Performance analysis of semi-active suspension in an electric vehicle with acceleration-driven damping

Bui Van Cuong[1], Le Van Quynh[1*], Nguyen Tien Dung[2], Hoang Anh Tan[1]

[1]Faculty of Vehicle and Energy Engineering, Thai Nguyen University of Technology, 666 3-2 Street, Thai Nguyen City, Thai Nguyen, Vietnam;

[2]Department of Vehicle and Energy Conversion Engineering, School of Mechanical Engineering, Hanoi University of Science and Technology, 1 Dai Co Viet, Hai Ba Trung, Hanoi, Vietnam.

[*]Corresponding author: lequynh@tnut.edu.vn

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ABSTRACT

In order to enhance ride comfort performance of an electric vehicle (EV) semi-active suspension system (SASs), a 5-degree-of-freedom dynamic model of an EV is proposed to analyze their effectiveness on ride comfort. A SASs with Acceleration-Driven Damping (ADD) is proposed for enhance ride comfort performance of EV SASs. The Firefly Algorithm (FA) is used to optimize the ride performance of the SASs. The root mean square (RMS) values of vertical driver's seat acceleration (a_{ws}), vertical body vehicle acceleration (a_{wb}), and pitching body vehicle acceleration (a_{wphi}) are selected based on the ISO 2631:1997(E) standard. The achieved results indicate that the a_{ws}, a_{wb} and a_{wphi} values with the proposed SASs respectively reduce by 12.07%, 6.85%, and 21.47% compared to the original passive suspension systems (PSSs) when vehicle moves on ISO road class B at a speed of 20 m/s and full load. In addition, the ride effectiveness is verified under various vehicle velocities.

Keywords: Semi-active Suspension; Electric Vehicle; Acceleration-Driven Damping; Optimization.

1. INTRODUCTION

The automotive industry has recently shifted toward electric vehicles, addressing key energy scarcity issues and environmental pollution inherent in internal combustion engine vehicles [1]. Despite these advancements, vibration remains a challenge in assessing electric vehicle quality. Therefore, the suspension system is crucial for preventing vibrations caused by external factors. The suspension system in electric vehicles, similar to basic passive systems consisting of metal springs and dampers, only responds to specific conditions [2]. Additionally, under varying road conditions, it struggles to meet the criteria for comfort and safety. To address this issue, active suspension systems have been introduced to enhance suspension flexibility [3, 4]. However, these systems are costly and consume large amounts of energy [5]. As a result, semi-active suspension systems have emerged as a promising solution [6, 7]. Semi-active vibration control strategies have increasingly attracted the attention of researchers. The initial approach for semi-active suspension control is skyhook control. Karnopp et al. [8] introduced the concept of skyhook control, which enhances ride comfort by connecting a linear viscous damper between the sprung mass and an imaginary fixed point. Yi et al. [9] presented an adaptive skyhook control for semi-active suspension, which includes a skyhook control law with road-adaptive gains and a road detection algorithm. This strategy is considered simple and effective in theoretical research. However, practical implementation poses challenges, leading to the development of groundhook control. Jiří Žáček et al. [10] designed a damper controlled by a modified Groundhook algorithm, which demonstrated a significant improvement in comfort. In another study, Valášek et al. [11] applied the groundhook concept to semi-active suspension for heavy trucks, aiming to minimize dynamic tire-road force and thus reduce road damage. To further enhance performance, many scholars have employed hybrid control strategies [12, 13]. Overall, these proposals have been effective in reducing vibrations. However, practical implementation using control data remains a challenge. To address this, the Acceleration-Driven Damper (ADD) control strategy was introduced by Savaresi et al. [14]. This control strategy is suitable for real-world applications, which has garnered significant interest and development from scholars [15, 16]. In addition, some studies on control algorithms for semi-active suspension systems are presented in the literatures [21, 22] and [23] . In this study, a Firefly Algorithm (FA) is applied to optimize the ADD controller, further enhancing ride comfort in electric vehicles. The remainder of the paper is organized as follows: Section 2 presents the design of a half-vehicle dynamic model. Section 3 focuses on the design and optimization of the control strategy. The ride performance of SASs is analyzed in section 4. Finally, conclusions are drawn in section 5.

