

Hysteresis Modeling of Automotive Electrohydraulic Semi-Active Dampers Using Tangent Functions and Simulation-Based Ride Comfort Evaluation [†]

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Abstract

This study develops a hyperbolic tangent-based model for the hysteretic behavior of automotive grade electrohydraulic semi-active dampers. Model parameters were identified from experimental force–velocity data gathered under sinusoidal excitations across 1–6 Hz and 0.38–1.6 A. The calibrated model was integrated into an IPG CarMaker 13.0/Simulink 2022b co-simulation to assess performance under ISO-compliant road profiles and realistic driving scenarios. Comparative analysis with conventional nonlinear damper models was conducted, focusing on ride comfort metrics such as vertical acceleration, pitch rate, and roll rate. The results demonstrate that the proposed model provides improved fidelity in replicating real damper behavior and enables more realistic assessment of semi-active suspension performance in virtual vehicle development platforms by providing reduced vertical acceleration errors by >5 dB (2–6 Hz) compared to nonlinear models.

Keywords: hysteresis; shock absorber; genetic algorithm

1. Introduction

Customer expectations for vehicle ride comfort drive ongoing innovation in suspension technology. Semi-active suspension systems have been developed and used in the automotive industry and remain a focal point of automotive research for over four decades [1,2]. These systems employ controllable dampers, most commonly magnetorheological (MR) or electro-hydraulic (EH) types, to modulate the damping force in real time. While MR units achieve faster response times (8–20 ms) by altering fluid viscosity via magnetic fields [3,4], EH dampers operate on passive-hydraulic principles, displacing fluid between chambers during piston movement. Dedicated EH modules typically respond slower than MR units [5,6], with off-the-shelf proportional valves (e.g., HydraForce SP10-24, Illinois, USA) exhibiting full open times near 40 ms [7]. EH dampers regulate flow through fast-acting valves, enabling continuous damping forces (1–6 kN) suitable for automotive applications.

However, internal valve hysteresis and asymmetric flow losses between compression and rebound strokes result in nonlinear force–velocity loops with pronounced hysteresis, complicating precise ride-comfort control [8]. Accurate modeling thus requires comprehensive understanding of the system. Hysteresis modeling approaches are generally split into



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parametric approaches (model-driven) and data-driven (black-box) methods [9,10]. Parametric models may be empirical (relying on curve fitting functions such as polynomial or splines) or semi-empirical/phenomenological, such as Bouc–Wen variants, and hyperbolic tangent formulations [11]. Data-driven methods ignore any predefined physical structure and instead use machine learning tools (most commonly artificial neural networks) to learn hysteresis behavior directly from measurements [12]. Nonlinear optimization algorithms, such as PSO, GA, and related metaheuristics, have been widely used to identify the parameters of both empirical and semi-empirical models under various operating conditions [11,13]. In the literature, most hysteresis-modeling research focuses on MR dampers, originally developed for seismic applications. Studies targeting the hysteresis behavior of production-grade EH dampers remain scarce; a rare parametric example by Wang & Hou [14], lacks validation at high-frequencies (>20 Hz). Semi-empirical Bouc–Wen models require integrating an internal hysteretic state variable, increasing complexity compared to algebraic hyperbolic-tangent approaches, that directly approximate loop shapes [11,13].

This paper addresses the gap in validated EH damper modeling by developing a hyperbolic-tangent hysteresis model for an automotive-grade EH damper. Key parameters were identified via genetic-algorithm (GA) optimization using bench-test data across operational frequencies (1–6 Hz) and solenoid currents (0.38–1.6 A). Selected hyperbolic-tangent parameters were extended beyond simple linear scaling to capture current-dependent nonlinearities. GA proved particularly suited for handling discontinuous force jumps induced by current steps [15]. The resulting model, validated in an IPG CarMaker/Simulink co-simulation environment with ISO 8608:2021 road profiles [16]. Ride-comfort evaluation via spectral analysis of vertical acceleration demonstrates that unlike ideal actuator representations, the hysteresis-capable damper model reveals significant deviations in the 1–6 Hz band, enabling more realistic virtual assessment of semi-active suspension performance.

2. Electro-Hydraulic Damper

Electro-hydraulic (EH) dampers operate on principles similar to passive hydraulic dampers utilizing controlled fluid displacement between compression and rebound chambers via piston movement (Figure 1). Hydraulic fluid flows through the piston, base valve, and solenoid regulated orifices, where the latter modulates flow area in response to input current. Key design elements, including piston geometry, valve configurations, and chamber volumes, collectively defines the damper's force–velocity characteristics.

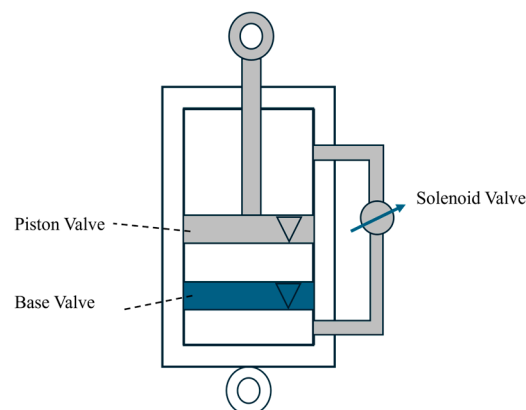


Figure 1. Electro-hydraulic damper schematic.

To optimize handling, EH dampers exhibit higher damping aggressiveness (steep force–velocity slope) than at high speeds, a characteristic which represents a symmetry across compression and rebound quadrants. Due to its inherent characteristics and continu-

ous fluid displacement, an EH damper exhibits nonlinear force–velocity characteristics and pronounced hysteresis behavior [8].

