



Optimizing battery based on the evaluation of the natural frequency of electric vehicle bodies

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ABSTRACT

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Transitioning from internal combustion engine vehicles to electric vehicles is a strategic approach to achieving zero carbon emissions by 2050. However, the battery system alters the vehicle's weight and load distribution, impacting its vibration characteristics. This study evaluates the natural vibration frequency of the vehicle body to optimize battery, ensuring safety and operational stability. The research focused on a HONDA Civic 1.8L converted into an electric vehicle. A simulation model of the suspension system, calibrated using a full-vehicle dynamics model, was developed and evaluated per the QCVN 09:2011/BGTVT standard issued by Vietnam's Ministry of Transport. The vehicle was placed on a test bench at a height of 120 mm, released freely, and the acceleration response was recorded and analyzed over time. The simulation results indicated that the natural frequencies of the original vehicle and three battery configurations were below the 2.5 Hz threshold specified by QCVN 09:2011/BGTVT. However, the frequencies varied across the configurations compared to the original vehicle. Specifically, the front axle frequency in the three cases increased by 10.7–18.4% compared to the original vehicle's 1.96 Hz, while the rear axle frequency decreased by 26.5–28.5% relative to the original 2.00 Hz. Based on the simulation, Case 3 is identified as the optimal battery configuration, as both the front and rear axle frequencies are lower than those of the other two configurations and comply with the QCVN 09:2011/BGTVT standard.

1. INTRODUCTION

Air pollution and climate change are driving the trend toward emission reduction in transportation, particularly the shift from fossil fuel-powered vehicles to electric vehicles. In Vietnam, the goal of achieving net-zero emissions by 2050 has been concretized in Decision No. 867/QĐ-TTg. According to a study by Sri, Reddy (2023), electric vehicles consume only about 36% of the input energy, compared to 72% for internal combustion engine vehicles, demonstrating their superior energy efficiency.

To contribute to this goal, the authors conducted a study on converting a HONDA Civic from an internal combustion engine vehicle to an electric vehicle. The vehicle's structure and bodywork were retained, while the powertrain system was replaced with an electric motor. The vehicle's weight before and after the conversion is presented in Table 1.

Table 1. Vehicle mass data after conversion (Honda Motor Co., 2008)

Content	Weight
Original	No load
	1,210kg
Converted	Full load
	1,585kg
	No load
	1,507kg
	Full load
	1,882kg
Wheelbase	2,700 mm
Front /rear wheel tracks	1,500/1,530 mm

One of the critical technical factors to consider after vehicle conversion is the natural frequency of the suspension system. The natural frequency plays a key role in determining the ride comfort and stability of a vehicle during operation. The transition from a traditional internal combustion engine to an electric motor can significantly affect the suspension's natural frequency.

This impact can be evaluated using the following equations:

$$f = \frac{1}{T} = \frac{1}{2\pi} \sqrt{\frac{k}{m}} \quad (1)$$

Where: f The natural frequency (Hz), K The stiffness of the system (N/mm), m : The effective mass (kg).

Additionally, the static stiffness of the suspension system (C_t) can be determined based on the static load (Z_t) and the static deflection (f_t) using the following formula:

$$C_t = \frac{Z_t}{f_t} \quad (2)$$

According to Equation (1), the natural frequency depends on both the suspension stiffness and the vehicle mass, varying with the square root of these parameters. Equation (2) shows the relationship between load and wheel axle displacement, governed by the elastic characteristics of the suspension (N/mm). Any change in stiffness or load distribution results in a variation of the natural frequency, which can reduce ride comfort if it exceeds the acceptable threshold. Therefore, reassessing the suspension system after powertrain conversion is essential.

Numerous studies have evaluated the natural frequency based on the ISO 2631 standard, using the root mean square (RMS) acceleration to assess ride comfort. Le and Thao (2020), and Tran and Nguyen (2016), simulated the vibration behavior of sleeper buses and city buses at different speeds. Dam, Nguyen và Thuy (2021) focused on analyzing the impact of electric vehicle conversions, showing that suspension system design parameters significantly influence ride quality. Similarly, Bakar, Said and Aziz (2015) found that variations in load distribution altered axle movement, thus affecting the suspension's dynamic behavior.