2. THE DYNAMIC MODEL OF ELECTRIC VEHICLE

2.1. Half-vehicle dynamic model

To evaluate the ride performance of EV, an 5-degree-of-freedom dynamic model of EV is established, as shown in figure 1.

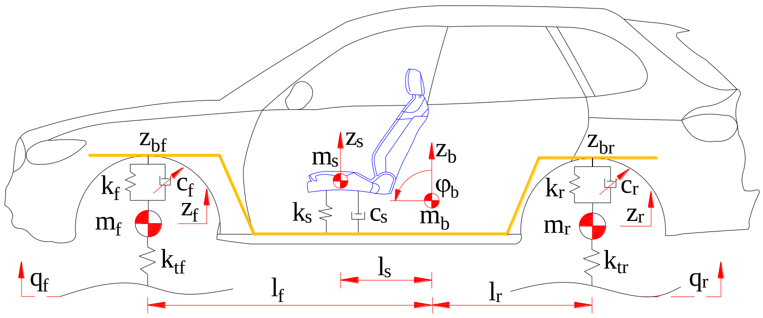

Figure 1. *Electric vehicle dynamic model.*

In this model, m_b and I_c denote the vehicle body mass and moment of inertia, respectively; m_f, m_r, and m_s represent the front and rear axle masses and driver's seat mass, respectively; l_f, l_s, *and* l_r represent the distances. z_b, z_f, z_r, z_s, and ϕ_b represent the vehicle body displacement, front and rea axle displacement, driver's seat displacement, and vehicle body pitching angle displacement, respectively. Additionally, k_f, c_f, k_r, and c_r represent the stiffness and damping coefficients of the front and rear suspension systems, while k_s and c_s represent the stiffness and damping coefficients of the driver's seat suspension. Finally, k_{tf} and k_{tr} represent the stiffness of the front and rear tires, respectively.

According to the Newton's second law, the differential equation could be established.

The vertical motion equation of the driver's seat is given by

$$m_s \ddot{z}_s = -F_s \tag{1}$$

Where: $F_s = [k_s(z_s - z_{sb}) + c_s(\dot{z}_s - \dot{z}_{sb})]$

The vertical motion equation of vehicle body is given by

$$m_b \ddot{z}_b = F_s - (F_f + F_r) \tag{2}$$

The body pitching motion equation is given by

$$I_b \ddot{\varphi}_b = F_f l_f - F_r l_r - F_s l_s \tag{3}$$

Where:

$$F_f = k_f(z_{bf} - z_f) + c_f(\dot{z}_{bf} - \dot{z}_f); \quad F_r = k_r(z_{br} - z_r) + c_r(\dot{z}_{br} - \dot{z}_r) \tag{4}$$

The vertical motion equation of the front wheel (unsprung mass) is given by

$$m_f \ddot{z}_f = F_f - F_{tf} \tag{5}$$

The vertical motion equation of the rear wheel (unsprung mass) is given by

$$m_r \ddot{z}_r = F_r - F_{tr} \tag{6}$$

Where: $F_{\mathrm{tf}} = k_{tf}(z_f - q_{\mathrm{f}})$; $F_{\mathrm{tr}} = k_{tr}(z_r - q_r)$

The vertical displacement of the two points of the vehicle body is given by:

$$z_{sb} = z_b - l_s \varphi_b ; z_{bf} = z_b - l_f \varphi_b ; z_{br} = z_b + l_r \varphi_b \tag{7}$$

2.2. Road roughness excitation

Road unevenness is the most crucial factor affecting driving comfort. Therefore, the road input model forms the basis of the vehicle's dynamic response and control, closely related to suspension system performance. Generally, the rougher the road, the poorer the suspension system's performance response. In this study, a random road model [19] is selected to evaluate ride performance of the propose SASs. A filtered white noise of Gaussian distribution is used to generate the random road profile, and the time domain expression of road displacement is obtained as follows:

$$\dot{q}(t) = -2\pi f_0 q(t) + 2\pi n_0 \sqrt{G_0 v} w(t) \tag{8}$$

where f_0 is the lower cut-off frequency, and the value is 0.1 Hz; n_0 is the space reference frequency, and the value is 0.1 m^{-1}; G_0 is the road roughness coefficient; v is the driving speed; w is the Gaussian white noise with a mathematical expectation of zero; q is the road displacement.

