then be expressed as:

$$\frac{V_x^2}{R} = V_x \dot{\psi}_{des} \quad (26)$$

Define $\ddot{e}_1$ and $\ddot{e}_2$ as follows

$$\ddot{e}_1 = (\ddot{y} + V_x \dot{\psi}) - \frac{V_x^2}{R} = \ddot{y} + V_x(\dot{\psi} - \dot{\psi}_{des}) \quad (27)$$

and

$$\ddot{e}_2 = \ddot{\psi} - \ddot{\psi}_{des} \quad (28)$$

Define

$$\dot{e}_1 = \dot{y} + V_x(\dot{\psi} - \dot{\psi}_{des}) \quad (29)$$

The approach taken is to assume the longitudinal velocity V_x is constant. This assumption simplifies the vehicle's dynamics into a Linear Time Invariant (LTI) system. Hence, the state space model for error tracking is given by [10]:

$$\begin{bmatrix} \dot{e}_1 \\ \dot{e}_1 \\ \dot{e}_1 \\ \dot{e}_2 \end{bmatrix} = \begin{bmatrix} 0 & 1 & 0 & 0 \\ 0 & -\frac{2C_{af} + 2C_{ar}}{mV_x} & \frac{2C_{af} + 2C_{ar}}{m} & \frac{-2C_{af}\ell_f + 2C_{ar}\ell_r}{mV_x} \\ 0 & 0 & 0 & 1 \\ 0 & -\frac{2\ell_f C_{af} + 2\ell_r C_{ar}}{I_z V_x} & \frac{2\ell_f C_{af} + 2\ell_r C_{ar}}{I_z} & -\frac{2\ell_f^2 C_{af} + 2\ell_r^2 C_{ar}}{I_z V_x} \end{bmatrix} \begin{bmatrix} e_1 \\ \dot{e}_1 \\ e_2 \\ \dot{e}_2 \end{bmatrix} + \begin{bmatrix} 0 \\ \frac{2C_{af}}{m} \\ 0 \\ \frac{2\ell_f C_{af}}{I_z} \end{bmatrix} + \begin{bmatrix} 0 \\ -V_x - \frac{2C_{af}\ell_f + 2C_{ar}\ell_r}{mV_x} \\ 0 \\ \frac{2\ell_f^2 C_{af} + 2\ell_r^2 C_{ar}}{I_z V_x} \end{bmatrix} \dot{\psi}_{des} \quad (30)$$

*Noted that influence of road bank is excluded.

2.5 Overall methods

For suspension performance and roll stability comparison, the transfer functions (9, 10, 11, 17) can be graphically represented in a frequency response plot, also known as a Bode plot. This plot allows the comparison between ICE and EV by illustrating their dynamic behavior across a range of frequencies. Unlike suspension performance and roll stability, the yaw stability results are obtained by simulating a driving scenario, where the system response of equation (30) is compared in terms of driving paths and errors.

The mathematical modeling approach used in this study is based on well-established vehicle dynamics principles, which provide a fundamental framework for comparing ICE and EV stability. This approach enables systematic analysis of key stability factors, such as mass distribution and suspension response. However, it is acknowledged that this methodology has limitations in capturing certain nonlinear and transient effects in vehicle dynamics. While the linearized models used in this study provide valuable insights, future studies could incorporate nonlinear vehicle dynamics models and real-time experimental validation to further enhance the accuracy and applicability of the results.

3. Results

3.1 Suspension functions

The Bode plot shows how the amplitude of the output signal varies with frequency, indicating how the suspension system amplifies or attenuates vibrations at different frequencies. By plotting both ICE and EV responses on the same plot, these two can be evaluated by observing which one has a higher or lower gain value. Typically, higher gain values indicate a harsher response. For example, if the sprung mass acceleration response of the EV is higher than that of the ICE, it indicates that the EV experiences greater acceleration, which may suggest a rougher ride.

As seen in Figure 7-A, at low frequency, the EV sprung mass acceleration is approximately the same as the ICE. However, from Figures 7-B and 7-C, the EV deflection is lower than that of the ICE. Overall, at lower frequencies, the EV's performance is better than the ICE's. At high frequency, the response of suspension and tire deflection between ICE and EV are similar, but the acceleration of the EV is slightly higher than that of the ICE. This behavior suggests a harsher acceleration response. Although some differences between ICE and EV may seem minor, when converting these responses back to the time domain, the differences can be significant.

From the recent results, the sprung mass acceleration suspension performance of EVs still lags behind that of ICEs. Suggested methods to improve EV performance include adjusting the suspension parameters, such as suspension stiffness, suspension damping coefficient, and tire stiffness. The adjustment can be done by re-simulating the previous setup, trying to increase and decrease the parameters, and then observing the resulting trend. Such adjustments can provide a better understanding of the impacts of these parameters. For this reason, Figure 8 presents the effect of adjusting these values, respectively.