3. Hysteresis Modeling Approaches

3.1. Bouc–Wen Model

Following Bouc’s original 1967 differential hysteresis model [17] and Wen’s 1976 Galerkin-based extension for random vibrations [18], the Bouc–Wen formulation remains a standard parametric tool for modeling hysteresis in mechanical systems. Unlike purely algebraic approaches, it incorporates a first-order differential equation for the internal hysteresis variable z (see Equation (1)), and combines this with a restoring force law (Equation (2)). α scales the contribution of the hysteretic state z to the total force, c and k are viscous damping and linear spring stiffness coefficients, respectively. δ , β , and γ (along with exponent n) control the smoothness, geometric shape, and width of the hysteresis loop. $\dot{x}$ and x are the velocity and displacement inputs, which were combined in Equation (1).

$$\dot{z} = \delta \dot{x} - \beta |\dot{x}| |z|^n - \gamma z |\dot{x}| |z|^{n-1} \quad (1)$$

$$f = c \dot{x} + kx + \alpha z + f_0 \quad (2)$$

3.2. Hyperbolic Tangent Model

The hyperbolic tangent model, originally proposed by Kwok et al. [11], is a simple alternative to the Bouc–Wen model. It represents hysteresis using a straightforward algebraic function that combines a damping coefficient, spring stiffness, an offset parameter, and a hysteretic term in Equation (3). The hysteretic behavior is shaped by parameters such as β , δ , α and the sign function of the displacement x in Equation (4). Figure 2 illustrates the model conceptually. Unlike the Bouc–Wen formulation, which relies on an internal differential equation for the hysteretic state, the tangent model does not require integration, making it computationally efficient and easier to identify parameters for implementation in control algorithms.

$$f = c \dot{x} + kx + \alpha z + f_0 \quad (3)$$

$$fz = \tanh(\beta \dot{x} + \delta \operatorname{sign}(x)) \quad (4)$$

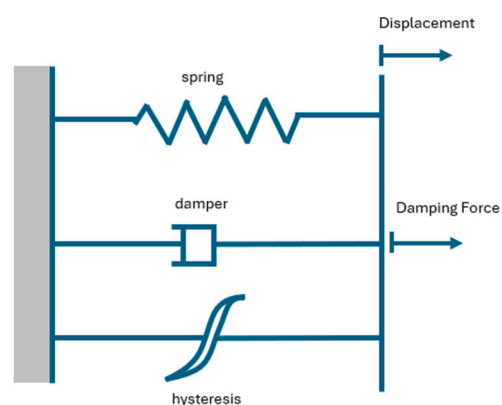


Figure 2. Hysteresis model.

4. Experimental Setup

Bench tests were performed using a CTW Automation RD20 system, CTW Automation, Lexington, NC, USA (Figure 3) with a Renault D-Segment EH damper, Tenneco, Valencia, Spain (± 25 mm stroke). Sinusoidal excitations (1–6 Hz) were applied while regulating solenoid currents (380, 900, 1200, 1600 mA) via a PI-controlled PWM signal. Force and

displacement data were sampled at 1 kHz, with the ambient temperature stabilized at 20 °C throughout the tests.

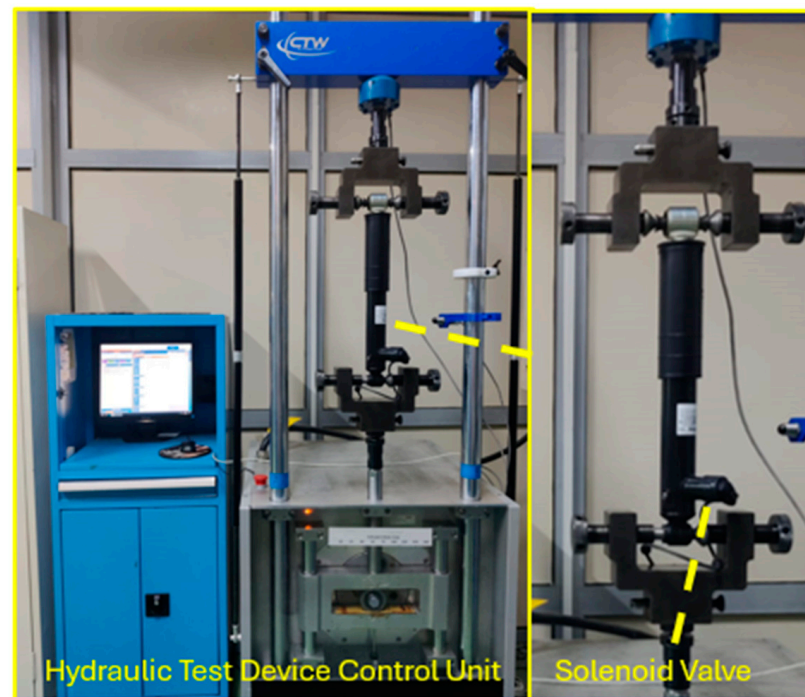


Figure 3. CTW hydraulic test device and control unit-left, detailed view of shock absorber on the test bench-right.

5. Parameterization of the Extended Tangent Model Using Genetic Algorithm

The hyperbolic tangent model (Equations (3) and (4)) requires precise parameter identification to accurately capture EH hysteresis. Genetic Algorithm (GA) is a stochastic optimization technique inspired by natural selection and genetics. It evolves a population of candidate solutions through selection, crossover, and mutation to explore complex search spaces and identify optimal solutions. Owing to its global search capability and robustness against local minima, GA has been widely adopted in engineering problems, including nonlinear system identification and parameter estimation of hysteretic models [19]. In this study, GA is employed to estimate the parameters of nonlinear hysteresis model, c_{pos} , $|k|$, $|\beta|$, $|\delta|$, $|f_0|$, $|c_{neg}|$, $|\alpha_i|$ ($\alpha_i = \alpha_i(\dot{x})$, for $i = 1, 2, \dots, 7$), where the objective function minimizes the root mean square error (RMSE) between measurement and simulated force responses in Equation (5).

$$RMSE = \sqrt{\frac{1}{N} \sum_{i=1}^N (F_{measured,i} - F_{est,i})^2} \quad (5)$$

where N is the number of data points, and $F_{measured}$, F_{est} denote measured and estimated forces, respectively. GA was employed to estimate the model parameters. Optimization was performed across all tested frequencies (1–6 Hz) and currents (0.38–1.6 A) using a population size of 1500 and generations up to 1600. Results in Figures 4–9 demonstrate the identified parameters under varying operational conditions.