Likewise, Thao studied the vibration of passenger transport vehicles through simulations conducted in MATLAB-SIMULINK, with evaluations based on TCVN 1694-1:2001. The author used the squared acceleration criterion to assess ride comfort, confirming that road surface conditions and suspension parameters significantly affect passenger comfort (Sri & Reddy, 2023).

However, in Vietnam, to be approved for road use, converted vehicles must meet the natural frequency requirement outlined in QCVN 09:2011/BGTVT, which specifies that the natural frequency must not exceed 2.5 Hz. This regulation provides two evaluation methods: Method 1 applies to two-axle vehicles, while Method 2 is intended for vehicles with more than two axles.

In this study, the authors adopted Method 1 for evaluation. A full-vehicle dynamic model was developed based on real-world measurements, including curb weight, gross vehicle weight, motor installation location, and post-conversion passenger load distribution. The suspension characteristics were integrated into the simulation model and calibrated. The vehicle was then subjected to a drop test from a height of 120 mm on a test rig, with a simulated impact force of 18,820 N used to collect vibration response data.

2. MATERIALS AND METHODS

2.1 Theoretical basis

The QCVN 09:2011/BGTVT standard specifies a detailed method for determining the natural frequency of the vehicle body. According to this standard, the procedure for identifying the natural frequency is carried out following the illustrated instructions in Figure 1. The obtained

natural frequency was then analyzed and evaluated based on formulas (3) and (4).

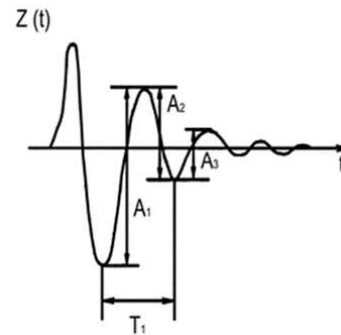


Fig. 1. The method for determining the natural oscillation period based on the results of the vehicle body's damping oscillation curve (National Technical Regulation QCVN 09:2011/BGTVT, 2011)

$$f_1 = \frac{1}{T_1} \quad (3)$$

$$T_1 = t_2 - t_1 \quad (4)$$

In there: f_1 : natural frequency of the vehicle body (Hz), T_1 : natural oscillation period of the vehicle body (s), t_2 : end time of the period being considered (s), t_1 : start time of the period being considered (s)

2.2 Simulation method

To determine the optimal battery configuration, the research team applied the National Technical Regulation QCVN 09:2011/BGTVT, which outlines two methods for measuring the natural frequency of a vehicle's suspension system. Specifically, Method 1 applied to two-axle vehicles and required the vehicle to be released freely from a height between 60 and 120 mm onto a test platform. Method 2, intended for vehicles with more than two axles, involved compressing the vehicle within the same height range and then releasing it to induce oscillation.

In this study, the authors selected Method 1 for simulation. A test platform was designed with a

height of 120 mm, and the suspension parameters were integrated and calibrated within the vehicle's dynamic model. The vehicle model was then released from the specified height with an applied impact force to simulate oscillation. The resulting simulation data was analyzed to determine the optimal battery configuration. The detailed dimensions and design of the test platform are shown in Figure 2.

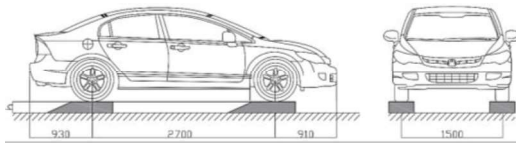


Fig 2. Technical specifications for creating oscillators according to method 1 (National Technical Regulation QCVN 09:2011/BGTVT, 2011)

Figures 3 to 6 illustrate the characteristics of the suspension system, which are then used as input data for the simulation model.

In Figure 3, the spring characteristic is illustrated through the force–displacement relationship. The curve is divided into three segments: OA, AB, and BC. In the first two segments (from point O to A and from A to B), the spring stiffness remains constant at $k=19.87\text{N/mm}$, indicating a linear relationship between force and displacement. However, from point B to C, the curve shows a nonlinear change as the slope increases, corresponding to a new stiffness of $k=23.18\text{N/mm}$. This change results from the uneven coil pitch design of the spring, which is intended to adjust the elastic characteristics to better suit load conditions during vehicle operation.

