

MULTI-OBJECTIVE OPTIMIZATION WITH GENETIC ALGORITHM OF AIR SUSPENSION SYSTEM FOR ENHANCING RIDE COMFORT AND ROAD-HOLDING PERFORMANCES

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This study presents a systematic investigation of air suspension control strategies for vehicular applications, employing computational optimization techniques to enhance both ride comfort and road holding performances. A comprehensive simulation framework was developed, incorporating a quarter-truck model with nonlinear air spring dynamics subjected to spectrally rich road excitations. The research methodology integrates conventional PID control with multi-objective genetic algorithm optimization to determine Pareto-optimal solutions for conflicting performance criteria. Extensive numerical simulations reveal that the optimized controller achieves significant improvements in vibration attenuation (58.4% reduction in sprung mass displacement) while maintaining superior road contact characteristics (41.5% enhancement in tire-road contact force) compared to a passive system. The analysis provides quantitative insights into the fundamental trade-space between ride comfort-oriented and stability-focused control objectives, demonstrating that intelligent optimization approaches can effectively navigate these design compromises. Furthermore, the results establish practical boundaries for control system performance under realistic operating conditions, offering valuable guidelines for automotive suspension system design. The technical contributions include novel control architectures, advanced performance evaluation metrics, and a rigorous methodology for controller optimization in nonlinear air suspension systems. These findings advance the state-of-the-art in vehicle dynamics control and identify several promising avenues for future research in adaptive control systems and hybrid optimization algorithms.

Keywords: air suspension system, genetic algorithm optimization, ride comfort, roadholding, PID-GA hybrid control

HIGHLIGHTS

- Investigated the relationship between indirect tensile strength and indirect tensile stiffness modulus in SBS-modified asphalt mixtures.
- Decreasing ITS and ITSM with increasing temperature reflects reduced tensile resistance and diminished load-spreading ability.
- ITS and ITSM are correlated and play complementary roles in mixture performance.

1 Introduction

The increasing demand for enhancing ride comfort, road handling, and operational safety in truck has led to continuous innovation in suspension system design. Trucks, which often operate under fluctuating load conditions and traverse uneven road surfaces, particularly benefit from suspension systems capable of adapting to dynamic environments [1, 2]. Air suspension systems, utilize compressed air chambers instead of traditional leaf springs, have emerged as a promising solution, offering better adaptability, self-leveling capability, and improved vibration isolation compared to traditional passive suspensions. However, due to their intrinsic nonlinearities and time-varying dynamics, controlling such systems remains a complex challenge, especially when seeking optimal performance across multiple conflicting objectives. To address the nonlinear, time-varying characteristics of these systems and improve control efficiency, numerous intelligent control strategies have been proposed and evaluated through both simulation and experimental methods [3, 4].

Several studies have applied fuzzy logic controllers to electronically controlled air suspension (ECAS) systems, utilizing inputs such as suspension deflection error and its rate of change to regulate the PWM duty cycle for actuator control. These approaches, as implemented in real-time hardware-in-the-loop simulations using dSPACE platforms [5], have demonstrated effective damping force regulation. Enhancements were also observed when combining fuzzy logic with PID strategies that applied to a 6×4 heavy vehicle reduced vertical acceleration and suspension dynamic [6]. Similarly, a fuzzy fractional-order PIλDμ controller applied to a nine-degree-of-freedom armored vehicle model significantly outperformed conventional PID and fuzzy-PID approaches in terms of ride comfort [7].

Optimization-based control techniques have also shown promise. Studies integrating Particle Swarm Optimization (PSO) with LQR controllers have achieved reductions of 31% in displacement amplitude and 85% in settling time under random vibration conditions (5–20 Hz), along with a 30% decrease in sprung mass acceleration [8]. Likewise,

Seeker Optimization Algorithm (SOA)-optimized PID controllers applied to transverse interconnected ECAS demonstrated up to 42.4% reductions in roll/pitch angles and improved response characteristics over conventional methods [9]. Genetic algorithms (GA) have been used to optimize both fuzzy-PID parameters [10] and air spring design variables [11], resulting in significant improvements in ride comfort and road safety margins across varied road conditions.

Recent advances have also explored adaptive and model-based control strategies. Adaptive Neuro-Fuzzy Inference Systems [12], adaptive neural network controllers [13], and adaptive control algorithms accounting for system uncertainties [14] have each demonstrated enhanced height tracking performance and reduced body acceleration under random road excitations. Nonlinear Model Predictive Control [15] and Hybrid Model Predictive Control [16] frameworks have further improved ride height accuracy and system responsiveness, especially under variable payload and operational conditions.

Innovative structural improvements have complemented control advancements. A dual-chamber air suspension design independently managing ride height and stiffness outperformed traditional single-chamber systems in tunability and posture control [17]. Energy-efficient designs, such as those employing active interconnected air springs with strategic airflow management, achieved a 72% reduction in roll angle and a 46% reduction in energy consumption [18]. Similarly, pressure optimization in tractor semi-trailers based on dynamic road stress led to a 20% decrease in road dynamic forces compared to leaf springs [19].

