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Optimization of Suspension Systems for Electric Vehicles Using a Genetic Algorithm

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Abstract
Electric vehicles equipped with in-wheel motors are particularly vulnerable to vertical vibration because the increased unsprung mass and motor-induced excitation can simultaneously degrade ride comfort and road friendliness. This study proposes a weighted multi-objective optimization framework for electric vehicle suspension design using a Genetic Algorithm (GA). A quarter-car dynamic model was developed to capture the coupled effects of stochastic road excitation and vertical motor excitation under realistic operating conditions. The suspension stiffness and damping coefficients were selected as the design variables, while the root mean square vertical body acceleration, a_{wbz}, and the dynamic load coefficient (DLC) were incorporated into a unified objective function to achieve a balanced trade-off between ride comfort and pavement-friendly performance. The governing equations were implemented in Matlab/Simulink and integrated with the GA-based optimization procedure, with practical constraints imposed on suspension natural frequency and damping ratio. Under the condition of a vehicle traveling on an ISO Class B road surface at 80 km/h with motor excitation at 50 Hz, the optimized suspension parameters reduced a_{wbz} by 13.3% and DLC by 11.2% compared with the original configuration. These results demonstrate that the proposed approach can effectively improve ride comfort while simultaneously mitigating the dynamic wheel load transmitted to the road surface, thereby providing a practical basis for suspension parameter selection in in-wheel-motor electric vehicles.

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Keywords: Electric vehicle, suspension optimization, genetic algorithm, ride comfort, dynamic load coefficient, in-wheel motor

1. Introduction

Electric vehicle (EV) suspension design has evolved from a conventional ride-handling tuning problem into a coupled vibration-control and parameter-optimization problem. In particular, for distributed-drive and in-wheel-motor electric vehicles, the increase in unsprung mass and the presence of motor-originated excitation significantly alter the vertical dynamic characteristics of the vehicle. Earlier studies showed that these features can deteriorate ride comfort, road holding, and dynamic stability if the suspension is designed in the same way as that of conventional vehicles [1, 4]. To mitigate these adverse effects, researchers proposed in-wheel vibration absorbers, dynamic vibration-absorbing structures, and coupled vehicle-motor dynamic models, confirming that electric vehicle suspension design must simultaneously account for road-induced and motor-induced disturbances rather than considering road excitation alone [5, 10]. The studies of Le Van Quynh and co-authors form a relatively coherent line of development within this research direction. Their early work quantified the effect of the in-wheel motor suspension system on electric vehicle ride comfort [11]. This was followed by a transition toward semi-active and active suspension control strategies, including modified Skyhook control [12], PID-based active suspension evaluation [13], acceleration-driven damping for semi-active suspension [14], and LQR-based active suspension design for distributed-drive electric vehicles [15]. In parallel with control-oriented studies, this group also developed optimization-oriented works, including

genetic-algorithm-based suspension parameter design for improving both ride comfort and road friendliness [16], improved artificial fish swarm optimization for electric vehicle suspension parameters [17], Firefly-based PID optimization [18], particle-swarm-optimization-based PID tuning [19], and a recent numerical study clarifying the influence of stiffness and damping ranges on electric vehicle ride comfort [20]. Overall, these studies show a clear evolution from performance assessment to
control design and, more importantly, toward systematic optimization of suspension parameters and control gains in electric vehicles [11] – [20]. However, despite these advances, an important research gap still remains. A considerable portion of the existing literature focuses either on improving ride comfort through controller design or on suppressing in-wheel-motor vibration through structural or semi-active solutions [5 ,10], [12, 15], [18, 20]. In contrast, fewer studies formulate the electric vehicle suspension design problem explicitly as a weighted multi-objective optimization problem in which ride comfort and road friendliness are minimized simultaneously at the parameter-design stage. This distinction is critical. If only the body acceleration is minimized, the suspension may become favorable in terms of subjective comfort but unfavorable in terms of tire dynamic load and road friendliness. Conversely, if only tire-load-related indices are emphasized, ride comfort may be sacrificed. This trade-off becomes even more pronounced in electric vehicles, especially those equipped with in-wheel motors, because the increased unsprung mass and electromechanical excitation intensify the conflict between comfort and dynamic tire loading [1, 10]. Therefore, this study addresses the weighted multi-objective optimization of EV suspension parameters by integrating the RMS vertical body acceleration and the dynamic load coefficient (DLC) into a single weighted objective function. The suspension stiffness and damping coefficients are then optimized to achieve a rational trade-off between ride comfort and road friendliness while meeting practical constraints on suspension working space, dynamic tire load, natural frequency, and damping ratio. Such a formulation is consistent with the vertical dynamic characteristics of electric vehicles and provides a more rigorous engineering basis for suspension parameter selection under realistic operating conditions.