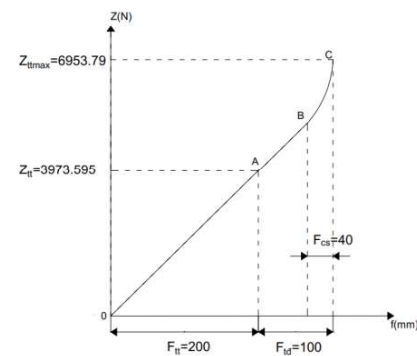


Fig 3. Characteristics of front suspension spring elements

Figure 4 shows the relationship between velocity and force represents the characteristic of the damping element. The horizontal axis indicates the displacement velocity, while the vertical axis represents the force. An analysis of this figure shows that the damping characteristic is divided into two parts, corresponding to the first and third quadrants of the coordinate plane. In the first quadrant, representing the rebound (return) stroke, the maximum damping force reaches 3926.99 N at a velocity of 50 mm/s. Meanwhile, in the third quadrant, representing the compression phase, the maximum damping force is 455.601 N, also at a velocity of 50 mm/s.

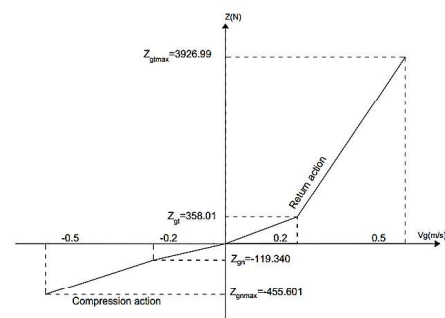


Fig 4. Characteristics of front suspension damper elements

In Figure 5, the spring characteristic is represented by the force–displacement curve, with the horizontal axis indicating displacement and the vertical axis indicating force. The rear suspension spring exhibits a linear response with

a constant stiffness of $k=16.6$ N/mm. This means it follows Hooke's law: $F=kx$, where F is the force (N), x is the displacement (mm), and k is the stiffness (N/mm). Thus, the equation becomes $F=16.6x$, meaning the force increases by 16.6 N for each millimeter of displacement.

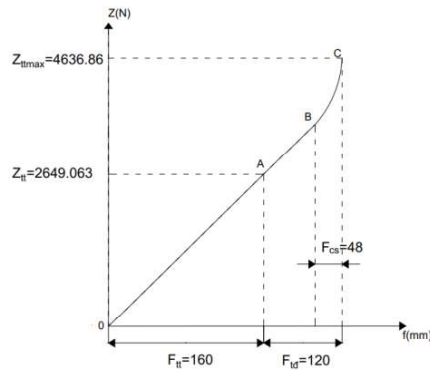


Fig 5. Characteristics of rear suspension spring elements

Figure 6 illustrates the relationship between velocity and force, representing the characteristic of the damping element. The damping behavior is divided into two parts: in the first quadrant, the maximum damping force during the rebound stroke reaches 2650.72 N at a velocity of 50 mm/s, while in the third quadrant, the maximum damping force during the compression stroke is 293.54 N at the same velocity.

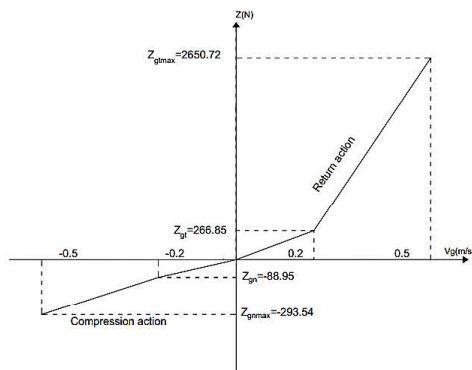


Fig 6. Characteristics of the damping elements of the rear suspension system

In the next step, the suspension characteristics from the Honda Civic were implemented in the

simulation model and retained for the converted vehicle version. The detailed parameters are summarized in Table 2.