To address vehicle posture stability and ride height accuracy, several studies have combined co-simulation environments (e.g., AEMSim-Simulink) with dedicated controllers [20], or developed layered hybrid control structures integrating magneto-rheological dampers and solenoid valves [21]. Furthermore, sliding mode controllers [22] and nonlinear air spring models validated via experiments [7] have enabled improved robustness and handling across diverse road profiles.

To address the limitations of conventional control strategies in ECAS systems, this study introduces a novel multi-objective GA-based optimization framework for PID controller tuning, specifically tailored to the dynamic behavior of nonlinear air suspension systems in trucks. The proposed approach aims to enhance ride comfort and road holding by simultaneously minimizing vertical acceleration, and dynamic tire load under diverse road and loading conditions.

A comprehensive nonlinear quarter-truck model is developed, incorporating air spring thermodynamics, actuator dynamics, and external disturbances to accurately reflect real-world behavior. The PID controller parameters are optimized using a GA-driven multi-objective cost function, eliminating the reliance on manual tuning while improving robustness and adaptability. Simulation scenarios include both smooth and rough road profiles to assess controller performance under varied operating conditions. The main outstanding contributions of this study are as follows:

Development of a nonlinear model of a quarter-truck using ECAS system suitable for control and optimization studies.

- Formulation of a multi-objective performance index for PID parameters optimization using GA.
- Design and implementation of a GA-optimized PID control strategy to suppress vibrations and enhance ride comfort quality.
- Comparative analysis of the proposed GA-PID controller against conventional PID methods under road excitations.

Simulation results confirm that the GA-optimized PID controller delivers superior performance in terms of ride comfort and tire-road contact force maintenance, demonstrating the effectiveness of evolutionary optimization for complex suspension systems. This research contributes to the advancement of intelligent, adaptive control strategies in commercial vehicle applications, offering a scalable and computationally efficient solution for real-world implementation.

This research is structured as follows: Section 2 presents the controller design methodology for the quarter-truck model with air suspension system, incorporating both conventional PID control and multi-objective optimization approaches to determine optimal control parameters for diverse performance criteria. Section 3 evaluates controller performance through numerical simulations under standard road excitation conditions, with comprehensive analysis of system responses. The paper concludes with Section 4, which summarizes key findings, discusses practical implications, and identifies promising directions for future research.

2 Materials and methods

2.1 Truck modelling with air spring actuator

Accurate dynamic modeling of truck suspension systems is essential for analyzing ride comfort, road holding, and system controllability. Among various suspension types, the air spring system - often coupled with an auxiliary accumulator - offers significant adaptability and damping characteristics, especially under varying load conditions. In this study, the authors focus on modeling a quarter-truck system equipped with an air spring and a connected accumulator, considering both thermodynamic and fluid dynamic behaviors.

The air spring system, as shown in Fig. 1, consists of a primary air chamber and an auxiliary accumulator linked via a pipeline. The airspring system dynamics are governed by fundamental physical assumptions, each supported by

empirical evidence and theoretical analysis. The compressed air behavior follows an extended polytropic relationship that accounts for both thermal and dynamic effects:

$$PV^{n(T,\dot{V})} = \text{constant} \quad (1)$$

where the polytropic index n varies as:

$$n(T, \dot{V}) = n_0 + \alpha_T \Delta T + \alpha_{\dot{V}} |\dot{V}| \quad (2)$$

In which, the temperature dependence through coefficient α_T , which models heat transfer during rapid compression/expansion cycles $\alpha_T = 0.05 \div 0.15 K^{-1}$, and the rate-dependent behavior via $\alpha_{\dot{V}}$, accounting for kinetic effects in transient operations. The baseline index n_0 transitions between 1.0 (isothermal limit) and 1.4 (adiabatic limit) depending on system configuration.

Gas flow through interconnecting pipelines operates firmly in the turbulent regime, as confirmed by Reynolds number analysis:

$$Re = \frac{\rho v D_h}{\mu} > 4000 \quad (3)$$

Following that, the pressure drop follows quadratic velocity dependence $\Delta P \propto v^2$, flow separation effects become negligible for length-to-diameter ratios $\frac{L_t}{D_h} > 30$ and the friction factor becomes Reynolds-number dependent per Colebrook's equation:

$$\frac{1}{\sqrt{f_t}} = -2 \log \left(\frac{\epsilon}{3.7 D_h} + \frac{2.51}{Re \sqrt{f_t}} \right) \quad (4)$$

The mass distribution of gas along the pipeline is modeled with spatial modulation:

$$m(x, t) = \rho A_t \left[1 + \epsilon \cos \left(\frac{\pi x}{L_t} \right) \right] \quad (5)$$

Assuming adiabatic compression and expansion of air, the pressures inside the chambers follow:

$$\begin{cases} \mathcal{P}_c = P_{ref} \left(\frac{V_{c0}}{V_{c0} - \delta A_p + \gamma A_t} \right)^{n(T,\dot{V})} + \frac{\beta \delta^2}{V_{c0}^{\frac{2}{3}}} \\ \mathcal{P}_r = P_{ref} \left(\frac{V_{r0}}{V_{r0} - \gamma A_t} \right)^{n(T,\dot{V})} + \frac{\xi \gamma |\gamma|}{A_t^{\frac{3}{2}}} \end{cases} \quad (6)$$

where γ is the adiabatic index.