2. Electric dynamic model

2.1. Quarter-car dynamic model of the electric vehicle

To describe the dominant vertical vibration behavior of the electric vehicle while maintaining a relatively simple structure for parameter optimization, a quarter-car dynamic model is adopted in this study, as shown in Fig. 1. The model represents one corner of the vehicle and consists of the sprung mass, the unsprung mass, the electric motor mass, the suspension spring-damper pair, the tire stiffness-damping pair, and the road excitation input. This model is widely used for suspension analysis because it can adequately capture the main vertical dynamic characteristics associated with ride comfort, suspension working space, and dynamic tire load.

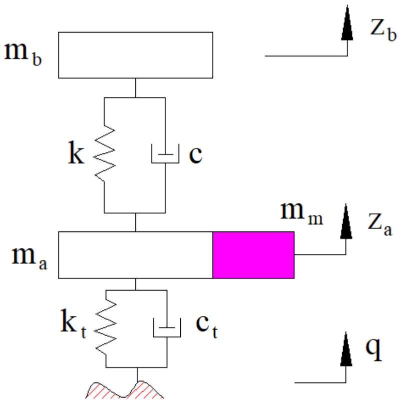

Fig 1: Quarter-car dynamic model of the electric vehicle

In the proposed model, m_b denotes the sprung mass corresponding to one quarter of the vehicle body, including the equivalent body and payload mass supported by the suspension. The quantity m_a represents the mechanical unsprung mass associated with the wheel assembly, while m_m denotes the mass of the electric motor mounted close to or integrated with the wheel. Since the motor is rigidly connected to the wheel-side assembly, the effective unsprung mass can be expressed as

$$m_u = m_a + m_m$$

The variables used in the model are defined as follows: m_u is the equivalent unsprung mass; k and c are the suspension stiffness and damping coefficients; k_t and c_t are the tire stiffness and damping coefficients; z_b and z_a are the vertical displacements of the sprung and unsprung masses; $q(t)$ is the road excitation input; Δz is the suspension deflection; and F_t is the dynamic tire force.

By applying Newton's second law to the sprung and unsprung masses, the equations of motion of the quarter-car electric vehicle model are obtained as follows:

$$m_b \ddot{z}_b = -\left[k\left(z_b - z_a\right) + c\left(\dot{z}_b - \dot{z}_a\right)\right]$$

$$m_u \ddot{z}_a = F_m + \left[k\left(z_b - z_a\right) + c\left(\dot{z}_b - \dot{z}_a\right)\right]$$
$$-\left[k_t\left(z_a - q\right) + c_t\left(\dot{z}_a - \dot{q}\right)\right]$$

$$m_b \ddot{z}_b = -\left[k\left(z_b - z_a\right) + c\left(\dot{z}_b - \dot{z}_a\right)\right]$$

where, F_m is the vertical excitation force of the in-wheel motor.

2.2. Excitation source model

In order to describe the dynamic response of the electric vehicle under actual operating conditions, both road-induced excitation and motor-induced excitation are taken into account in this study. The excitation source model therefore consists of two components: a stochastic road excitation model and a vertical excitation model of the in-wheel motor.

Road excitation model: To simulate the effect of random road irregularities, an excitation model derived from ISO 8608:2016 [21] is employed in this study. Under this approach, road roughness is described in terms of its spectral characteristics, and the corresponding road displacement excitation *q(t)* for the coupled vehicle–terrain system is expressed as

$$q(t) = \sum_{i=1}^{N} \sqrt{2G_d\left(n_i\right)\Delta n_i}\cos\left(2\pi i \Delta n t + \beta_i\right)$$

where, $G_d(n_i)$ denotes the power spectral density (PSD) of the road surface at spatial frequency n_i corresponding to ISO roughness classes from A (smooth) to H (very rough). The term Δn_i represents the frequency bandwidth, while β_i is the randomly assigned phase angle within the interval [0, π], ensuring a realistic representation of random surface profiles.
Electric motor excitation model: Yu *et al*. (2019) developed a nonlinear dynamic model for an electric vehicle equipped with an in-wheel motor drive system by taking into account the motor excitation force and the nonlinear bearing force [22]. Based on that model, the nonlinear bearing forces acting along the X- and Z-directions can be expressed as