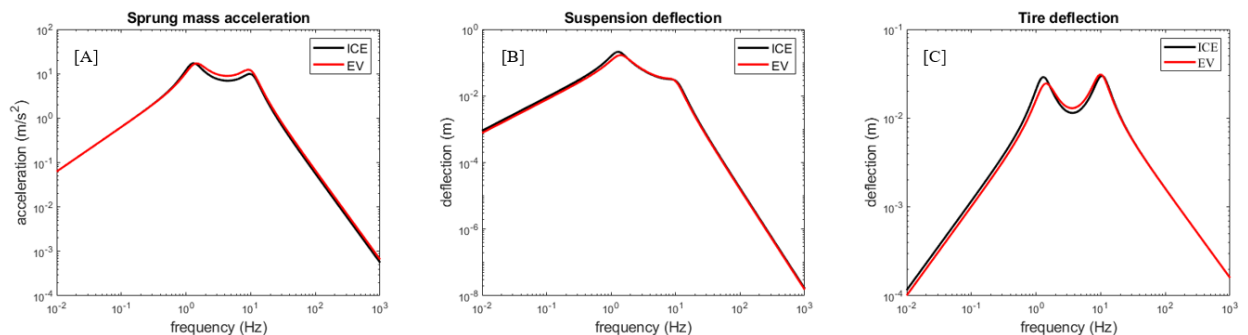


Fig. 7. Response of sprung mass acceleration (A), suspension deflection (B), tire deflection (C) between ICE and EV.

By examining Figure 8, adjusting k_s , b_s and k_t may improve some functions but worsen others. Starting from [1A-3A] decreasing k_s will favor the impact on sprung mass acceleration and tire deflection by reducing the resonant peak

at low frequency; however, this adjustment causes an increase in the range of deflection at low frequency for the suspension deflection aspect. Effects of adjusting k_s and the remaining b_s and k_t are further described in Table 2. These results explain the merits and demerits of varying these parameters. From the simulation, it was found that decreasing k_s , increasing b_s and decreasing k_t will provide more benefit to the suspension performance.