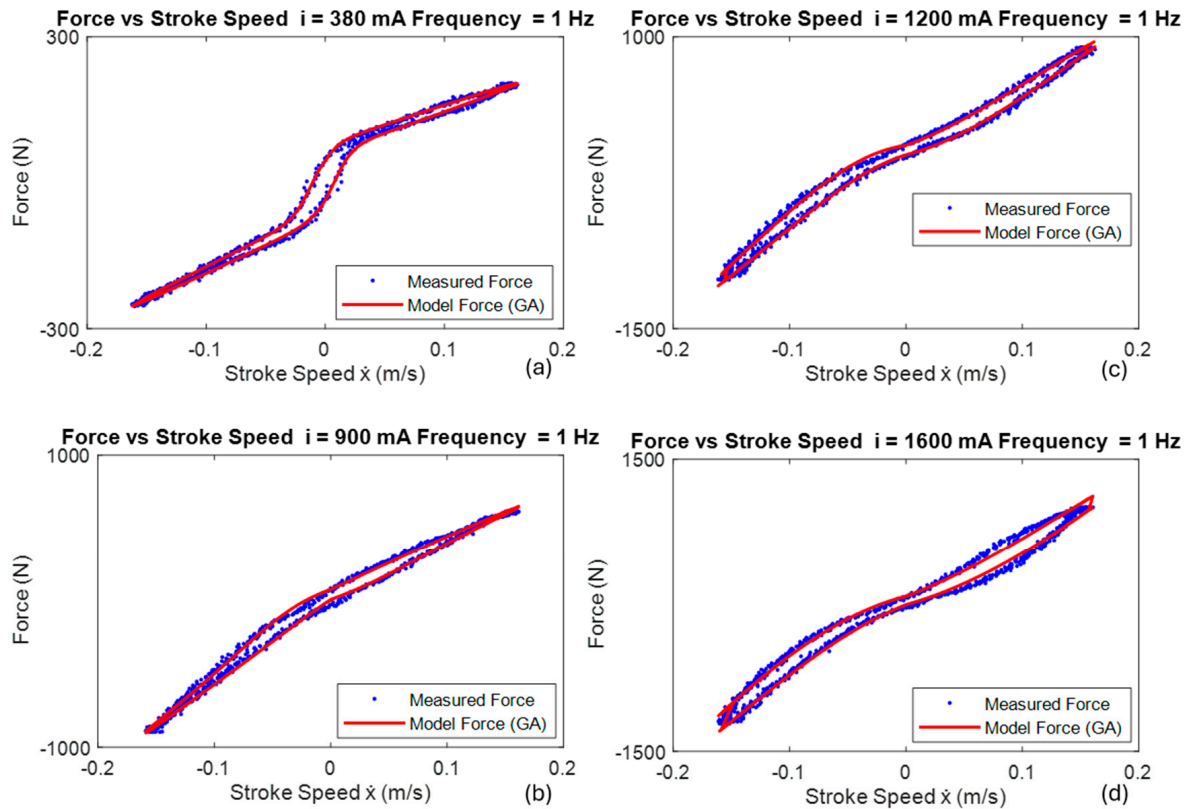


Figure 4. Parameter identification under 1 Hz excitation of shock absorber displacement ± 25 mm for (a) 380 mA, (b) 900 mA, (c) 1200 mA, (d) 1600 mA.