$$F_x = \sum_{j=1}^{N} F_j \cos\theta_j$$

$$F_z = \sum_{j=1}^{N_b} F_j \sin\theta_j$$

where, θ_j is the angular position of the *j* th rolling element.
In the present study, only the vertical excitation generated by the motor is considered. Therefore, Eq. (6) can be simplified and rewritten as

$$F_{mz0} = m_s e\omega_R^2 \cos\omega_R t$$

where, m_s is the total mass of the tyre, the rim and the motor rotor; *e* is the eccentricity of the rotor; ω_R is the angular velocity of the rotor.

3. Suspension Parameter Optimization Using a Genetic Algorithm

In this study, a Genetic Algorithm (GA) was employed as a global optimization method to determine the optimal suspension parameters for the electric vehicle (EV) suspension system. The suspension design problem is nonlinear and subject to multiple technical constraints; therefore, GA is an appropriate choice because of its strong global search capability and robustness in handling constrained optimization problems. Each individual in the population represents a candidate suspension design, expressed by the design vector $X = [k, c]$. Through the processes of selection, crossover, and mutation, the algorithm iteratively improves the solution quality over successive generations until a satisfactory compromise between ride comfort and road friendliness is achieved.
To obtain the optimal suspension design, the GA minimizes a weighted objective function that simultaneously accounts for ride comfort and dynamic wheel load. Specifically, the optimization problem is formulated as

$$F\left(X\right) = w_1\left\{a_{wbz}(X)\right\} + w_2\left\{DLC(X)\right\} \rightarrow \min$$

$$s.t\begin{cases} X = \left[k, c\right] \\ \Delta z = \left|\left(z_b - z_a\right)\right| \le 0.027 \\ F_{t,rms} \le 1192 \\ 1.0 \le \dfrac{1}{2\pi}\sqrt{\dfrac{k}{m_b}} \le 2.0 \\ 0.2 \le \dfrac{c}{2\sqrt{m_b k}} \le 0.4 \end{cases}$$

where w_1 and w_2 are the weighting coefficients of the two objective components ($w_1 + w_2 = 1$); $F_{t,rms}$ is the root mean square of the vertical dynamic tire force. The last two inequalities define the allowable ranges of the suspension natural frequency and damping ratio, respectively, thereby ensuring the technical feasibility of the optimized design.
To evaluate ride comfort in the time domain, the frequency-weighted root mean square (RMS) vertical acceleration defined in ISO 2631-1:1997 [23] was adopted. This index reflects the cumulative effect of vibration exposure over the measurement interval and is widely used for assessing vehicle ride quality. The weighted acceleration is expressed as

$$a_{wbz} = \left[\frac{1}{T}\int_0^T a_z^2(t)dt\right]^{1/2}$$

where, $a_{wbz}(t)$ is the weighted acceleration (translational and rotational) as a function of time, m/s^2; *T* is the duration of the measurement, s.
In addition to ride comfort, the dynamic load coefficient (DLC) was used to quantify the variation of the tire dynamic load acting on the road surface. This index is widely adopted to evaluate the road-damaging effect induced by wheel load fluctuations, since excessive dynamic tire loads may accelerate pavement deterioration even when the static wheel load remains unchanged. The DLC is defined as [25]–[27]:

$$DLC = \frac{F_{t,rms}}{F_{t0}}$$

where, $F_{t,rms}$ is the root mean square of the vertical dynamic tire force and, F_{t0} is the corresponding static tire load.
Accordingly, the suspension parameter optimization problem was solved by minimizing both a_{wbz} and *DLC* in a weighted manner. This formulation allows the optimized suspension to improve vehicle ride comfort while simultaneously reducing dynamic wheel load transfer to the pavement, thereby achieving a balanced performance from the perspective of both passenger comfort and road protection.