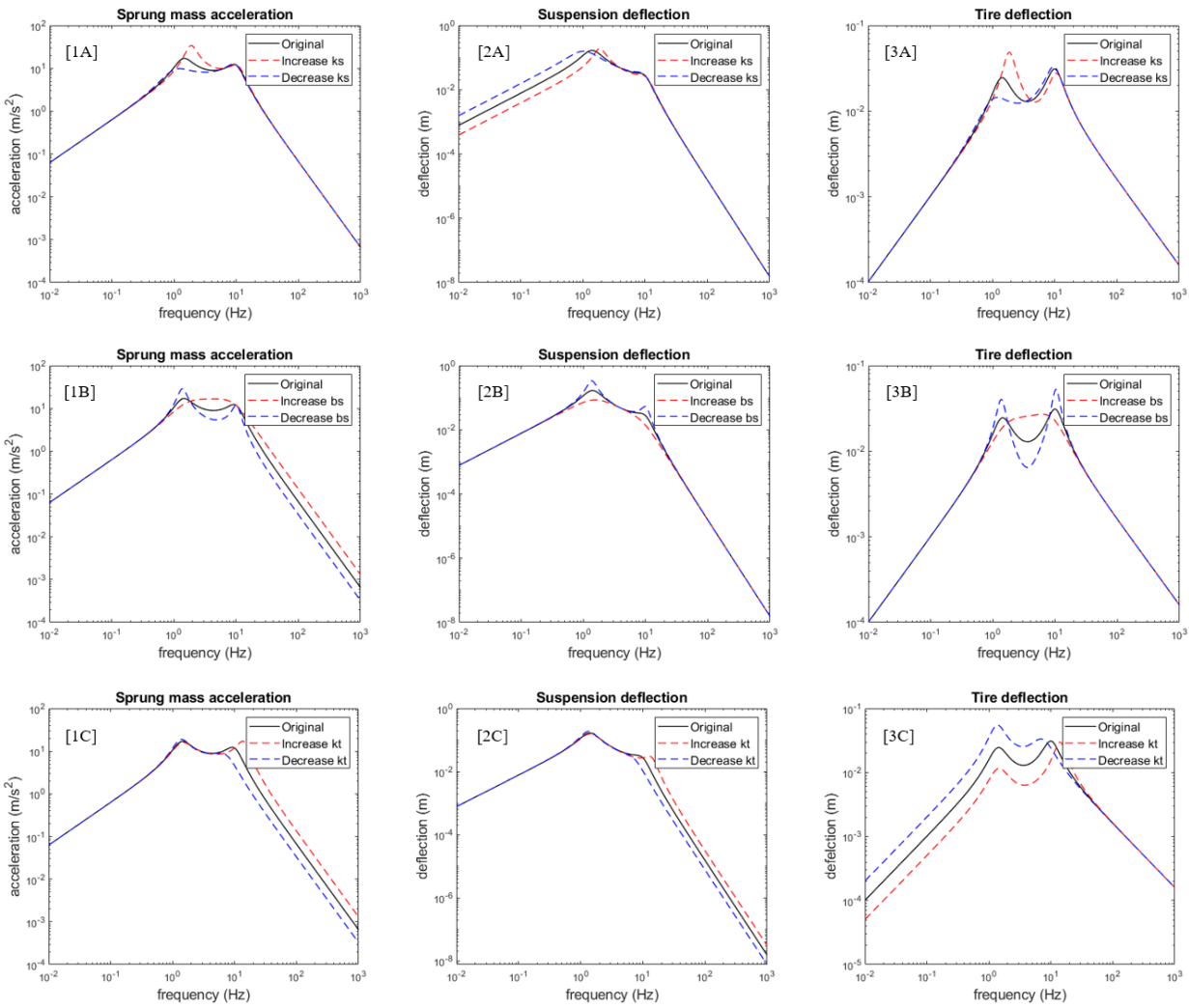


Fig. 8. Response of sprung mass acceleration, suspension deflection, tire deflection between ICE and EV. With the influence of varying suspension stiffness (1A-3A), the influence of varying damping coefficient (1B-3B) and the influence of varying tire stiffness (1C-3C).

Table 2: Effects of adjusting suspension parameters.

Suspension changes	Impact on sprung mass acceleration	Impact on suspension deflection	Impact on tire deflection
Decreased spring stiffness (k_s)	Reduced resonant peak at low frequency	Increased range of deflection at low frequency	Reduced resonant peak at low frequency
Decreased spring coefficient (b_s)	Reduced resonant peak at low frequency but increased range of deflection at high frequency	Reduced resonant peak at both low and high frequency	Reduced resonant peak at both low and high frequency
Decreased tire stiffness (k_t)	Reduced resonant peak and reduced range of deflection at high frequency	Reduced range of deflection at high frequency	Increased range of deflection at both low and high frequency

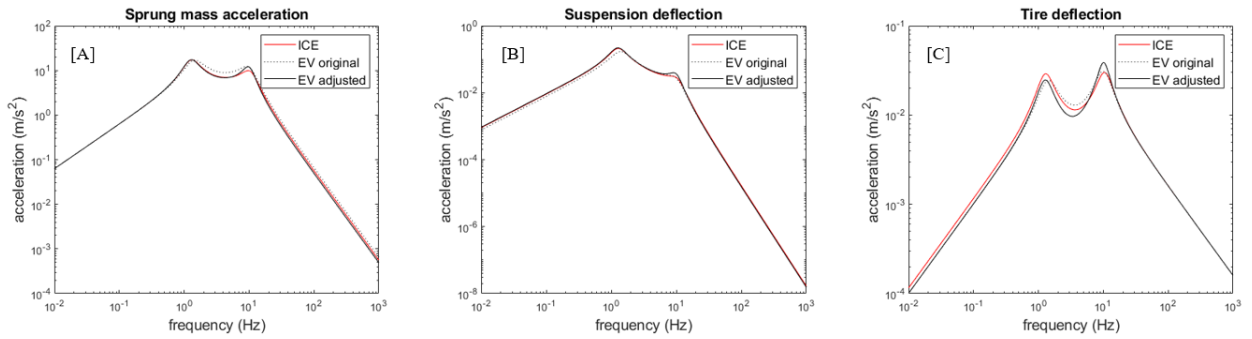


Fig. 9. ICE, EV Original and EV adjusted response comparison of sprung mass acceleration (A), suspension deflection (B), tire deflection (C).

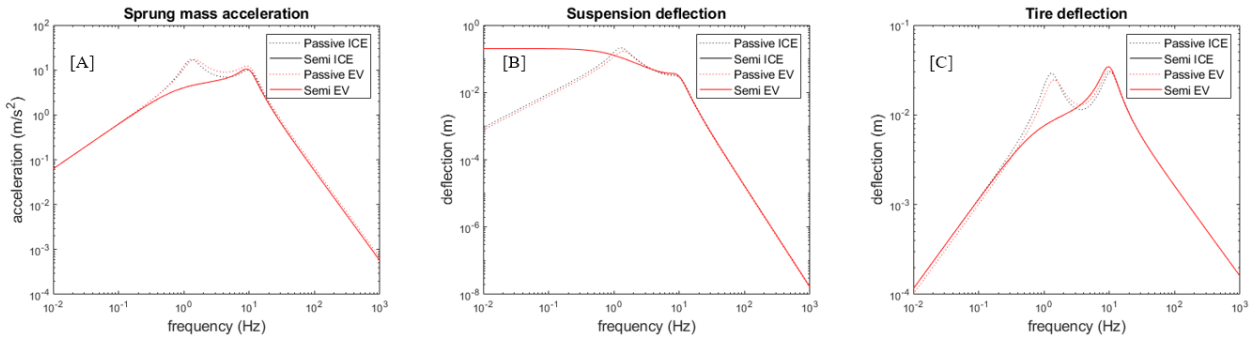


Fig. 10. Response of sprung mass acceleration (A), suspension deflection (B), tire deflection (C) between ICE and EV with and without controller.