$$F = \underbrace{\frac{(P_{ref} - P_{atm}) A_p}{V_{c0} + V_{r0}}}_{\text{Static}} + \underbrace{K_e \delta \left[1 + \lambda \tanh \left(\frac{\delta}{\delta_0} \right) \right]}_{\text{Stiffness varying}} + \underbrace{K_c (\delta - \eta) \left[1 + \frac{(\delta - \eta)^2}{\Lambda^2} \right]^{-1}}_{\text{Geometric effect}} + \underbrace{C_1 \delta |\delta|^{0.7} + C_2 \text{sign}(\delta)}_{\text{Complex dissipation}} \quad (7)$$

with $K_e = \frac{P_{ref} A_p^2 \gamma^2}{V_{c0} + V_{r0}}$; $K_c = K_e \left(\frac{V_{r0}}{V_{c0}} \right)$; $\eta = \frac{A_t V_{c0} + V_{r0}}{A_p V_{r0}} \gamma$.

The dynamic equation for the distribution of gas mass inside the pipeline can be written as follows:

$$\frac{\partial}{\partial x} \left(m_{eq}(x, t) \frac{\partial \eta}{\partial t} \right) = \frac{A_t^2}{\rho} \frac{\partial \mathcal{P}}{\partial x} - \frac{f_t}{2 D_h} \rho \left| \frac{\partial \eta}{\partial t} \right| \frac{\partial \eta}{\partial t} \quad (8)$$

By combining the mechanical and thermodynamic subsystems, the complete nonlinear dynamics of the truck's air suspension can be represented. This system of differential equations can then be linearized around an operating point or used directly in a numerical simulation or control optimization framework.

$$\begin{cases} F = (P_{ref} - P_{atm})A_p + K_e\delta + K_c(\delta - \eta) \\ \rho L_t A_t \ddot{\eta} = K_c(\delta - \eta) - \frac{1}{2} \rho \xi A_t |\dot{\eta}|^\beta \text{sign}(\dot{\eta}) \end{cases} \quad (9)$$

where L_t is the connecting pipeline length; A_p is the effective area; V_{c0} is the initial chamber volume.

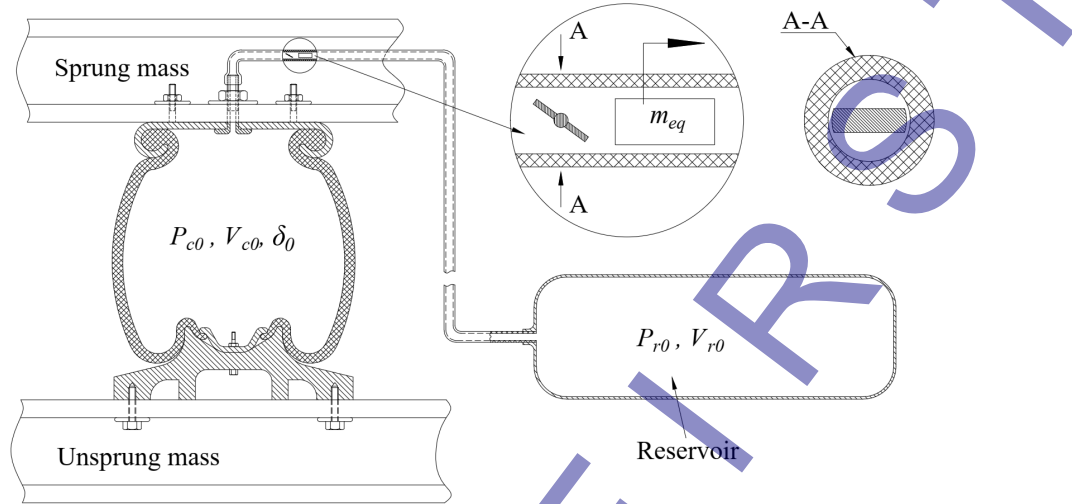


Fig. 1. Modelling of an air spring actuator in a quarter truck model

Table 1. Parameters for the air in airspring.

Symbol	Description	Value, Unit
\mathcal{K}	Polytropic coefficient	1.4
β	Velocity exponent	0.5
ξ	Turbulent flow coefficient	0.3
ρ	Air density	1.225 kg/m ³

As illustrated in Fig. 1, the quarter-truck model consists of a sprung mass m_s and an unsprung mass m_u with their respective vertical displacements denoted by Z_s and Z_u , respectively. These masses are interconnected through a damper characterized by a damping coefficient c_s and an air spring actuator, in which the equivalent air mass flow within the chamber m is governed by the nonlinear characteristics described in equation (9). The tire-road interaction is modeled as a linear spring with stiffness coefficient K_t , responding to the road disturbance q . The model is simplified in Fig. 2, and the governing equations for the quarter-truck system with the air suspension are presented in equation (10).