4. Results and discussion

The equations governing the system motion, presented in Section 2, were implemented and simulated in the Matlab/Simulink environment to evaluate the objective functions used in the optimization process. On this basis, the Genetic Algorithm (GA)-based optimization program was integrated with the simulation model to determine the optimal design parameters of the electric vehicle suspension system. The GA parameters, such as population size, number of generations, crossover probability, and mutation probability, were selected appropriately based on the reference data reported in Ref. [24]. In this study, the optimization process was carried out through the integration of the suspension dynamic model and the GA, allowing the ride comfort and dynamic wheel load criteria to be evaluated simultaneously under the condition that the in-wheel electric motor generated vibration excitation according to Eq. (7) at a frequency of 50 Hz, while the vehicle was traveling on an ISO Class B road surface at a speed of v = 80 km/h.

The obtained optimal parameters are presented in Table 1. The results indicate that, with the optimized parameter set, the values of a_{wbz} and *DLC* were reduced by 13.3% and 11.2%, respectively, compared with those of the original vehicle configuration. These reductions demonstrate that the optimized suspension parameters improve both ride comfort and the capability to mitigate the dynamic load transmitted to the road surface.

Table 1: Comparison of suspension parameters before and after optimization

Parameter	Initial value	GA-optimized value	Before optimization	After optimization	Difference (%)
k/(N/m)	22000	11330	-	-	-
c/ (N·s/m)	1218	1401	-	-	-
a_{wbz} /(m/s^2)	-	-	0.328	0.284	13.3
DLC	-	-	0.049	0.0435	11.2

The time histories of the vehicle body vertical acceleration and the vertical dynamic tire forces at the front and rear wheels are presented in Figs. 2 and 3, respectively. As observed from these figures, the peak amplitudes of the body vertical acceleration a_{bz} and the dynamic tire force F_t decrease noticeably after optimization. This confirms that the optimized design parameters not only enhance vehicle ride comfort but also reduce the fluctuation of the dynamic wheel load acting on the pavement, thereby improving the overall performance of the suspension system.

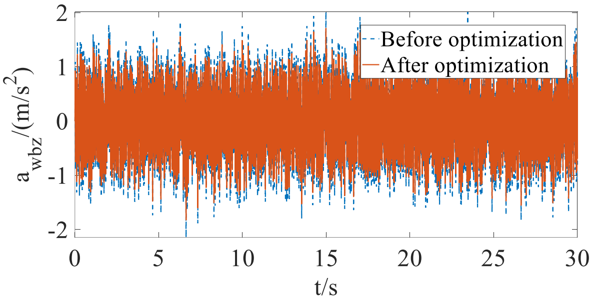

Fig 2: Comparison of vehicle body vertical acceleration before and after optimization

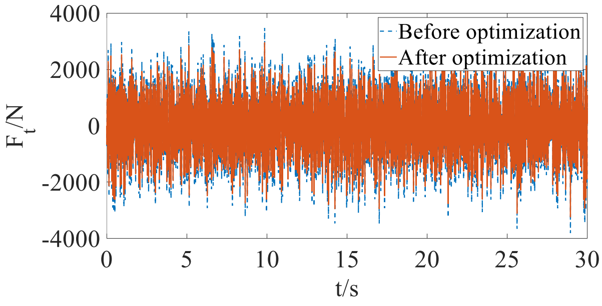

Fig 3: Comparison of dynamic wheel load on the road surface before and after optimization.

4. Conclusions

This study presented a weighted multi-objective optimization of suspension parameters for an electric vehicle using a Genetic Algorithm (GA). A quarter-car dynamic model was developed by considering both stochastic road excitation and vertical in-wheel motor excitation, and the governing equations were implemented in Matlab/Simulink to evaluate the suspension performance under realistic operating conditions. The suspension stiffness and damping coefficients were optimized by simultaneously minimizing the frequency-weighted vertical body acceleration, $\boldsymbol{a_{wbz}}$, and the dynamic load coefficient (DLC), subject to practical constraints on suspension natural frequency and damping ratio. Under the condition of an ISO Class B road surface, a vehicle speed of 80 km/h, and motor excitation at 50 Hz, the optimized suspension parameters reduced $\boldsymbol{a_{wbz}}$ by 13.3% and DLC by 11.2% compared with the original configuration. In addition, the time-domain responses showed noticeable reductions in the peak amplitudes of vehicle body vertical acceleration and dynamic wheel load after optimization. These results confirm that the proposed optimization approach can effectively improve ride comfort while simultaneously reducing the dynamic load transmitted to the road surface. Therefore, the present study provides a practical and effective basis for suspension parameter selection in electric vehicles equipped with in-wheel motors. Future work will focus on extending the proposed approach to more complex full-vehicle models and experimental validation under different operating conditions.

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