3. DESIGN AND OPTIMIZE CONTROL FOR ADD

3.1. Acceleration driven damping control

Acceleration Driver Damper (ADD) control was developed by Savaresi et al. in 2005. The primary objective of ADD control is to improve vehicle ride comfort and handling by optimally adjusting the damping force based on the acceleration signals from the vehicle body. This approach contrasts with traditional passive damping systems, which have a fixed damping characteristic and cannot adapt to changing road conditions or driving dynamics. Interestingly, the SH and ADD algorithms share a very simple and similar structure. The ADD control law is defined as:

$$c_f = \begin{cases} c_{maxf}, & \ddot{z}_{bf}(\dot{z}_{bf} - \dot{z}_f) \geq 0 \\ c_{minf}, & \ddot{z}_{bf}(\dot{z}_{bf} - \dot{z}_f) < 0 \end{cases} ; \; c_r = \begin{cases} c_{maxr}, & \ddot{z}_{br}(\dot{z}_{br} - \dot{z}_r) \geq 0 \\ c_{minr}, & \ddot{z}_{br}(\dot{z}_{br} - \dot{z}_r) < 0 \end{cases} \tag{9}$$

3.2. Optimal controller design

The FA, developed by Yang in 2010 [17], is inspired by the natural behavior of fireflies, using their self-luminosity to move closer to each other in the dark. It has become well-known for solving complex optimization problems [18]. The FA is developed based on three ideal assumptions: Firstly, all fireflies are unisex, meaning each can be attracted to other fireflies regardless of gender. Secondly, attractiveness is linked to intensity, a function of the distance between the firefly in question and the other fireflies. The attractiveness decreases as the distance increases. Finally, the luminosity or light intensity of a firefly is determined by the value of the cost function of the problem at hand.

The light intensity *(I)* can be determined as follows

$$I = I_0 \exp(-\gamma . r_{ij}) \tag{10}$$

where: γ is the absorption coefficient and (I_0) is the initial value at (r = 0).

The attractiveness β is expressed by

$$\beta = \beta_0 \exp(-\gamma . r_{ij}^m) \quad (m \geq 1) \tag{11}$$

The distance between fireflies *i* and *j* is defined via Euclidean distance

$$r_{ij} = \left| r_i - r_j \right| = \sqrt{\sum_{k=1}^{D} (x_{ik} - x_{jk})^2} \tag{12}$$

The motion equation of the *i-th* firefly to the *j-th* one is determined by

$$x_i(t+1) = x_i(t) + \beta(x_j(t) - x_i(t)) + \alpha(rand - 0.5) \tag{13}$$

where $x_i(t + 1)$ is the position of firefly *i* at iteration *t +1* displacement. The optimization flow of the Firefly Algorithm is illustrated in figure 2.

The optimization process aims to enhance three performance evaluation indices of the suspension system under random road input conditions when the vehicle is moving at a speed of 20 m/s, namely RMS values of the driver's seat acceleration (*SA*), body acceleration (*BA*), and pitching body acceleration (*PA*). The fitness function optimized by the firefly algorithm is expressed as follows:

$$minJ = \alpha_1 \frac{SA(X)}{SA_{pass}} + \alpha_2 \frac{BA(X)}{BA_{pass}} + \alpha_3 \frac{PA(X)}{PA_{pass}} \tag{14}$$

$$X = [c_{\max f} \ c_{\min f} \ c_{\max r} \ c_{\min r}] \tag{15}$$

$$lb < X_i < ub \qquad i{=}1,2,3,4 \tag{16}$$

The formula provided uses J to denote the fitness function and *X* to represent the set of parameters to be optimized. *SA(X)*, *BA(X)*, and *PA(X)* indicate the a_{ws}, a_{wb} and a_{wphi} values of proposed SASs. In contrast, SA_{pass}, BA_{pass}, and PA_{pass} indicate the a_{ws}, a_{wb} and a_{wphi} values of the passive suspension systems (PSSs). α_1, α_2, and α_3 represent weighting coefficients and the sum of the three is 1.The ranges of the parameters to be optimized, represented by lb and ub, are set to [0, 0, 0, 0] and [5000, 2500, 5000, 2500], respectively. Table 1 presents the relevant parameters of the vehicle's passive suspension model.