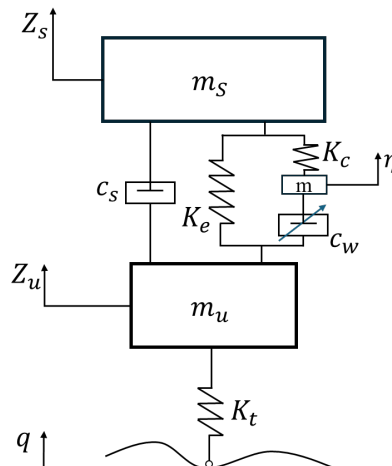


Fig. 2. An airspring actuator on a quarter truck model

$$\begin{cases} m_s \ddot{Z}_s = -c_s(Z_s - Z_u) - K_e(\dot{Z}_s - \dot{Z}_u) - K_c(Z_s - \eta) \\ m_u \ddot{Z}_u = c_s(Z_s - Z_u) + K_e(\dot{Z}_s - \dot{Z}_u) + c_w |\dot{Z}_s - \dot{\eta}|^\beta \text{sign}(\dot{Z}_s - \dot{\eta}) - K_t(Z_u - q) \end{cases} \quad (10)$$

2.2 Genetic algorithm for PID problem

The PID (Proportional–Integral–Derivative) controller is one of the most widely used control strategies in engineering due to its simplicity, robustness, and effectiveness in regulating dynamic systems. In the context of vehicle suspension, the PID controller aims to minimize oscillations and enhance ride comfort by adjusting the control force based on position and velocity feedback [23, 24]. This study employs a multi-objective genetic algorithm (MOGA) to optimally tune the PID controller parameters for air suspension system in truck. The primary aim is to improve both ride comfort and dynamic road-holding capability under various operating conditions. The air suspension system, being nonlinear and highly coupled, benefits significantly from evolutionary optimization techniques that can efficiently explore high-dimensional and non-convex solution spaces.

The PID controller parameters - proportional gain K_p , integral gain K_i , and derivative gain K_d - were optimized simultaneously to minimize two conflicting performance objectives:

- The vertical acceleration of the sprung mass \ddot{Z}_s , which affects passenger comfort.
- The vertical acceleration of the unsprung mass \ddot{Z}_u , which is related to road grip and tire wear.

The multi-objective problem is formally defined as:

$$\min_{K_p, K_i, K_d} \mathbf{F} = \begin{bmatrix} F_1(K_p, K_i, K_d) \\ F_2(K_p, K_i, K_d) \end{bmatrix} = \begin{bmatrix} \max(|\ddot{Z}_s(t)|) \\ \max(|\ddot{Z}_u(t)|) \end{bmatrix} \quad (11)$$

Subject to:

$$\begin{cases} 0 \leq K_p \leq 5 \times 10^6 \\ 0 \leq K_i \leq 5 \times 10^6 \\ 0 \leq K_d \leq 5 \times 10^6 \end{cases} \quad (12)$$

and constrained by:

$$\begin{aligned} g_1(K_p, K_i, K_d) &= \max(|\ddot{Z}_s(t)|) \rightarrow 0 \\ g_2(K_p, K_i, K_d) &= \max(|\ddot{Z}_u(t)|) \rightarrow 0 \end{aligned} \quad (13)$$

To address the defined optimization problem, MATLAB's *gamultiobj* function was utilized, incorporating the real-time Pareto front visualization tool *gaplotpareto* to monitor the evolutionary progress [25]. The population sizes are selected to balance computational efficiency and convergence performance, given that each fitness evaluation involves nonlinear dynamic simulation. Although larger populations may improve global exploration, the chosen sizes are sufficient to ensure consistent convergence and reliable comparative analysis of controller performance. To evaluate the impact of population diversity on convergence behavior and solution quality, three cases were considered with varying population sizes:

- Case 1: population size = 10,
- Case 2: population size = 20,
- Case 3: population size = 30.

Each configuration was executed with a maximum of 150 generations. To ensure consistent exploration and exploitation across all runs, the crossover fraction and Pareto fraction were both fixed at 0.6. This balance was chosen to maintain genetic diversity while facilitating the preservation of high-quality, non-dominated solutions throughout the optimization process. The fitness function was formulated as follows:

$$F = [\max(|\ddot{Z}_s(t)|), \max(|\ddot{Z}_u(t)|)] \quad (14)$$

This bi-objective formulation captures the essential trade-off in PID controllers' air suspension design: increasing comfort may reduce road contact stability, and vice versa. By applying MOGA, the Pareto front of non-dominated solutions was obtained, as figures out in Fig. 3, each representing a viable configuration for a specific design priority. Only feasible solutions that satisfied the nonlinear constraints were retained in the final front.