After weighing the benefits and drawbacks of each suspension parameter, it is possible to achieve a set of values that make EV suspension outperform ICE in every function.

Figure 9 shows the performance comparison of ICE and EV after adjusting the parameters to $k_s = 20,000$ N/m, $b_s = 1,500$ Ns/m and $k_t = 220,000$ N/m. In the sprung mass acceleration response, originally EVs have worse comfort, but after adjusting the parameters, EVs can provide better comfort than ICE vehicles. The tire deflection response indicates that EVs can already provide more grip than ICE vehicles, and the parameter adjustment can further enhance road grip. However, the suspension deflection shows a different outcome, as the suspension lifetime could be worsened after such adjustments. Despite this advancement, adjusting these parameters accordingly is not practical in the real world. For example, decreasing the spring stiffness can improve EV stability, but this parameter is also responsible for carrying the entire vehicle weight. Reducing its stiffness might cause damage to the suspension spring due to the load.

To solve the issue, applying a skyhook damping control method can improve the response of both ICE and EV without affecting other systems. The active suspension simulation was performed by constructing a transfer function that uses $A - B_1K$ as the state matrix. The results are then displayed using a Bode plot comparing both the before and after employing control law and ICE versus EV comparison.

Figure 10 illustrates the response of ICE and EV before and after skyhook-damped control. It can be observed from (A) and (C) that when applying the control method, the resonant peak at low frequency of the active suspensions is diminished, resulting in better comfort and road holding. However, suspension lifetime is significantly reduced, as observed from the increased range of deflection in (B). The use of active suspension control can significantly improve both comfort and road holding performance but does have a trade-off in faster suspension deterioration.

3.2 Roll stability

Similar to the suspension performance comparison in Section 3.1, roll stability can also be evaluated using a Bode diagram by employing transfer function (15). In this analysis, lateral acceleration is used as the input over a range of frequencies, and the roll angle is measured as the gain. The simulation is carried out by having the vehicle drive in a sinusoidal wave scenario ($a_y = A \sin(2\pi ft)$, where A is amplitude and f is frequency (Hz)) as present in Figure 11. This can be done by steering the steering wheel angle as a sinusoidal wave while keeping the longitudinal velocity constant.

Figure 12 presents the frequency response of the roll angle gain, where EVs are shown to have more resistance to rolling at low frequency, but at high frequency the results are similar. This result indicates that a lower c.g. height, characteristic of EVs, seems to provide greater roll stability at low frequency. Figure 13 shows the roll angle comparison from Figure 12 in the time domain. At $A = 1 \text{ m/s}^2$ ($a_y = \sin(t)$) EVs showed a similar roll angle compared to ICEs. Although both vehicles exhibit a similar roll angle trend when cornering at high speed, the EV shows superior performance compared to the ICE during low-speed cornering.

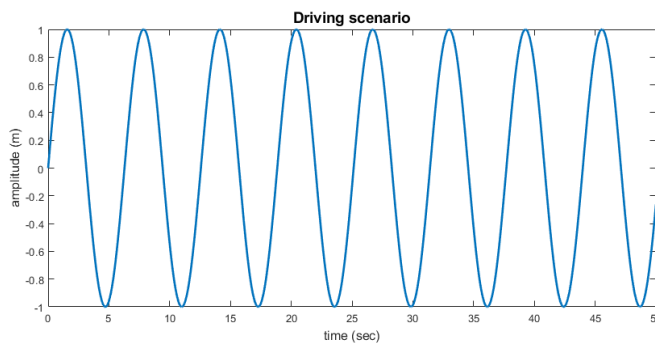


Fig. 11. Driving scenario for roll angle estimation at ($a_y = \sin(t)$).

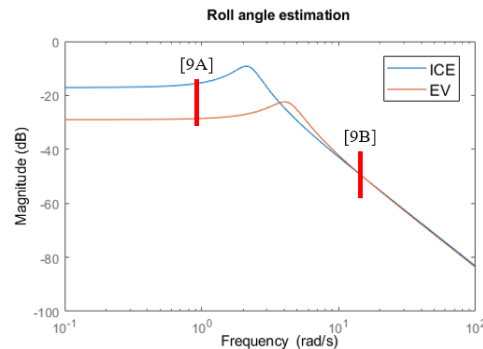


Fig. 12. Response of roll angle estimation of ICE and EV in frequency domain.