Table 1. *Parameters of the passive suspension.*

Parameter	Unit	Value	Parameter	Unit	Value
m_s	kg	70	I_b	Kg. m^{-2}	1940
m_b	kg	1100	k_{tr}	($N \cdot m^{-1}$)	200000
m_f	kg	60	c_s	($N \cdot s \cdot m^{-1}$)	1250
m_r	kg	65	c_{f_pas}	($N \cdot s \cdot m^{-1}$)	1500
k_f	($N \cdot m^{-1}$)	17600	c_{r_pas}	($N \cdot s \cdot m^{-1}$)	1700
k_r	($N \cdot m^{-1}$)	22300	l_f	m	1.15
k_s	($N \cdot m^{-1}$)	12000	l_r	m	1.30
k_{tf}	($N \cdot m^{-1}$)	180000	l_s	m	0.68

At the end of the optimization process, the optimal parameters of the semi-active suspension system for the electric vehicle are presented in table 2.

Table 2. *The optimized parameters of the semi-active suspension system.*

Optimized Parameters	Unit	Value
The maximum damping coefficient c_{maxf}	($N \cdot s \cdot m^{-1}$)	3500
The minimum damping coefficient c_{minf}	($N \cdot s \cdot m^{-1}$)	502.6
The maximum damping coefficient c_{maxr}	($N \cdot s \cdot m^{-1}$)	3600
The minimum damping coefficient c_{minr}	($N \cdot s \cdot m^{-1}$)	519.3