Table 2. Set Up suspension simulation conditions

Content		Setting
Front suspension	Spring characteristics	K=19.9N/mm
	Damping properties	<pre> [UNITS] LENGTH = 'mm' ANGLE = 'degrees' FORCE = 'newton' MASS = 'kg' TIME = 'second' \$- [CURVE] { vel force -500 -455.601 -200 -119.340 0.0 0.0 200 358.01 500 3926.99 } </pre>
Rear suspension	Spring characteristics	K=16.6N/mm
	Damping properties	<pre> [UNITS] LENGTH = 'mm' ANGLE = 'degrees' FORCE = 'newton' MASS = 'kg' TIME = 'second' \$- [CURVE] { vel force -500 -293.54 -200 -88.95 0.0 0.0 200 266.85 500 2650.72 } </pre>

After updating the suspension characteristics in the full vehicle dynamic model, the next step is to adjust the mass distribution. First, the wheelbase and front-rear track widths were modified according to the specifications presented in Table 1 and illustrated in Figure 7. Once the basic dimensions and vehicle mass were updated, the center of gravity coordinates were determined based on the load distribution between the front and rear axles, corresponding to each loading condition. Detailed information for the original vehicle is provided in Table 3, while the data for the converted vehicle is presented in Table 4.

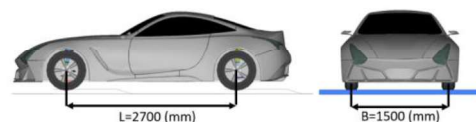


Fig 7. Basic vehicle specifications

After adjusting the base dimensions and weight of the vehicle, the coordinates of the

original vehicle's center of gravity were determined based on the load distribution on the front and rear axles for each load case, as shown in Table 4.

Table 3. Simulation parameters of the original vehicle

Full load	Weight of self	1,585 kg
	Front axle distribution	951 kg
	Rear axle distribution	634 kg
	Coordinates of the entire vehicle's center of gravity in the x direction	1,080 mm

In the previous study, the selected battery pack had dimensions of 1,816×1,362×234 mm. The battery was installed beneath the vehicle floor, with different placement configurations considered along the longitudinal direction from near to far relative to the front axle center.

In Case 1, with a wheelbase of 2,700 mm, the battery pack was positioned so that its center of gravity was located at the midpoint of the vehicle body, i.e., halfway between the front and rear axles. As a result, the battery load was evenly distributed between both axles.

In Case 2, the battery pack was shifted closer to the front axle, causing its center of gravity to move forward, resulting in the majority of the battery load being applied to the front axle.

Conversely, in Case 3, the battery was positioned toward the rear of the vehicle, concentrating more of the load on the rear axle.

The detailed load distribution for each configuration is provided in Table 4 and visually illustrated in Figure 8.

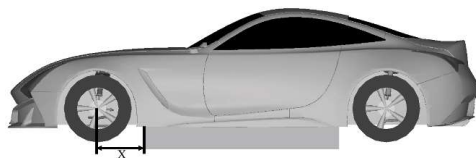


Fig 8. Battery installation location representation

Table 4. Battery installation layout scenarios converted vehicle

	Content	Case 1	Case 2	Case 3
Full load	Weight of self	1,882 Kg		
	Front axle distribution	1,098 Kg	1,185 Kg	1,082 Kg
	Rear axle distribution	784 Kg	697 Kg	800 Kg
	Coordinates of the entire vehicle's center of gravity in the x direction	1,124 mm	1,000 mm	1,148 mm
	Distance from front wheel center to battery	X=442 mm	X=292 mm	X=592 mm

Next, to collect acceleration data during the drop test, accelerometers were fixed to the vehicle body at two locations: one near the front axle and the other near the rear axle, with a distance of 2,700 mm between the two measurement points. This arrangement allows for accurate recording of acceleration at both ends of the vehicle during the simulation.

Finally, the vehicle model was placed on the test platform at a height of $h=120$ mm. A pushing force of 18,820 N was applied in the direction opposite to the x-axis. This force was determined based on the vehicle's total weight of 1,882 kg, simulating a scenario in which the vehicle engine is turned off and the vehicle is stationary but displaced from the platform, leading to a free-fall condition.

3. RESULTS AND DISCUSSION

3.1 The natural frequency of oscillation with the test bench height set at $h=120$ mm

Based on the acceleration measurement results of the Front axle in Fig. 9, we observe that one oscillation cycle is identified by two

consecutive peaks. The times $t_1=3.299\text{s}$ and $t_2=3.809\text{s}$. Using formula (1).