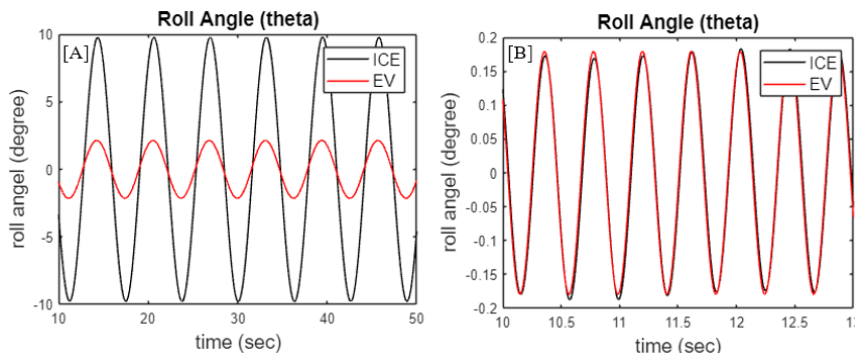


Fig. 13. Response of roll angle estimation of ICE and EV in time domain at $a_y = 1$ (A) and at $a_y = 15 \text{ m/s}^2$. (B)

3.2.1 Rollover index

In addition to roll angle estimation in the previous section, a rollover index is used to quantify the risk of a vehicle rolling over during dynamic maneuvers. The rollover index is a reliable measure that accurately indicates real-time rollover danger. Phanomchoeng et al. [20] suggested that the rollover index can be described as a function of vertical tire forces. Although there are various types of rollover indices, they are all derived from the same fundamental model, as illustrated in Figure 5. The fundamental definition of the rollover index is described by:

$$R = \frac{F_{zr} - F_{zl}}{F_{zr} + F_{zl}} \quad (31)$$

where $-1 \leq R \leq 1$ the right and left vertical tire forces are denoted by F_{zr} and F_{zl} respectively. A vehicle is considered to roll over when R is more than 1 or less than -1 . Given that the vertical tire forces F_{zr} and F_{zl} cannot be measured, the definition of R in (31) cannot be implemented in real time. However, using the same model in Figure 5, $F_{zr} + F_{zl}$ and $F_{zr} - F_{zl}$ can be calculated. This allows for an implementable version of the rollover index R to be obtained in terms of a_y and ϕ . The corresponding example of rollover index equation is given by:

$$R = \frac{F_{zr} - F_{zl}}{F_{zr} + F_{zl}} = \frac{2m_s a_y h_R}{m g l_w} + \frac{2m_s h_R \tan \phi}{m l_w} \quad (32)$$

where $-1 \leq R \leq 1$, $m = m_s + m_u$ and l_w represented track width, assumed to be equal to l_s ($l_w = l_s$). Noted that this type of rollover index is used for detecting untripped vehicles only. As observed from Figure 12, the resonant peak frequencies, where the roll of ICE and EV is maximized are 2.12 and 4.03 rad/s respectively, these values are then used to find the largest possible roll index of each vehicle. The inputs for equation (16) will be set as $a_y = \sin(2.12t)$ and $a_y = \sin(4.03t)$, respectively. After the simulation of equation (16) in time domain, the results of a_y and ϕ are used to compute roll index as per equation (32). The results show that ICE had the highest value of roll index at 0.544 while EV had only 0.100.

3.3 Yaw stability

As mentioned before, the yaw stability comparison can be quite different from the suspension performance and roll stability. First, the yaw rate comparison can be derived directly by substituting Table 1 parameters into equation (24). The result shows that EV had a yaw rate 9.5 times higher compared to ICE. For further verification, a simulation concerning errors in position and orientation relative to the road was employed.

The comparison was done by simulating the error tracking lateral dynamics model from equation (30) in Simulink. The simulation included the following driving scenario: the road is initially straight for the first 10 seconds and then becomes a circle with a radius of 500 meters, with the vehicle traveling at a constant speed of 30 m/s. The road trajectory is shown in Figure 14.

As seen in previous section, the lateral dynamics of the vehicle is given by:

$$\dot{x} = Ax + B_1 \delta + B_2 \ddot{\psi}_{des} \quad (33)$$

where $x = [e_1 \quad \dot{e}_1 \quad e_2 \quad \dot{e}_2]$

The first result showed in Figure 15 was a closed-loop system with the state-feedback controller given by:

$$\dot{x} = (Ax + B_1 K)x + B_2 \ddot{\psi}_{des} \quad (34)$$

The command $K = \text{place}(A, B_1, P)$ in Matlab yielded a feedback matrix K such that the eigenvalues of the matrix $A - B_1 K$ was placed at $[-5-3j-5+3j-7-10]^T$. This procedure led to the simulation results shown in Figure 15.

Even though the matrix $A - B_1 K$ is asymptotically stable, due to the presence of the $B_2 \ddot{\psi}_{des}$ term, the tracking errors will not converge to zero. Hence, the EV showed a more error of the distance of the c.g. of the vehicle from the center line of the lane and the orientation error of the vehicle with respect to the road when compared to ICE, which is concerning given that lane departure is a significant contributor to traffic accidents.