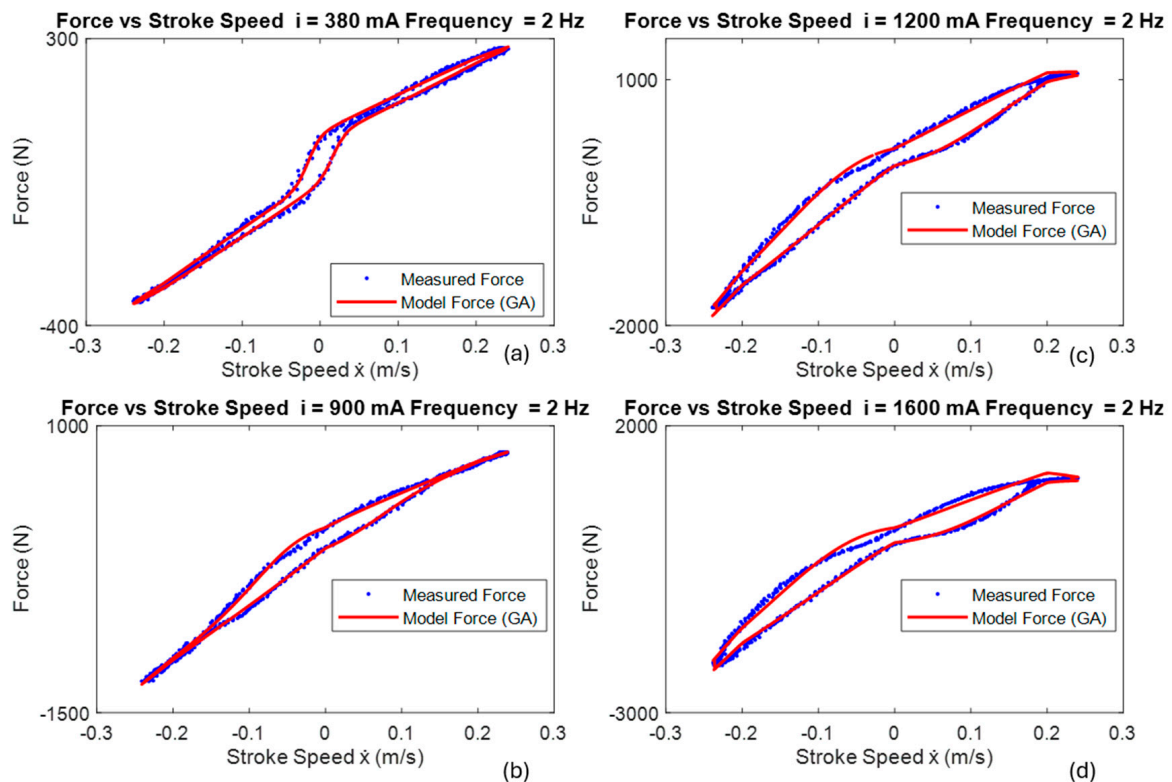


Figure 5. Parameter identification under 2 Hz excitation of shock absorber displacement ± 25 mm for (a) 380 mA, (b) 900 mA, (c) 1200 mA, (d) 1600 mA.

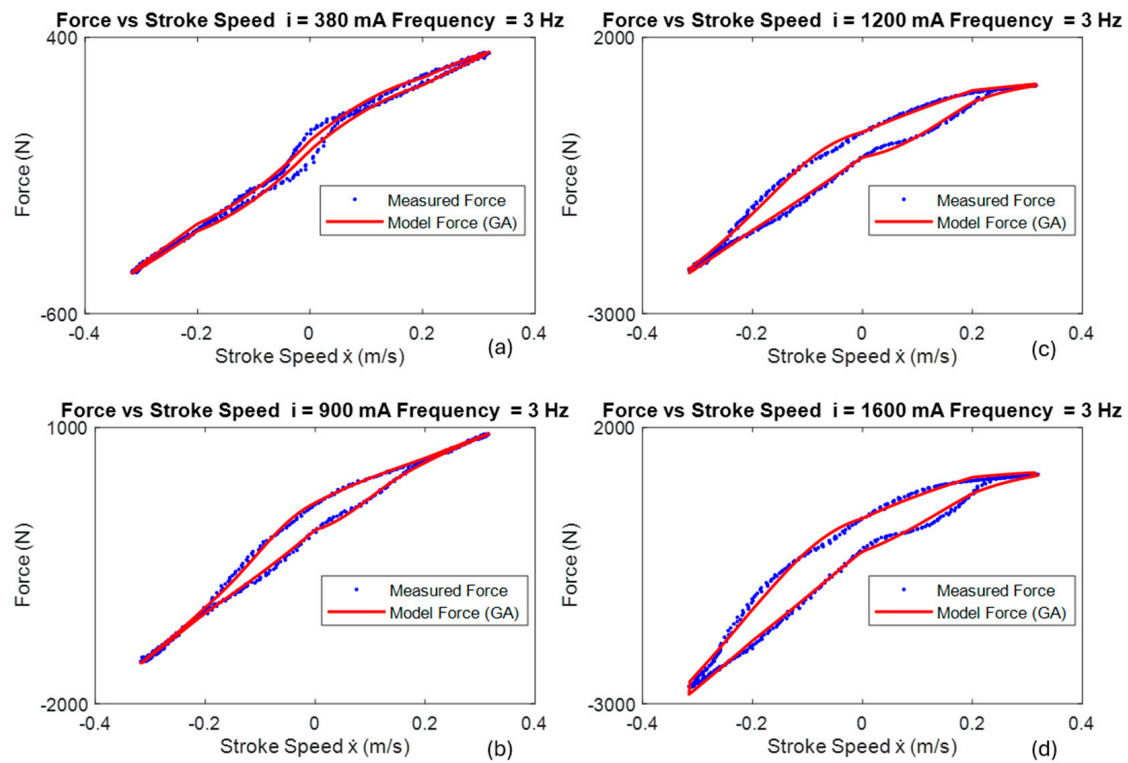


Figure 6. Parameter identification under 3Hz excitation of shock absorber displacement ± 25 mm for (a) 380 mA, (b) 900 mA, (c) 1200 mA, (d) 1600 mA.