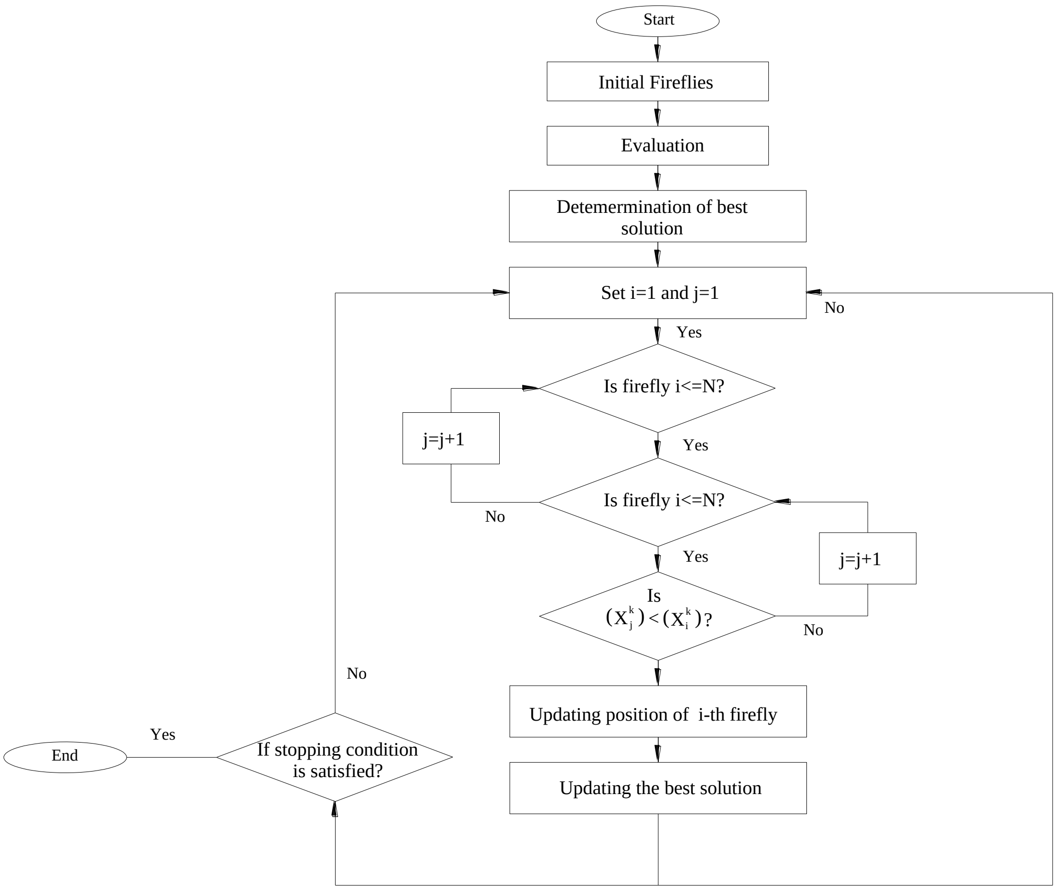

Figure 2. *The flowchart of firefly algorithm.*

4. RESULTS AND DISCUSSION

4.1. Simulation of the semi-active suspension system

The content of this section involves the simulation and ride performance analysis of the proposed SASs compared to PSSs when the vehicle moving on an ISO class B at a vehicle speed of 20 m/s and full load. The input parameters for EV with PSSs are taken from table 1, while the input parameters for SASs are provided in table 2. Matlab/Simulink software is chosen to carry out this work. The a_s, a_b and a_{phi} on the time domains with the proposed SASs compared to PSSs are shown in figure 3. From the a_s, a_b and a_{phi} values on the time domains in figure 3, the a_{ws}, a_{wb} and a_{wphi} values are defined based on ISO 2631 (1997) [20], as shown in table 3. The results in table 3 indicate that the a_{ws}, a_{wb} and a_{wphi} values with the proposed SASs respectively reduce by 12.07%, 6.85%, and 21.47% compared to the original passive suspension systems (PSSs) under survey conditions.

Table 3. *The ride performance of the proposed SASs compared to PSSs.*

Performance	Unit	RMS value		Amplification
		PSSs	Proposed SASs	(%)
a_{ws}	$m.s^{-2}$	0.4028	0.3542	12.07
a_{wb}	$m.s^{-2}$	0.4644	0.4326	6.85
a_{wb}	$rad.s^{-2}$	0.2510	0.1971	21.47

4.2. Analysis of ride performance of the proposed SASs under various vehicle velocity

In this section, we delve into the ride performance analysis of the proposed SASs under various velocities when the vehicle moves on ISO road surface class B and full load. The speed range examined spans from 5 to 30 m/s are selected for analysis. The results of the ride performance of the proposed SASs compared with PSSs under various velocities are shown in figures 4.

