



Article citation information:

Nguyen, A.N., Nguyen, X.H., Pham, T.Q., Vu, H.Q. Enhancing ride comfort for the driver and passengers in a five-seat vehicle using regression modeling, TOPSIS, and PID control. *Scientific Journal of Silesian University of Technology. Series Transport*. 2026, **132**, 243-265. ISSN: 0209-3324. DOI: <https://doi.org/10.20858/sjsutst.2026.132.13>

Anh Ngoc NGUYEN¹, Xuan Hien NGUYEN², Tuan Quang PHAM³, Hai Quan VU⁴

ENHANCING RIDE COMFORT FOR THE DRIVER AND PASSENGERS IN A FIVE-SEAT VEHICLE USING REGRESSION MODELING, TOPSIS, AND PID CONTROL

Summary. This paper examines the effects of realistic and randomly varying loading conditions – including variations in driver mass, longitudinal seat position, and the number and seating positions of passengers – on the ride comfort of both the driver and passengers in a five-seat passenger vehicle. A thirteen-degree-of-freedom full-vehicle dynamic model is developed, incorporating an actively controlled driver seat suspension governed by a Proportional-Integral-Derivative (PID) controller. Regression analysis is employed to characterize the combined influence of loading and seat position variations on the root-mean-square (RMS) acceleration and RMS displacement of the driver seat. Subsequently, a multi-criteria decision-making method is applied to determine the optimal driver seat position. Three PID tuning methods – Ziegler-Nichols, Auto-Tune, and Particle Swarm Optimization (PSO) – are evaluated and compared. To improve computational efficiency, Ziegler-Nichols and Auto-Tune are used to constrain

¹ School of Mechanical and Engineering, Hanoi University of Industry (HaUI), 298 Cau Dien street, Hanoi, Vietnam. Email: ngocnguyencnoto@haui.edu.vn. ORCID: <https://orcid.org/0000-0003-4899-1694>

² School of Mechanical and Engineering, Hanoi University of Industry (HaUI), 298 Cau Dien street, Hanoi, Vietnam. Email: hien.nguyen15@haui.edu.vn. ORCID: <https://orcid.org/0000-0002-1552-1867>

³ School of Mechanical and Engineering, Hanoi University of Industry (HaUI), 298 Cau Dien street, Hanoi, Vietnam. Email: tuanquangpham2003@gmail.com. ORCID: <https://orcid.org/0009-0005-9705-9862>

⁴ School of Mechanical and Engineering, Hanoi University of Industry (HaUI), 298 Cau Dien street, Hanoi, Vietnam. Email: quanhv@haui.edu.vn. ORCID: <https://orcid.org/0000-0002-9560-8662>

the PSO search space. Simulation results indicate that the integrated application of regression analysis, TOPSIS, and PID control can achieve up to a 78% reduction in RMS seat acceleration while maintaining floor vibration levels within the comfort limits specified by ISO 2631. In addition, the proposed PSO search-space optimization strategy demonstrates a notable improvement in computational time.

Keywords: ride comfort, PSO-based optimization, multi-criteria decision making, Polynomial regression

1. INTRODUCTION

Ride comfort is a key factor influencing driver performance, passenger satisfaction, and overall safety, particularly in five-seater passenger vehicles operating under variable loads and road conditions. Excessive road-induced vibrations can lead to discomfort, fatigue, and long-term musculoskeletal disorders, as defined in ISO 2631-1 [1]. The vibration behavior of such vehicles is inherently nonlinear due to variations in seat properties, asymmetric passenger distribution, and luggage placement.

Conventional passive suspension systems (PSS) are simple and economical but fail to adapt to changing road excitations and payloads, thus limiting vibration isolation [2, 3]. Although parameter optimization improves their performance, effectiveness remains limited under dynamic conditions. Active and semi-active suspensions (ASS, SASS) overcome these limitations by adjusting damping and stiffness in real time. Modern control strategies, such as model predictive control (MPC), fuzzy logic control (FLC), and hybrid GA-LQR optimization, have improved both ride comfort and handling [4-6]. Furthermore, intelligent control methods, including deep learning, genetic-based PID tuning, and hybrid fuzzy-genetic algorithms, show strong potential for vibration mitigation [7, 8], while machine learning (ML) introduces adaptive self-optimization capabilities [9]. However, issues of data reliability, interpretability, and safety remain open challenges.