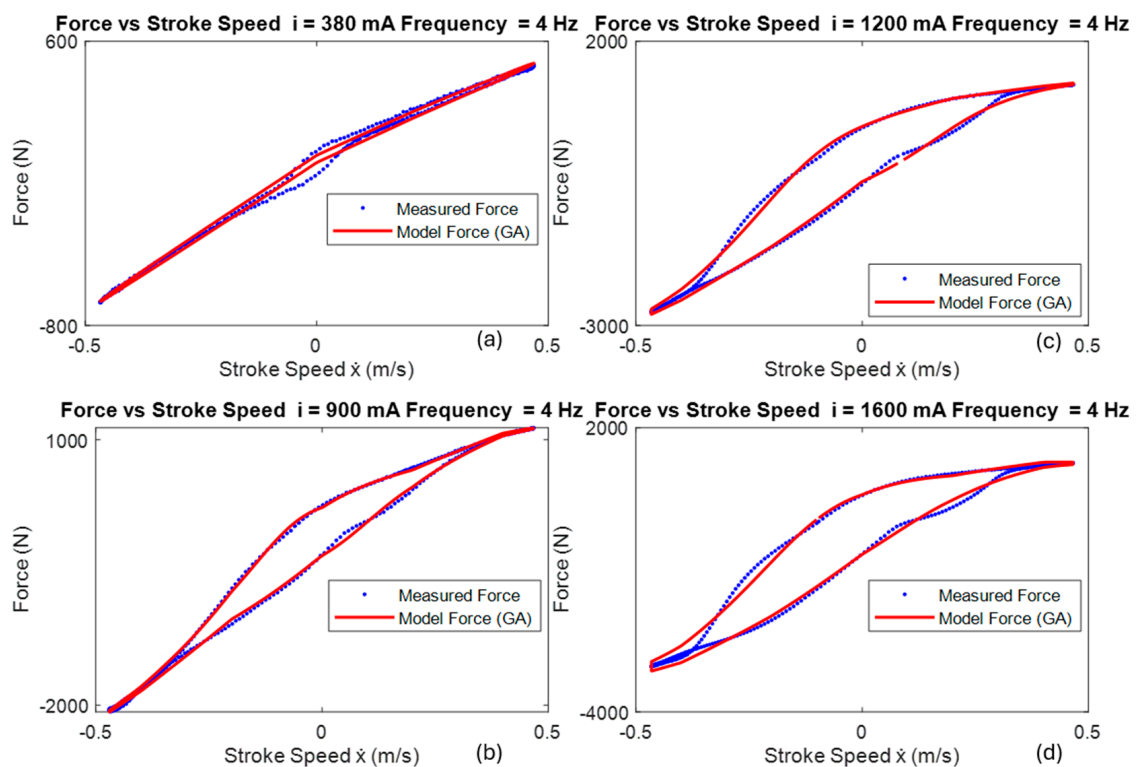


Figure 7. Parameter identification under 4Hz excitation of shock absorber displacement ± 25 mm for (a) 380 mA, (b) 900 mA, (c) 1200 mA, (d) 1600 mA.

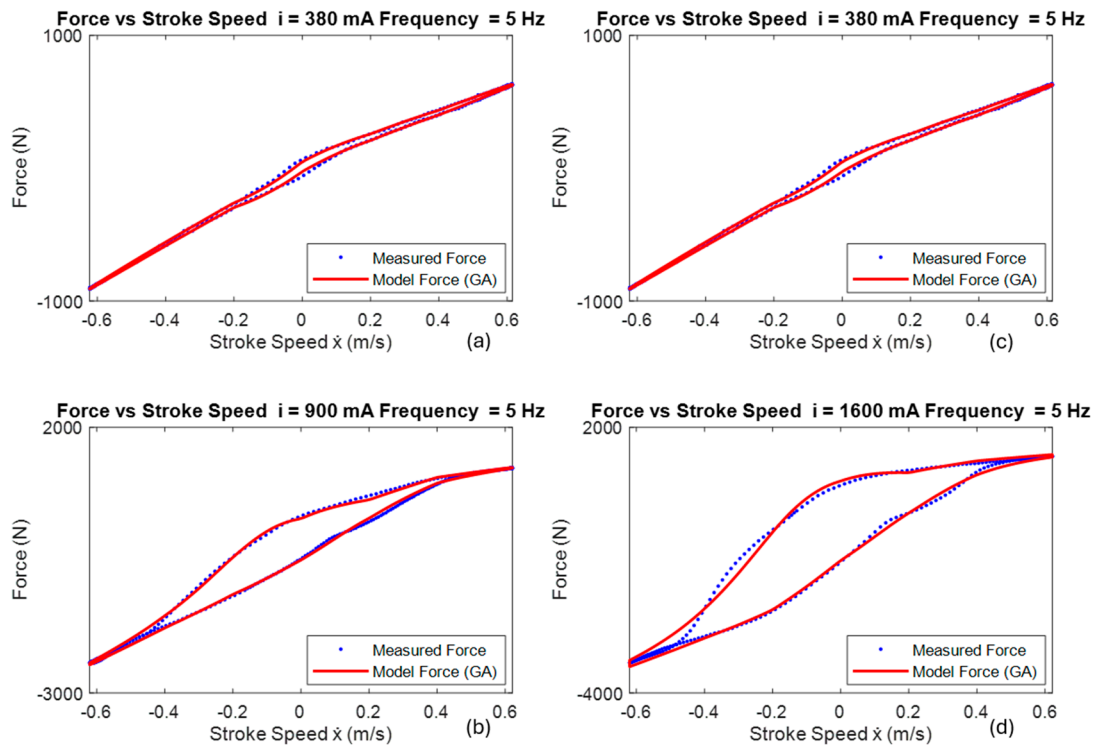


Figure 8. Parameter identification under 5Hz excitation of shock absorber displacement ± 25 mm for (a) 380 mA, (b) 900 mA, (c) 1200 mA, (d) 1600 mA.