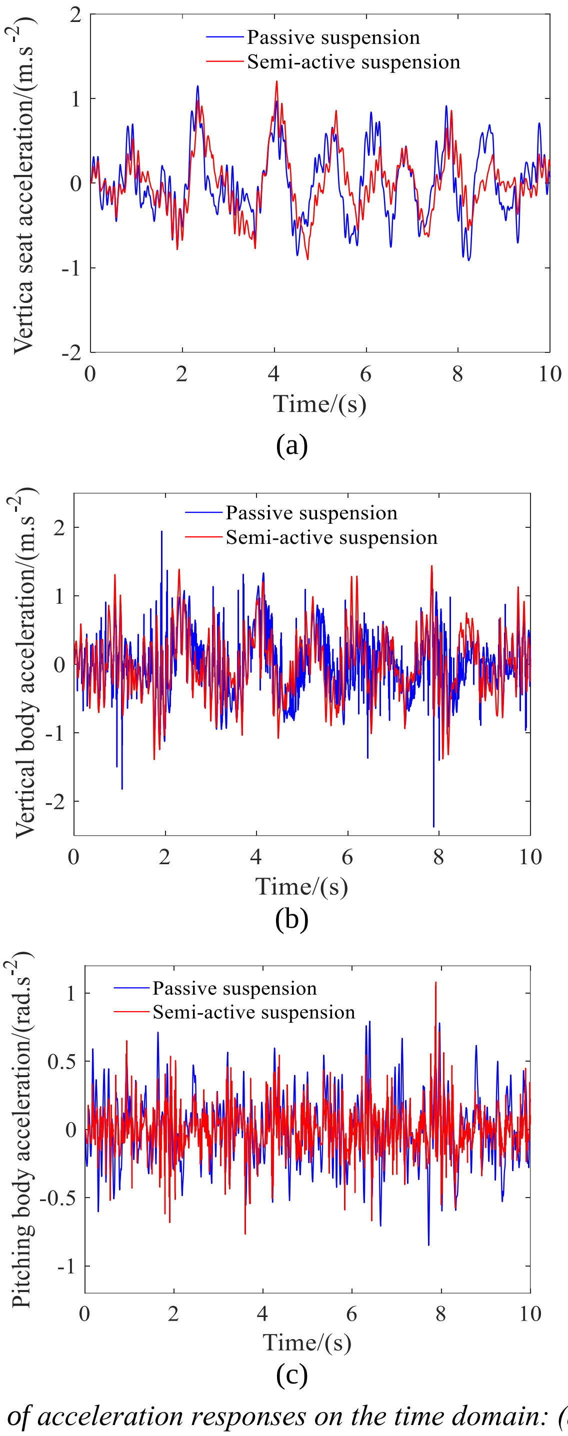

Figure 3. *Comparison of acceleration responses on the time domain: (a) vertical driver's seat, (b) vertical vehicle body, and (c) pitching vehicle body.*