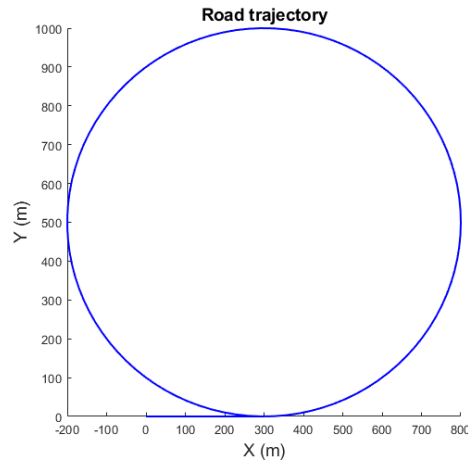


Fig. 14. Road trajectory input for yaw stability simulation.

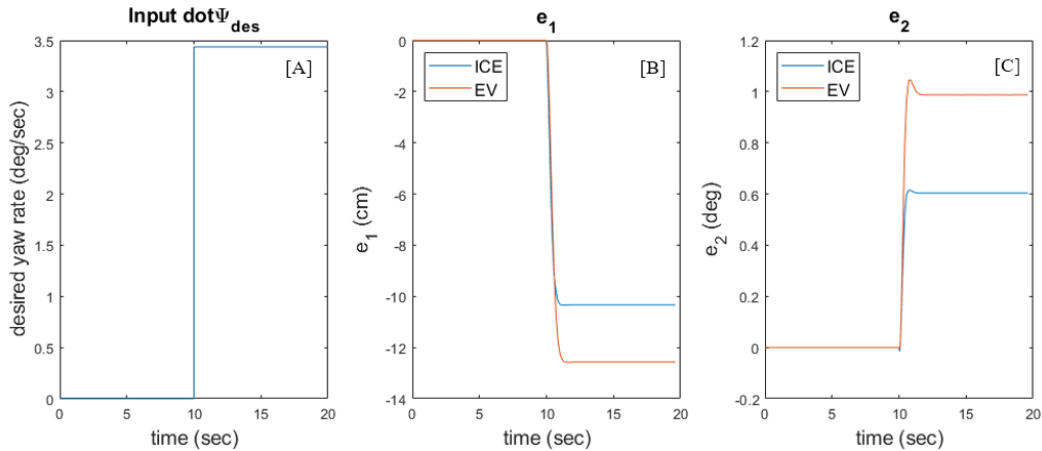


Fig. 15. Input psi dot desire (A) with the corresponding error of distance of the c.g. with respect to the road centerline (B) and the orientation error of the vehicle with respect to the road (C) of ICE and EV.

3.3.1 Lateral control

Unintended lane departures are recognized as a major cause of road accidents involving passenger cars. Advanced driver assistance systems (ADAS) and autonomous driving features, such as lane-keeping and lane-changing controls, have the potential to significantly reduce the severity of, or even prevent, many of these accidents. The success of these driving features largely relies on the efficiency and performance of the vehicle's steering control system [33]. Similarly, a lateral control method was considered to evaluate the response of ICE and EV.

From now on, the investigation will focus on whether the use of a feed-forward term of steering input $\delta = -Kx + \delta_{ff}$ can ensure zero steady-state feedback on a curve. The closed-loop system is then given by: [6]

$$\dot{x} = (A - B_1 K)\delta + B_1 \delta_{ff} + B_2 \Psi_{des} \quad (35)$$

where:

$$\delta_{ff} = \frac{L}{R} + K_V a_y + k_3 e_{2,ss} \quad (36)$$

* More details about this controller can be found in [10]

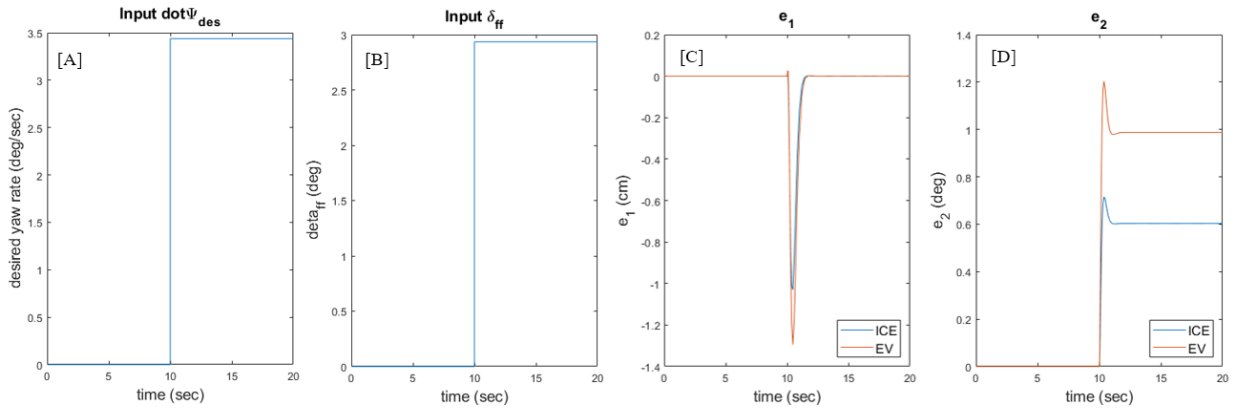


Fig. 16. Input psi dot desire (A), input of steering control angle (B) the corresponding error of distance of the c.g. with respect to the road centerline (C) and the orientation error of the vehicle with respect to the road (D) of ICE and EV.