The natural frequency of the rear axle is determined to be $f=1.96\text{Hz}$. According to *QCVN09:2011/BGTVT*, the suspension system's natural frequency should not exceed $f=2.5\text{Hz}$, and the simulation results fall within this range. Therefore, the natural frequency of the front suspension system is within the allowable limit.

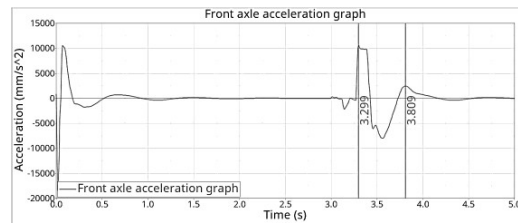


Fig 9. Front axle acceleration graph

Fig. 10 shows the natural frequency of the front suspension exhibits a discrepancy compared to the rear suspension, with a natural oscillation period of $T=0.5\text{s}$. However, the natural frequency of the Rear axle, $f=2\text{Hz}$, falls within the permissible limits as per the regulations.

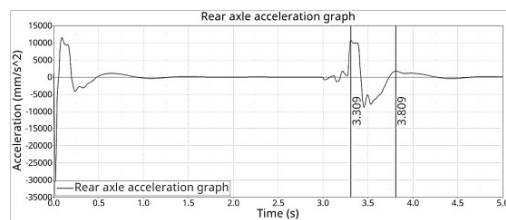


Fig 10. Rear axle acceleration graph

3.2 The natural frequency of oscillation with the test bench height set at $h=120\text{ mm}$ in case of changing battery installation location

For Case 1, the battery position is fully described in Table 4. The simulation results show that for the front axle, as depicted in Figures 11 and 12, the frequency simulation results for the rear axle are provided.

Fig. 11 shows the natural frequency of the front suspension exhibits a discrepancy compared

to the front suspension, with a natural oscillation period of $f=2.27\text{Hz}$.

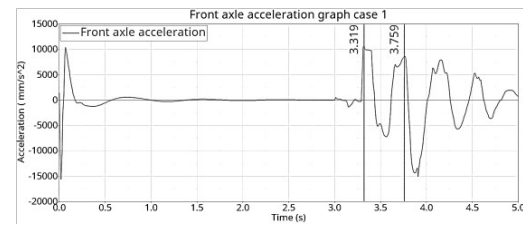


Fig 11. Front axle acceleration graph Case 1

Fig. 12 shows the natural frequency of the front suspension exhibits a discrepancy compared to the rear suspension, with a natural oscillation period of $f=1.45\text{Hz}$.

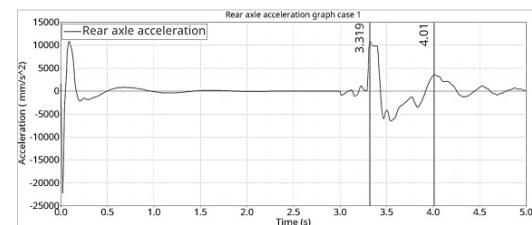


Fig 12. Rear axle acceleration graph Case 1

The natural frequency simulation results for Case 2 are presented in Figures 13 and 14.

Fig. 13 shows the natural frequency of the front suspension exhibits a discrepancy compared to the front suspension, with a natural oscillation period of $f=2.32\text{Hz}$.

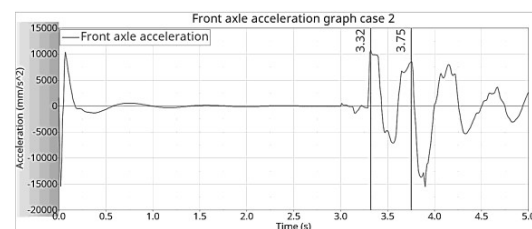


Fig 13. Front axle acceleration graph Case 2

Fig. 13 shows the natural frequency of the rear suspension exhibits a discrepancy compared to the rear suspension, with a natural oscillation period of $f=1.47\text{ Hz}$.