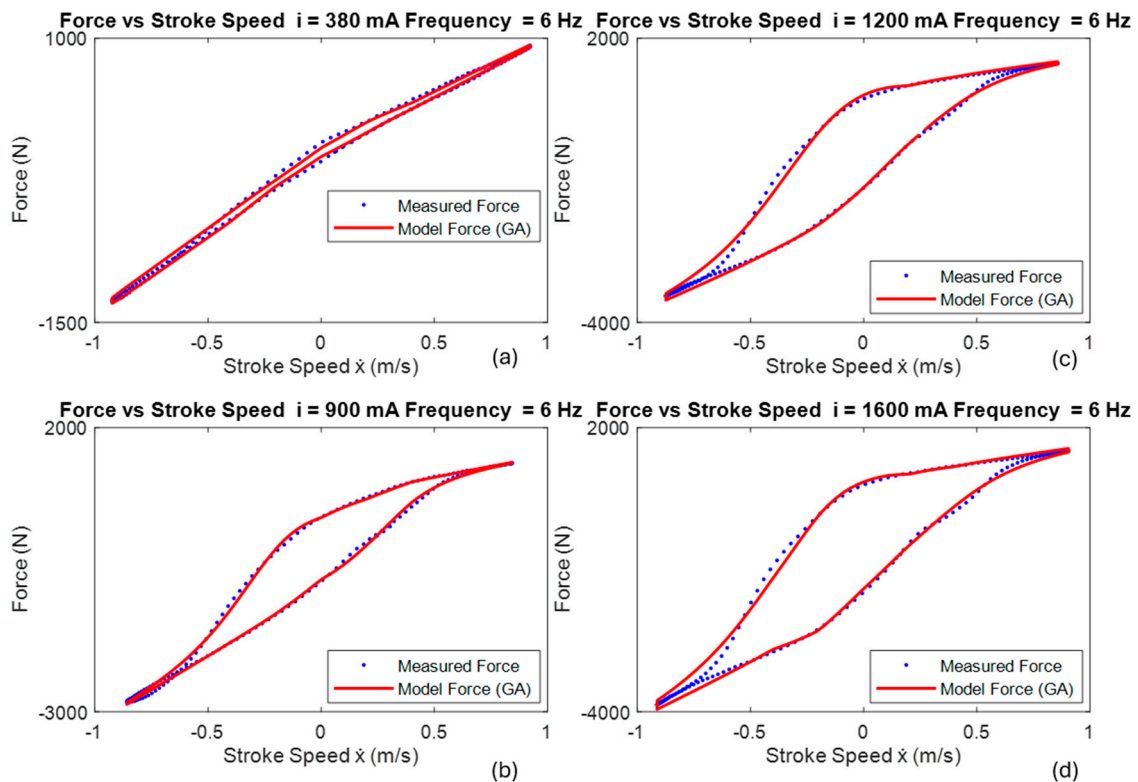


Figure 9. Parameter identification under 6Hz excitation of shock absorber displacement ± 25 mm for (a) 380 mA, (b) 900 mA, (c) 1200 mA, (d) 1600 mA.

The hysteresis loops in Figures 4–9 reveal diminishing geometric similarity in loop shapes as current and stroke velocity increase. This makes it challenging for the hyperbolic

tangent function's ability to represent system behavior across all operating conditions with a fixed parameter set. Therefore, parameterizing the effect of the current through linearly scaled GA-optimized coefficients proved insufficient. This highlights a fundamental divergence from MR dampers: accurate modeling requires additional parameters to explicitly map the geometric transformation of force–velocity curves under varying control inputs.

Automotive-grade dampers require asymmetric damping between compression and rebound regions due to design specifications: (c_{neg}) (c_{pos}) . Additionally, the shape parameter (α) was optimized across seven discrete stroke velocity levels to capture nonlinear transitions. Experimental noise reduced the model accuracy at higher frequencies compared to low-frequency domains, suggesting a potential need for higher dimensional parameterisation. Despite this, global RMSE remained below 50N across all operating conditions.

Moreover, when the optimized parameters are linearly parameterized with respect to the current, the resulting model exhibits significant errors. To resolve this, a mesh-based interpolation scheme was implemented: Optimized parameters were stored as 6×4 matrices (current $\times$ velocity) in a 1×13 cell array. During simulation, 2D linear interpolation (given in Equation (6)) based on current and stroke velocity was employed to dynamically extract the relevant parameters, which were then utilized within the hyperbolic tangent function (Figure 10). The resulting 384-parameter structure enables continuous hysteresis prediction across the operation envelope.

$$P(i, \dot{x}) = \frac{1}{(i_2 - i_1)(\dot{x}_2 - \dot{x}_1)} [(i_2 - i)(\dot{x}_2 - \dot{x})P_{11} + (i - i_1)(\dot{x}_2 - \dot{x})P_{21} + (i_2 - i)(\dot{x} - \dot{x}_1)P_{12} + (i - i_1)(\dot{x} - \dot{x}_1)P_{22}] \quad (6)$$

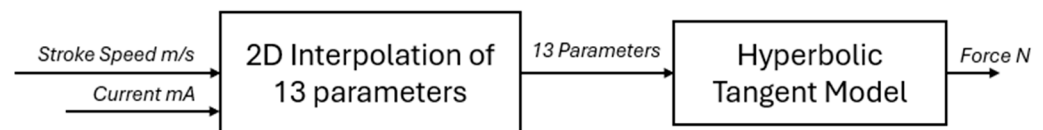


Figure 10. Mesh grid 2D interpolation input–output relationships.

6. Implementation and Comparative Analysis

6.1. Skyhook Control and Damper Model Implementation

Following the methodology proposed by Büyükköprü, et al. [20], a global skyhook controller was implemented to control the semi-active dampers in the simulation environment, with the corresponding parameters listed in Table 1. Filtered bounce, roll, and pitch signals were multiplied by the corresponding skyhook gains to compute the vertical force F_z , roll moment M_ϕ , and pitch moment M_θ . These force and moment components were then distributed to each wheel based on the geometric dimensions of the vehicle, resulting in the geometric transformation denoted by Equations (7)–(13). The high-level controller only computes the desired force F_d for each damper. Then, the equivalent current required to generate this force is calculated and used to drive the actuator. This conversion is performed within the region bounded by the maximum and minimum curves of the standard force–velocity characteristics, where hysteresis is neglected. Specifically, the desired force is projected onto this static characteristic envelope, and the equivalent current is computed based on the distance between the desired force and the intersection points of the force–velocity curves at a given stroke velocity. In subsequent figures, damping characteristics labeled “normal” derived from these hysteresis-excluded force–velocity curves.