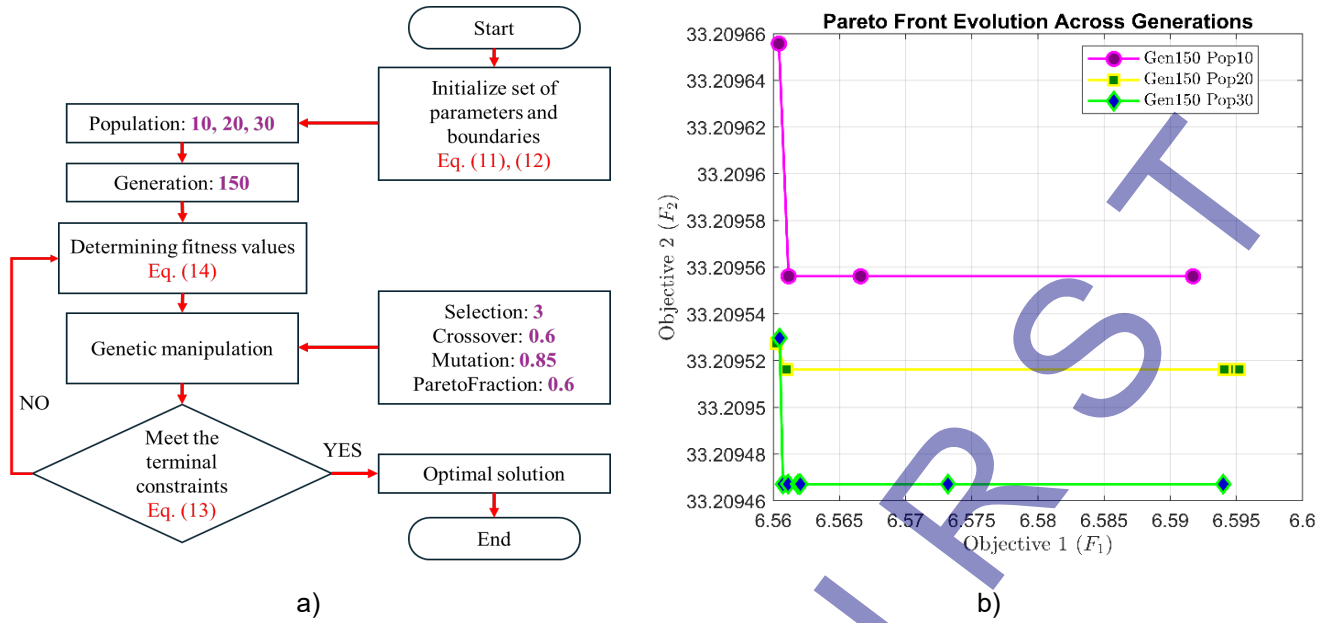


Fig. 3. The GA for PID problem flow (a) and the cost function (b)

3 Results and discussion

The road disturbance profile employed in this study, as illustrated in Fig. 4, is formulated as a structured non-stationary excitation signal designed to capture key characteristics of real-world road irregularities while maintaining controllability and repeatability for comparative analysis. The excitation profile is mathematically defined as follows:

$$z_r(t) = \sum_{i=1}^n A_i e^{-\alpha_i t} \sin(f_i t + \phi_i) \tag{15}$$

The characteristics of the road disturbance from (15) are defined in Table 2.

Table 2. Values for defining the road disturbance.

Symbol	Description	Value	Unit
A_i	Amplitude components	0.1	m
α_i	Decay constants for transient features	0	-
f_i	Frequency range of typical road disturbances	5	rad/s
ϕ_i	Random phase shifts ensuring non-periodicity	0	rad

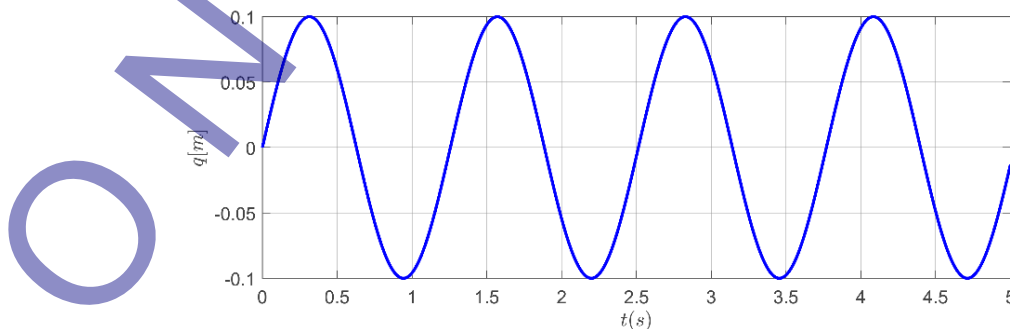
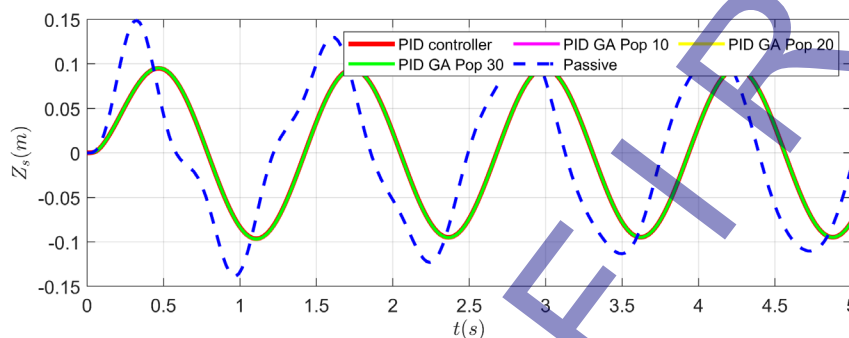