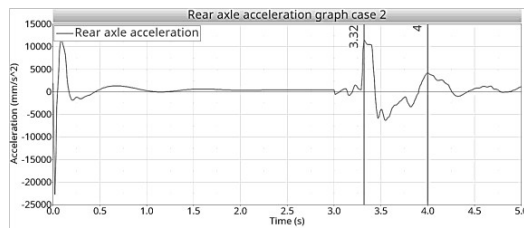


Fig 14. Rear axle acceleration graph Case 2

The natural frequency simulation results for Case 3 are presented in Fig. 15 and Fig. 16.

Based on the acceleration measurement results of the Front axle in Fig. 15, we observe that one oscillation cycle is identified by two consecutive peaks. The times $t_1=3.32\text{s}$ and $t_2=3.78\text{s}$. The natural frequency of the front axle is determined to be $f=2.17\text{Hz}$.

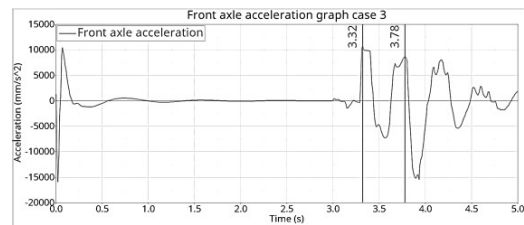


Fig 15. Front axle acceleration graph Case 3

Based on the acceleration measurement results of the rear axle in Fig. 16, we observe that one oscillation cycle is identified by two consecutive peaks. The times $t_1=3.32\text{s}$ and $t_2=4.02\text{s}$. The natural frequency of the front axle is determined to be $f=1.43\text{Hz}$.

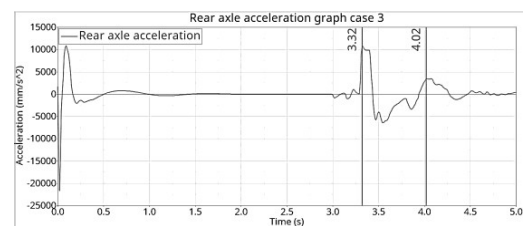


Fig 16. Rear axle acceleration graph Case 3

Simulation results show that retaining the original chassis structure when converting to an electric vehicle significantly affects the system's natural vibration frequencies. Specifically, the front axle frequencies in all three battery scenarios are higher than that of the original vehicle (1.96 Hz), with increases of 16% (Case

1), 18.4% (Case 2), and 10.7% (Case 3), respectively. In contrast, the rear axle frequencies are lower than the original (2.00 Hz), with reductions of 27.5%, 26.5%, and 28.5%, mainly due to the additional 297 kg weight from the battery, which alters the load distribution between axles. Moreover, battery clearly impacts vibration behavior: Case 2 (battery closest to the front axle) results in the highest front axle frequency, followed by Case 1, and the lowest in Case 3. In terms of vibration distribution, Case 3 is considered the most optimal.

4. CONCLUSION

This study analyzed the natural frequencies of a modified HONDA Civic 1.8L after being converted into an electric vehicle. The conversion process particularly the addition of the battery system significantly altered the vehicle's load distribution, thereby affecting its vibration frequencies. Simulations were carried out in accordance with QCVN 09:2011/BGTVT, evaluating three different battery configurations. Results showed that the front axle's natural frequency increased by 10.7% to 18.4% (from the original 1.96 Hz), while the rear axle's frequency decreased by 26.5% to 28.5% (from 2.00 Hz). Despite these changes, all values remained within the standard's allowable limit of 2.5 Hz. Among the configurations, Case 3 where the battery was positioned furthest from the front axle produced the most balanced results (2.17 Hz at the front axle, 1.43 Hz at the rear), and was identified as the optimal layout. This setup helped reduce vibration and ensured compliance with regulatory requirements. Overall, the findings highlight the critical role of battery placement in managing load distribution and vibration characteristics in EV conversions. This research contributes to

Vietnam's net-zero emissions target by 2050 (Decision No. 867/QĐ-TTg). Building upon the optimized battery placement determined in this study, the next phase of research will focus on further evaluating the vehicle's dynamic behavior and ride comfort. This will be carried out through *ISO 2631* based simulations using road surface profiles specified in Decree No. 116/2017/NĐ.

5. ACKNOWLEDGMENTS

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