$$F_z = C_{\text{sky-bounce}} \dot{Z}_s \quad (7)$$

$$M_\theta = C_{\text{sky-pitch}} \dot{\theta} \quad (8)$$

$$M_{\phi} = C_{\text{sky-roll}} \dot{\phi} \quad (9)$$

$$F_z = -F_{dfl} - F_{dfr} - F_{drl} - F_{drr} \quad (10)$$

$$M_{\theta} = L_f \cdot F_{dfl} + L_f \cdot F_{dfr} - L_r \cdot F_{drl} - L_r \cdot F_{drr} \quad (11)$$

$$M_{\phi} = 0.5 \cdot t_f \cdot (-F_{dfl} + F_{dfr} - F_{drl} + F_{drr}) \quad (12)$$

$$\begin{bmatrix} F_{dfl} \\ F_{dfr} \\ F_{drl} \\ F_{drr} \end{bmatrix} = \begin{bmatrix} \frac{-1}{4L_f} & \frac{-L_f}{2(L_f+L_r)} & \frac{1}{2t_f} \\ \frac{-1}{4L_f} & \frac{-L_f}{2(L_f+L_r)} & \frac{-1}{2t_f} \\ \frac{-1}{4L_r} & \frac{L_r}{2(L_f+L_r)} & \frac{1}{2t_f} \\ \frac{-1}{4L_r} & \frac{L_r}{2(L_f+L_r)} & \frac{-1}{2t_f} \end{bmatrix} \begin{bmatrix} F_z \\ M_{\theta} \\ M_{\phi} \end{bmatrix} \quad (13)$$

Table 1. Vehicle parameters.

Total mass	2205 + 70 kg driver
Sprung mass	1900 kg
Tire spring constant, damping coefficient	$k_{tf,r} = 350 \text{ kN/m}$, $C_{tf,r} = 1000 \text{ Ns/m}$.
Anti-roll bar stiffness	$k_{sf} = 29 \text{ kN/m}$, $k_{sr} = 32 \text{ kN/m}$
L_f, L_r Distance between axles and -CoG, and axle width	1.35, 1.6 m
t_f	1.7 m

6.2. Simulation Setup and Results

Simulations were carried out using the IPG CarMaker 13.0-MATLAB/Simulink 2022b co-simulation framework. Three 500 m segments, one each for ISO 8608:2021 [16] classes A, B, and C, were generated in CRG (Curved Regular Grid) format at 50 mm spatial resolution, yielding a total road length of 1500 m, as shown in Figure 11. This configuration represents typical paved-road excitation spectra from smooth (Class A) to moderately degraded (Class C) surfaces.

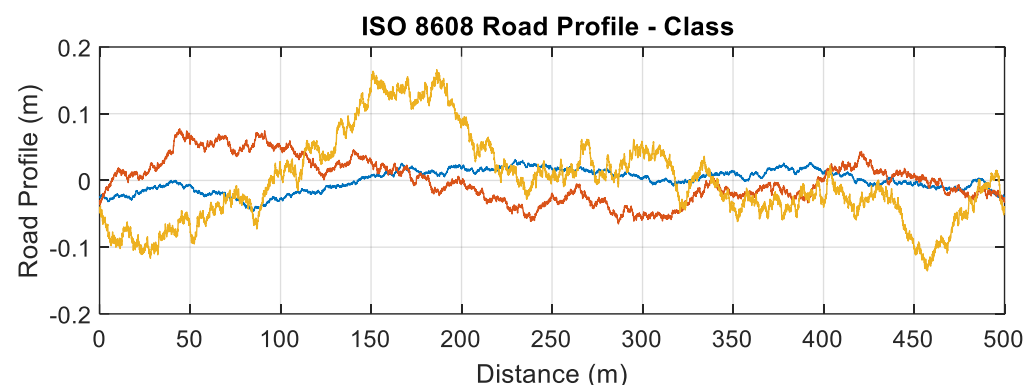


Figure 11. ISO A–B–C road profiles over 500 m (Class A: Blue, Class B: Red, Class C: Yellow).

The co-simulation framework, as seen in Figure 12, implemented two damper models: the conventional static force–velocity characteristic model and a hyperbolic tangent hysteresis model developed in this study. Damping force levels near the damper’s maximum damping capability (1.5 A) were selected for comparative analysis. The vehicle velocity was fixed at 87 km/h, and a high degree-of-freedom D-segment SUV model was used for all simulations. Time–domain responses were sampled at 1 kHz, and power spectral density (PSD) estimates were computed and then smoothed with a 15-sample moving average filter to improve readability.

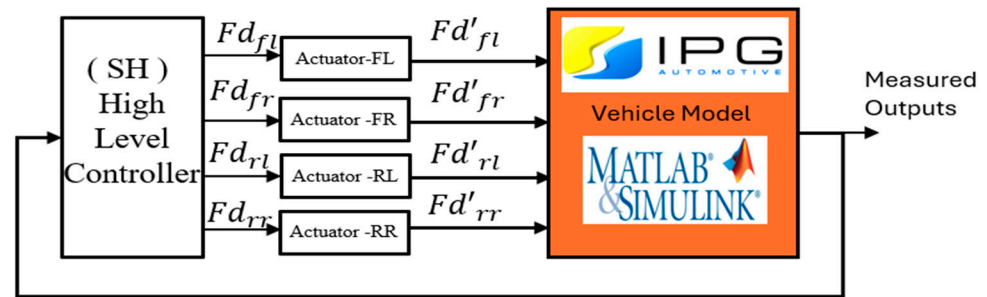


Figure 12. Control block diagram and co-simulation environment.