Fig. 4. Road exogenous on simulation

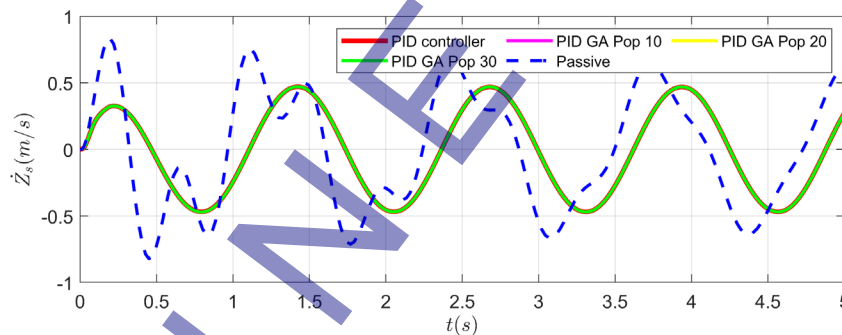
Fig. 5 evaluates the efficacy of Passive (blue line), PID (red line), and GA-based PID for air suspension control systems by analyzing their impact on key ride comfort metrics: vertical displacement (Z_s), vertical velocity (\dot{Z}_s), and vertical acceleration (\ddot{Z}_s). The GA-based PID with 3 models as the population varying of 10, 20, 30 illustrates in purple, yellow and green lines, respectively. The root mean square (RMS) values reveal substantial performance differences among the control strategies [11]. The GA-optimized controller with a population size of 10 (GA-Pop10)

demonstrates particularly noteworthy performance, achieving 58.37% and 33.08% reductions in vertical displacement and acceleration respectively compared to the passive system. While the PID controller provides consistent improvements across all measured parameters, the GA-Pop10 implementation shows superior performance in displacement and acceleration mitigation. The velocity metric (\dot{Z}_s), representing the sprung mass velocity, shows more nuanced results - the PID controller maintains a 27.67% improvement over the passive system, whereas GA-Pop10 exhibits a marginal 5.20% degradation relative to PID. This suggests a potential trade-off between displacement control and movement rate management in the GA-optimized solution.

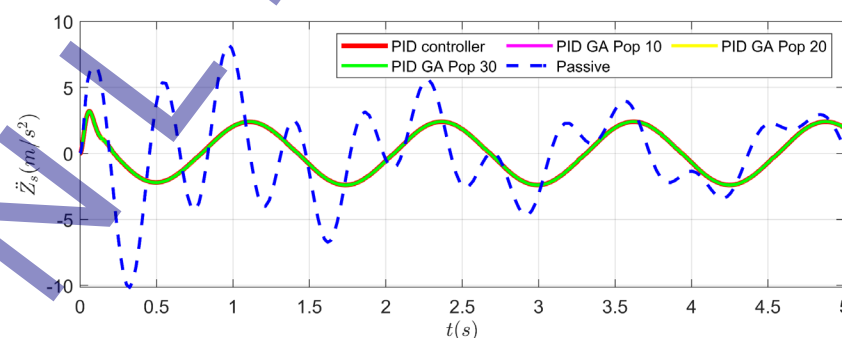
Notably, increasing the GA population size beyond 10 yields diminishing returns, with GA-Pop30 showing reduced effectiveness in certain metrics. These findings suggest that while GA-optimized controllers can significantly enhance ride comfort by reducing both low-frequency vibrations (displacement) and high-frequency oscillations (acceleration), careful parameter selection is crucial to maintain balanced performance across all comfort indicators. The results underscore the potential of intelligent control algorithms in advanced suspension systems while highlighting the need for multi-objective optimization in controller design.