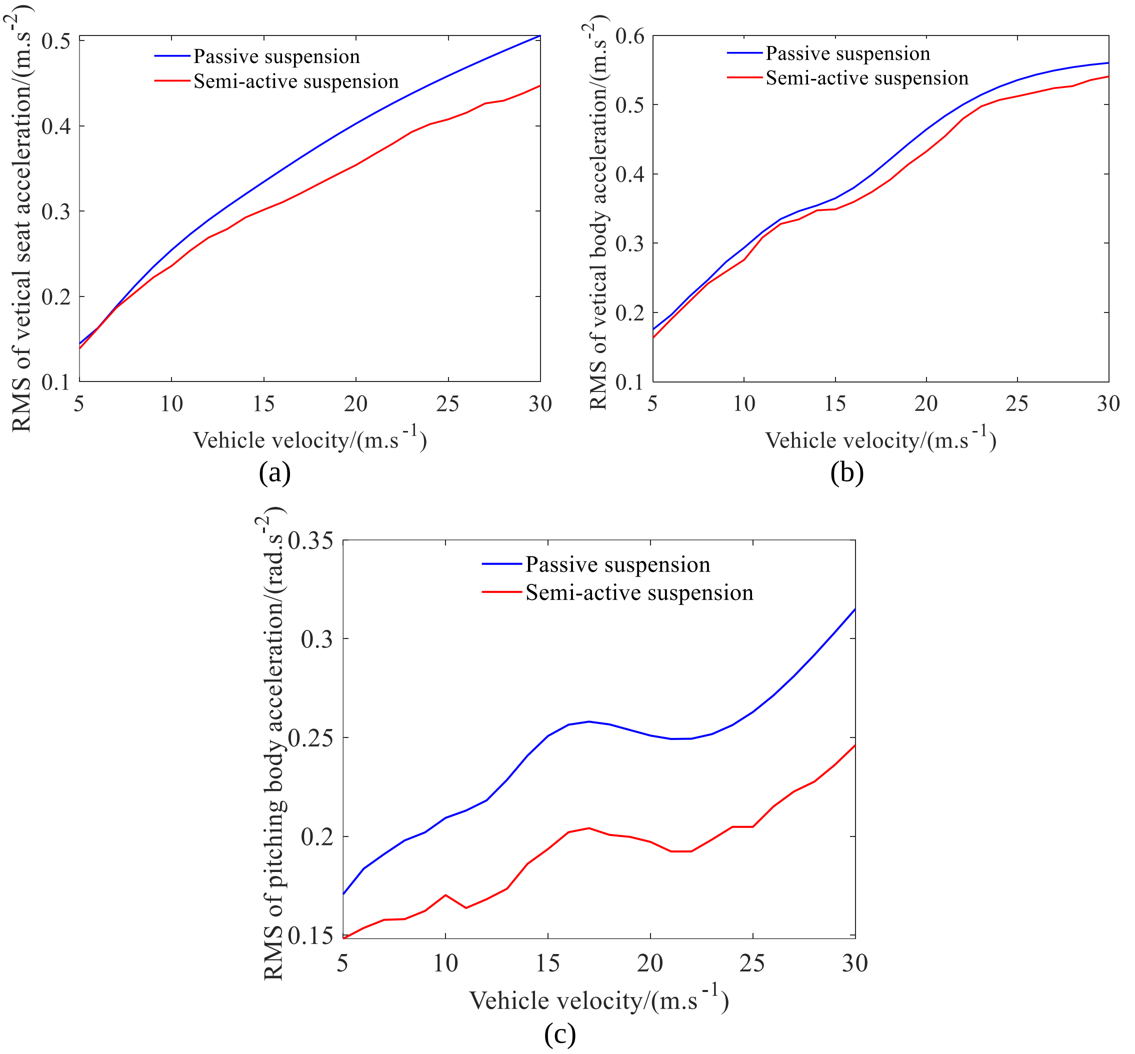

Figure 4. *Comparison of ride performance of proposed SASs under varying velocities*.

As illustrated in figure 4, the vehicle velocity values negatively impact the ride performance of both two types of the proposed SASs and PSSs. The a_{ws}, a_{wb} and a_{wphi} values with the proposed SASs respectively reduce compared to PSSs when the vehicle velocity value increases.

5. CONCLUSIONS

In this study, a 5-degree-of-freedom dynamic model of an EV was proposed to analyze their effectiveness on ride comfort. A SASs with Acceleration-Driven Damping (ADD) was proposed to enhance the ride comfort performance of EV SASs under some investigation conditions. The major conclusions drawn from the analysis can be summarized as follows: (i) The a_{ws}, a_{wb} and a_{wphi} values with the proposed SASs respectively reduce by 12.07%, 6.85%, and 21.47% compared to PSSs when the vehicle moving on an ISO class B at a vehicle speed of 20 m/s and full load and (ii) The a_{ws}, a_{wb} and a_{wphi} values with the proposed SASs respectively reduce compared to PSSs when the vehicle velocity value increases. In future research directions, the authors focus on analyzing and controlling for SASs with optimal controllers to improve vehicle ride comfort via the full-vehicle dynamic model of EVs with the combination of two excitation sources.

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TÓM TẮT

Phân tích hiệu suất của hệ thống treo bán chủ động trên xe điện với giảm chấn gia tốc

Để nâng cao độ hiệu quả độ êm dịu của hệ thống treo bán chủ động của một xe điện, một mô hình động lực học của xe điện với 5 bậc tự do được đề xuất để phân tích ảnh hưởng của chúng đến độ êm dịu. Một hệ thống treo bán chủ động với bộ điều khiển giảm chấn theo hướng gia tốc, (ADD- Acceleration-Driven Damping) được đề xuất cho việc nâng cao hiệu quả êm dịu của hệ thống treo bán chủ động. Thuật toán đom đóm được sử dụng để tối ưu hiệu quả êm dịu của hệ thống treo bán chủ động. Các giá trị bình phương bình (RMS) của gia tốc thẳng đứng của ghế ngồi người điều khiển (a_{ws}), gia tốc thẳng đứng tại vị trí thân xe (a_{wb}) và gia tốc lặc dọc thân xe (a_{wphi}) được lựa chọn như là các hàm mục tiêu dựa tiêu chuẩn quốc tế ISO 2631:1997(E). Các kết quả đạt được chỉ ra rằng, các giá trị a_{ws}, a_{wb} và a_{wphi} lần lượt giảm 12.07%, 6.85%, và 21.47% so với hệ thống treo bị động của xe nguyên bản khi ô tô chuyển động trên mặt đường ISO cấp B với vận tốc 20 m/s và đầy tải. Ngoài ra, hiệu quả êm dịu của hệ thống treo bán chủ động đề xuất được kiểm chức dưới điều kiện vận tốc khác nhau.

Từ khoá: Xe điện; Hệ thống treo bán chủ động; ADD; Thuật toán đom đóm; Êm dịu.