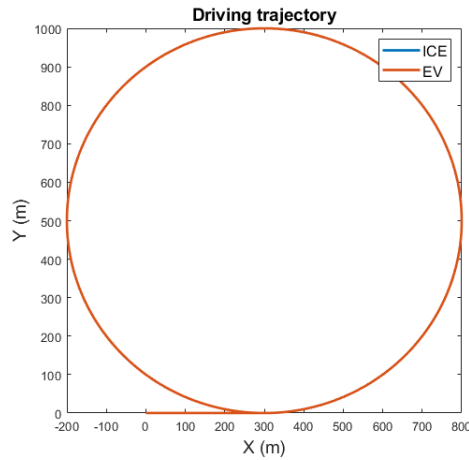


Fig. 17. ICE and EV driving simulation after lateral control.

According to the simulation, the vehicle is made to track a lane with a radius of 500 meters, as shown in Figure 14. With the model in Section 2.4, the lateral position error (e_1) and yaw angle error (e_2) of the vehicle are defined in equation (30). Using the state feedback control, the steering angle depends on the state variable ($\delta = -kx$) and the desired yaw rate can be computed by equation (25). The results of using this control are shown in Figure 15. Figure 15(A) shows the desired yaw rate, and Figures 15(B) and 15(C) show the errors in the lateral position and yaw angle for both ICE and EV vehicles, respectively. Figures 15(B) and 15(C) show that the steady-state errors of EV vehicles are slightly more than those of ICE vehicles. The steady-state lateral position error of EV vehicle is approximately 2 cm more than that of ICE vehicles, and the steady-state yaw angle error of EV vehicles is approximately 0.4 degrees more than that of ICE vehicles. This is because the geometry parameters (m , l_f , l_r) of ICE and EV vehicles are different.

Using a feedforward controller with state feedback, as shown in equations (35)-(36), helps compensate for the steady-state lateral position error. The results of this control are shown in Figure 16. Additionally, Figure 17 illustrates the driving scenario after applying the control. Figures 16(A) and 16(B) show the inputs of the system. Figure 16(C) shows that the feedforward controller with state feedback successfully compensates for the steady-state lateral position errors for both ICE and EV vehicles. The steady-state lateral position errors become zero. However, the feedforward controller cannot compensate for the steady-state yaw angle error. The yaw angle error remains the same as with state feedback control. In short, the performance of the lateral control, specifically the feedforward controller with state feedback, is similar for both ICE and EV vehicles. It can reduce the steady-state lateral position error.

4. Discussion

It is important to note that the ICE and EV models used in this study are constructed from averaged parameters across 13 prior works. As such, they serve as representative vehicle models, rather than being based on a single production vehicle. While this approach provides a useful generalization, results may not apply to all vehicle designs and configurations.

The Bode plot is a crucial tool for understanding the frequency response of a suspension system, particularly in comparing ICE and EV configurations. It illustrates how the amplitude of the output signal varies with frequency, indicating how the suspension system amplifies or attenuates vibrations at different frequencies. In this context, the Bode plot helps evaluate the performance differences between ICE and EV by examining the gain values at various frequencies. Typically, higher gain values indicate a harsher response, suggesting lower performance.

We emphasize the necessity of comparing ride comfort, suspension lifetime, and road holding, as these functions play a crucial role in determining a vehicle's stability and riding performance for users. The findings show that an increase in both sprung and unsprung mass negatively impacts ride comfort, as stated in [3] and [4]. However, our findings yield different results in road holding performance, as the tire deflection response suggests that EVs could provide more road grip than ICEs.

To mitigate the adverse effects of increased unsprung mass in EVs, a skyhook-damped semi-active suspension control strategy was applied to the specific vehicle model used in this study. The simulation results demonstrated improvements in both ride comfort and road holding under the tested conditions. However, the effectiveness of this control method depends on specific vehicle configurations and tuning parameters. Further research is required to evaluate its applicability across a wider range of EV designs and real-world driving conditions.

It is important to note that this study evaluates a specific set of vehicle configurations and does not aim to provide a generalized comparison of all ICE and EV suspension characteristics. Future work should incorporate a broader range of vehicle models and consider additional factors such as drivetrain dynamics, powertrain response, and variations in drivetrain natural frequencies to achieve a more comprehensive understanding of EV-specific suspension behavior.