(a) Spung mass displacement



(b) Spung mass velocity

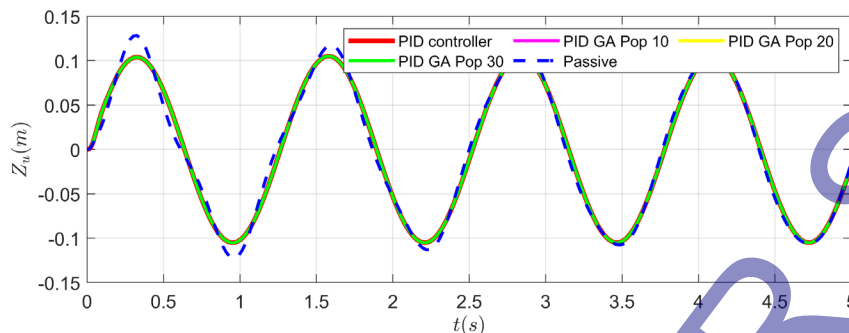


(c) Spung mass acceleration

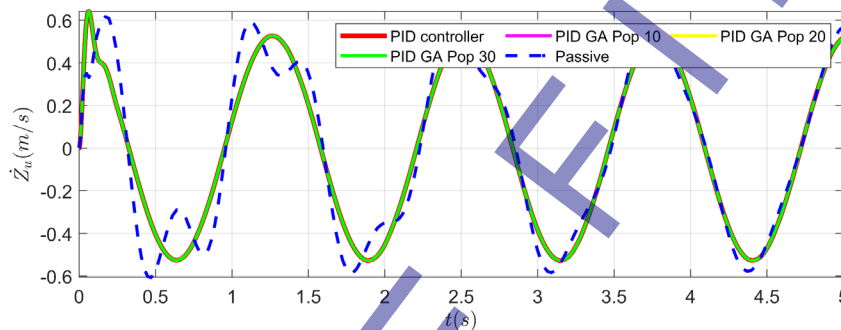
Fig. 5. Time respond of the spung mass due to road disturbance

In addition to ride comfort, the suspension system's ability to maintain optimal tire contact with the road - critical for vehicle stability and safety - was evaluated through the dynamic tire load metrics: tire displacement (Z_u), tire velocity (\dot{Z}_u), and tire acceleration (\ddot{Z}_u) which are displayed in Fig. 6. The RMS values indicate that the GA-Pop10 controller achieves superior road-holding performance compared to both passive and PID control systems. Specifically, it reduces tire displacement by 28.70% relative to the passive system while maintaining a 14.44% improvement in tire velocity. However, the PID controller exhibits a marginal advantage in minimizing unsprung mass acceleration, suggesting that while GA optimization enhances overall road contact, PID may offer better transient response in high-frequency disturbances.

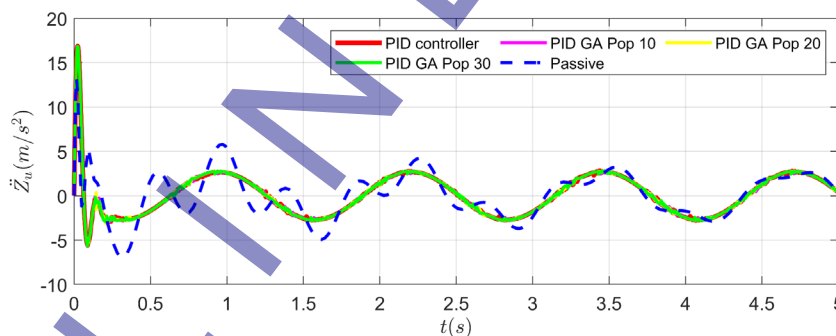
Interestingly, larger GA population sizes (Pop20 and Pop30) show progressive improvements in unsprung mass displacement reduction (up to 33.58% for GA-Pop30), but this comes at the expense of slightly degraded performance in unsprung mass velocity and acceleration control. This trade-off implies that while evolutionary algorithms can effectively optimize steady-state road-holding characteristics, fine-tuning may be necessary to ensure dynamic stability under varying road conditions.



(a) Unsprung mass displacement

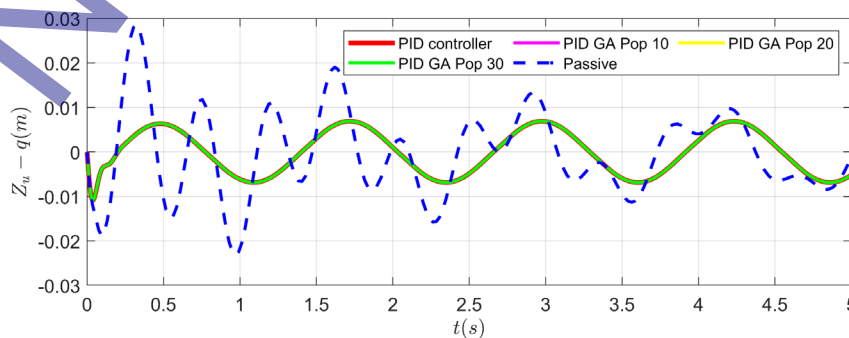


(b) Unsprung mass velocity



(c) Unsprung mass acceleration

Fig. 6. Time respond of the unsprung mass due to road disturbance.