Figure 13 shows critical hysteresis effects on ride dynamics. Vertical acceleration deviations between hysteresis and non-hysteresis models increase progressively from about 1 dB at body resonance (1.5 Hz) to >5 dB at 6 Hz. All four simulation cases (controlled vs. uncontrolled, hysteretic vs. hysteresis-free) exhibit the same overall trend, and although differences above 6 Hz become more pronounced, the 2–6 Hz band, critical to human sensitivity, demonstrates the largest deviations when hysteresis is included. A comparable pattern appears in the pitch rate PSD: under skyhook control, the hysteresis-induced difference near 1.7 Hz is approximately 3 dB, and constant current tests confirm the same effect (Figure 14). Because the ISO 8608:2021 road profiles are symmetric around the vehicle's longitudinal axis, roll inputs from road excitation are negligible, hence the lower roll rate magnitudes, but even roll rate PSD shows measurable energy differences between the two model types (Figure 15). These consistent, non-random deviations confirm hysteresis substantially influences ride-comfort prediction accuracy.

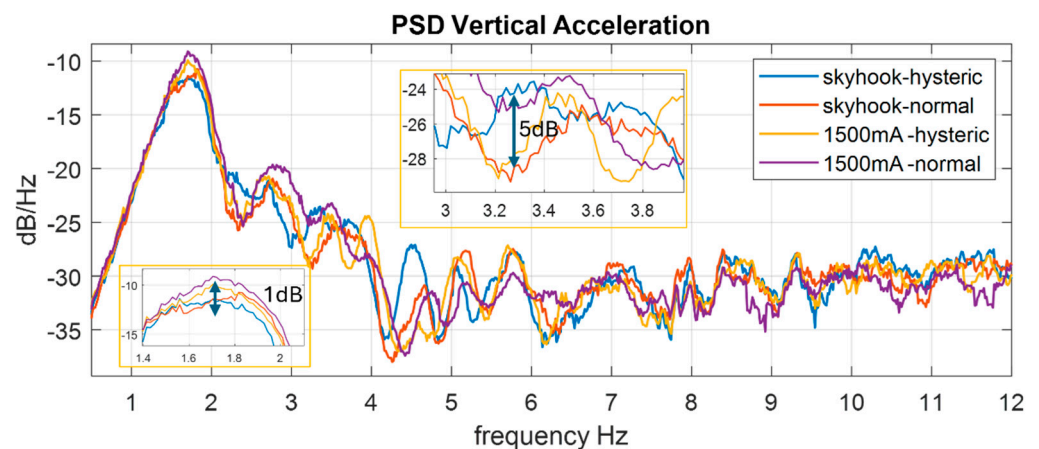


Figure 13. PSD of vertical acceleration at center of gravity.

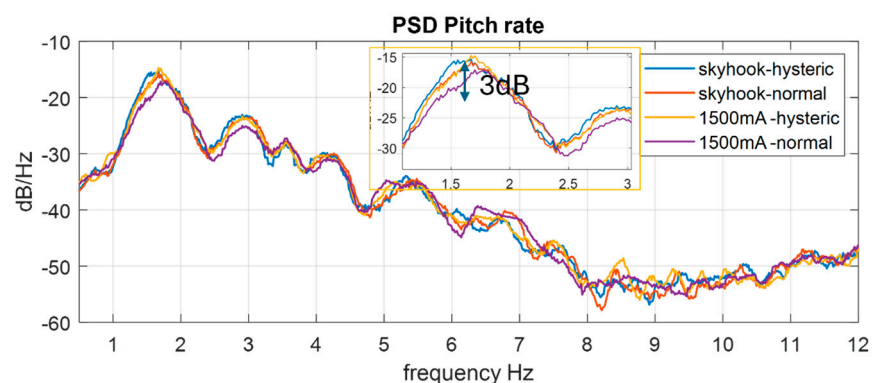


Figure 14. PSD of pitch rate at center of gravity.

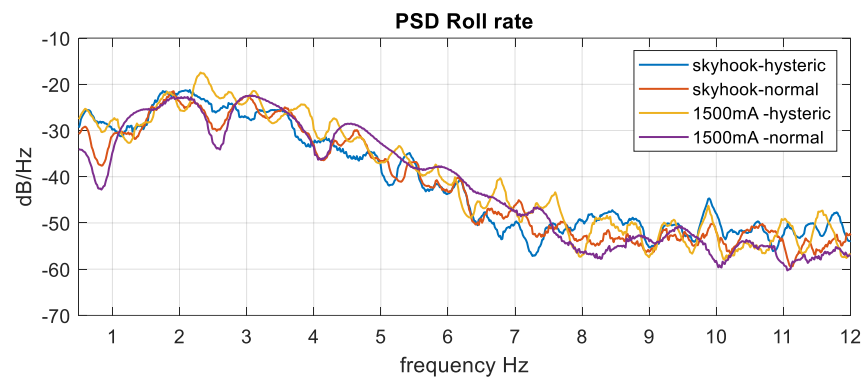


Figure 15. PSD of roll rate at center of gravity.

7. Conclusions

This study conclusively demonstrates that incorporating hysteresis effects into electrohydraulic damper models significantly alters predicted dynamic responses, particularly within the 1.5–6 Hz frequency band critical for human ride comfort. Vertical acceleration and pitch rate analyses reveal deviations exceeding 5 dB when hysteresis is neglected, a discrepancy consistent across skyhook-controlled and constant current scenarios. These results underscore that accurate hysteresis modeling is essential for reliable virtual assessment of suspension performance. The proposed hyperbolic tangent formulation, parametrized via GA optimization and mesh-based interpolation, successfully captures current–velocity nonlinearities while maintaining computational viability, extending originally Kwok et al.’s [11] observations on algebraic hysteresis models to production electrohydraulic dampers.

8. Future Work

Future studies will focus on the integration of hysteresis-aware current profiles into the solenoid-valve control algorithm. Additionally, development of physics-informed black-box models, combining the hyperbolic-tangent core with data-driven corrections, is proposed to improve accuracy and generalizability across a broader range of operating conditions.

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