In particular, ICE vehicles equipped with boxer engines also exhibit lower centers of gravity similar to EVs. These vehicle types were not considered in this study, and future research should explore such configurations to broaden the applicability of the findings.

Roll angle comparison between ICE and EV can be evaluated using a Bode diagram. This analysis uses lateral acceleration as the input over a range of frequencies, with the roll angle measured as the gain. By simulating a sinusoidal wave driving scenario, vehicles with a lower center of gravity, such as EVs, can withstand greater lateral acceleration before experiencing the risk of rollover, in a similar manner to [13]. For further analysis, the rollover index is used to verify the rollover chances of both vehicle types. This index helped by showing that EVs can offer significantly better roll resistance than ICEs.

In this paper, a lateral control study was conducted to evaluate yaw compensation performance between ICE and EV vehicles. The control strategy used is a Feedforward Controller with State Feedback, guided by [10]. The advantage of the Feedforward Controller with State Feedback is that it is simple and a technique currently in use. It makes it possible to maintain cornering with zero error, and using existing sensors for feedback does not require adding extra sensors to the vehicle. While other research [16], [17], and [18] highlighted the benefits of EVs' yaw moment compensation, we addressed the issue of EV parameters that turned out to be gradually altered from ICE. Nevertheless, the usage of lateral control could improve EV's performance by eliminating vehicle orientation error with respect to the road.

Table 3 concludes the models used for each stability aspect and their corresponding simulation results. Initially, the suspension performance indicated that EVs provided less comfort compared to ICEs. However, by applying a control system, this flaw in EVs can be mitigated. Similarly, in the yaw stability aspect, EVs were more prone to yaw instability. Implementing the control system effectively eliminated this issue in EVs. On the other hand, in terms of roll stability, EVs demonstrated superior performance, thus negating the need for any additional control methods.

Table 3: Model used in simulation, before and after control result conclusion table.

	Suspension performance	Roll stability	Yaw stability
Model	Quarter-car model	Un-tripped rollover model	Lateral dynamic model
Before control	Neutral EV provide less comfort but more traction	Good EV have more resist to roll characteristic	Bad EV showed a more road centered and yaw angle error
After control	EV provide more comfort	-	Remove EV road centered error

While this study focuses on a single set of vehicle parameters, the findings highlight key trends in EV stability and suspension tuning. Future work should incorporate parametric variations, including different battery placements, suspension stiffness levels, and control strategies, to establish a more generalized understanding of EV dynamics. The proposed skyhook-damped semi-active suspension control has demonstrated potential in improving EV ride comfort and handling stability, suggesting that further optimization of control strategies may enhance EV performance across a wider range of configurations. Additionally, exploring adaptive suspension systems that dynamically adjust to different driving conditions could further enhance the stability and ride quality of EVs.

5. Conclusion

In conclusion, this study provides a comparative analysis of suspension behavior, roll stability, and yaw stability between internal combustion engine (ICE) vehicles and electric vehicles (EVs). Through mathematical modeling and control strategies, the study quantifies stability differences and highlights key characteristics of each vehicle type under different configurations.

The findings indicate that while EVs generally exhibit lower ride comfort compared to ICE vehicles due to increased unsprung mass, they demonstrate better road holding characteristics. Adjusting suspension parameters can improve EV ride comfort but may lead to reduced suspension lifespan. The implementation of a skyhook-damped semi-active suspension control strategy was shown to mitigate some of these drawbacks, improving both ride comfort and stability. In terms of roll stability, EVs benefit from a lower center of gravity, resulting in greater resistance to roll at low frequencies. However, at higher frequencies, their roll behavior is comparable to that of ICE vehicles. Regarding yaw stability, EVs exhibit greater deviations from the desired trajectory when cornering. However, applying a lateral control system effectively minimizes yaw errors, improving EV cornering stability to levels similar to ICE vehicles.

This study contributes to the understanding of EV dynamics and informs future suspension tuning and stability enhancement strategies for next-generation vehicles. While the modeling approach used provides a structured framework for evaluating vehicle stability, it primarily captures frequency-domain behavior and does not fully reflect transient dynamics or real-world conditions. Future research should explore nonlinear vehicle dynamics models, real-time experimental validation, and a broader range of vehicle configurations to further refine these findings. Additionally, integrating adaptive suspension control strategies tailored specifically for EVs could provide further enhancements in ride comfort and handling.

The findings are based on simulations of representative vehicle models built from averaged data, and thus may not fully capture the behavior of individual vehicle designs. These findings contribute to the development of optimized suspension strategies for EVs, improving ride comfort, stability, and road handling. Future advancements in EV suspension design, including nonlinear modeling and experimental validation, will further refine vehicle control strategies and enhance driving performance. Further investigations involving alternative ICE layouts, such as those with boxer engines, are recommended to validate and extend the present comparative results.

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