(a)

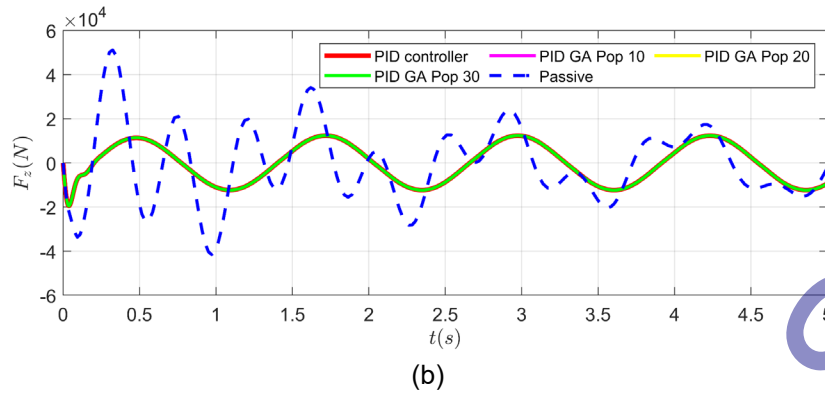


Fig. 7. Time respond of the tyre-road deflection (a) and the dynamic load (b)

The dynamic tire load (F_z) illustrated in Fig. 7, a critical indicator of road-holding capability, was evaluated to assess the suspension system's effectiveness in maintaining consistent wheel-to-road contact under varying conditions. The results demonstrate that the GA-Pop10 controller significantly reduces dynamic load fluctuations compared to the passive system, achieving a 41.51% improvement in F_z RMS values. While the PID controller also enhances road-holding performance (42.04% improvement over Passive model), the GA-optimized solution provides more stable load distribution, particularly in transient scenarios. However, the performance gains diminish with larger GA populations, as GA-Pop30 shows only marginal improvement (42.50%) over PID model, suggesting an optimal trade-off between computational complexity and control precision. These findings indicate that intelligent control strategies, particularly GA with moderate population sizes, can effectively minimize dynamic load variations, thereby improving traction and vehicle stability without compromising ride comfort. Further refinement of the fitness function to prioritize F_z minimization could yield additional improvements in road-holding performance.

Table 3. Comparison the RMS values and reductions of different models

		\ddot{Z}_s	\dot{Z}_s	Z_s	\ddot{Z}_u	\dot{Z}_u	Z_u	F_z
RMS values	Passive	3.509	0.437	0.083	2.933	0.388	0.07854	17751.95
	PID	1.634	0.316	0.064	2.158	0.462	0.04552	8112.53
	GA Pop10	1.461	0.333	0.056	2.091	0.445	0.04594	7253.77
	GA Pop20	1.487	0.320	0.061	2.030	0.439	0.04591	7339.93
	GA Pop30	1.539	0.336	0.061	1.948	0.462	0.04516	7583.88
Improvement to PID (%)	GA Pop10	10.60	-5.20	13.21	3.07	3.78	-0.91	10.59
	GA Pop20	8.97	-1.17	5.57	5.92	4.97	-0.85	9.52
	GA Pop30	5.78	-6.23	4.76	9.71	-0.09	0.79	6.52
Improvement to Passive (%)	PID	53.44	27.67	22.90	26.44	-18.94	42.04	54.30
	GA Pop10	58.37	23.90	33.08	28.70	-14.44	41.51	59.14
	GA Pop20	57.61	26.82	27.19	30.79	-13.03	41.55	58.65
	GA Pop30	56.13	23.16	26.56	33.58	-19.05	42.50	57.28

For the velocity of the unsprung mass, all control strategies - both PID and GA-based - exhibit higher RMS values compared to the passive system, with increases ranging from 13.03% to 19.05%. However, as illustrated in Fig. 7b, the amplitude differences remain relatively small. This suggests that while the controllers enhance overall ride comfort and sprung mass dynamics, they introduce a slight compromise in suppressing unsprung mass velocity - likely reflecting the inherent trade-off between ride comfort and road-holding performance.

4 Conclusion

This study investigates the air suspension system in heavy-duty trucks, beginning with the dynamic modeling of the air spring actuator mechanism based on kinematic equations. A controlled air suspension model utilizing a PID controller is subsequently developed to enhance system performance.

The results demonstrate that the active air suspension system, optimized using a Genetic Algorithm (GA), significantly outperforms both passive and conventional PID-controlled systems in terms of ride comfort and road-holding capabilities. Specifically, the GA-based controller achieves improvements of up to 58.37% in ride comfort and 41.51% in roadholding compared to the passive system. Although the PID controller remains effective for specific transient responses, the GA-optimized system with a population size of 10 (GA-Pop10) delivers superior overall performance, offering a well-balanced trade-off between vibration suppression and stability. Notably, increasing the population size beyond 10 results in marginal performance gains, indicating an optimal point for computational efficiency.

These findings underscore the potential of GA-based optimization in the development of advanced air suspension control systems. Future research will focus on incorporating actuator dynamics, such as valve delays and force saturation, to develop a more explicit and detailed representation of the air suspension system, while also exploring adaptive GA tuning techniques and hybrid control architectures to further enhance performance and robustness under varying real-world operating conditions.

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7 Conflict of interest statement

There are no conflicts affecting the research.

8 Author contributions

Vu Van Tan: Supervision, methodology, conceptualization, software, validation, writing – reviewing and editing. Do Trong Tu: Methodology, investigation, software, conceptualization, writing – reviewing and editing. Pham Tat Thang: Methodology, software, writing. Nguyen Trung Nguyen: Software, investigation, writing and editing. Sename Olivier: Methodology, supervision, validation, writing – reviewing and editing.

9 Availability statement

There is no dataset associated with the study or data is not shared.

10 Supplementary materials

There are no supplementary materials to include.

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