Despite these advancements, most studies rely on simplified quarter- or half-vehicle models that neglect coupling between full-vehicle dynamics, seat suspension, and load distribution. Influences such as seat cushioning and luggage position are often overlooked. Classical PID controllers remain prevalent in practice, yet few works have systematically compared different tuning techniques – Ziegler-Nichols, Auto-Tune, and PSO – within a full-vehicle context [10-13]. Moreover, multi-criteria decision-making (MCDM) and regression-based prediction models are rarely employed to comprehensively assess ride comfort [14, 15].

To address these gaps, this study proposes an integrated framework for optimizing ride comfort in five-seater vehicles by: Developing a 13-DOF full-vehicle vibration model incorporating seat suspension and load distribution; deriving analytical formulations for adaptive suspension tuning; comparing three PID tuning approaches; implementing a regression-based vibration prediction model; and applying a TOPSIS-based MCDM method to evaluate ride comfort under diverse operating conditions.

2. METHODS AND MATERIAL

2.1. Vibration Modeling

Based on Newton's second law, D'Alembert's principle, and the principle of conservation of energy, a dynamic model of the full suspension system for a five-passenger vehicle with one luggage compartment is constructed in Fig. 1.

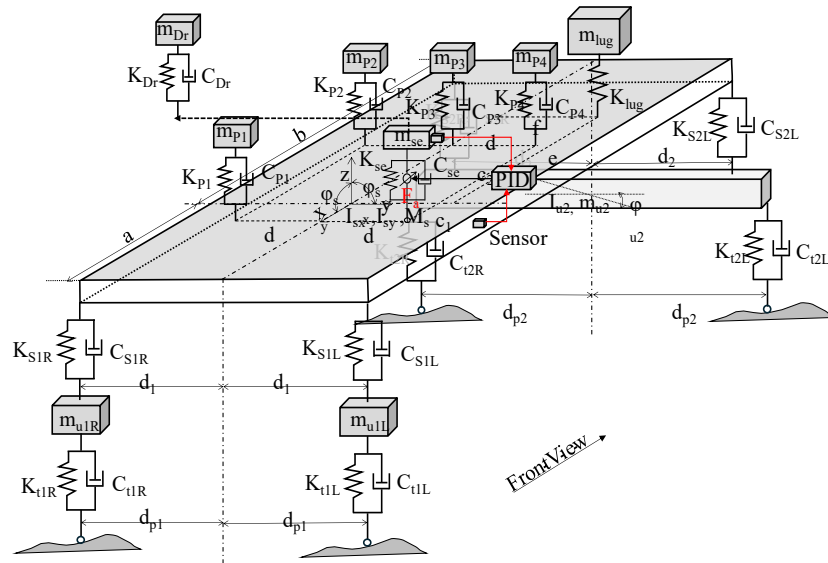


Fig. 1. Physical model

2.1.1. Full-vehicle Model

The full-vehicle model consists of thirteen degrees of freedom (DOF), representing the vertical and angular motions of a five-seat passenger vehicle under varying load conditions. The model includes the front unsprung mass m_{u1} , a rear unsprung mass m_{u2} with rotational inertia I_{u2} , the two-stage driver-seat subsystem comprising the seat frame mass m_{se} and the driver mass m_{Dr} , four lumped passenger-seat masses m_{p1-P4} , and the luggage mass m_{lug} . These DOFs collectively describe the primary pathways through which tire-road excitations propagate toward the driver and occupants.

2.1.2. Sprung Mass and Body Coordinates

The vehicle body is modeled as a rigid mass m_{scap} capable of heave, pitch, and roll motions, governed respectively by the inertial properties I_y (pitch) and I_x (roll). All subsystem coordinates are referenced to the geometric center of gravity (CG).

The longitudinal distances from the CG to the front and rear axles are denoted by a and b , while the lateral offsets of the suspension mounting points are given by d_1 and d_2 . The driver-seat and passenger-row longitudinal locations are specified by C_1 and C_2 . The driver's seat and passengers also exhibit a lateral offset d . The front and rear tire positions relative to the roll axis are defined by dp_1 and dp_2 . These geometric parameters determine the transformation between local vertical displacements and the resulting pitch-roll motions.

2.1.3. Unsprung Masses, Tire, and Suspension Assemblies

Each front wheel is modeled as an independent unsprung mass m_{u1} connected to the sprung mass through suspension stiffness K_{s1} and damping C_{s1} . The corresponding tire dynamics are described using Kelvin-Voigt elements with parameters K_{t1} and C_{t1} .

The rear axle is represented by a single rigid unsprung body m_{u2} linked to the chassis via suspension parameters K_{s2} and C_{s2} . Its tire-road interaction is characterized by K_{t2} and C_{t2} .

2.1.4. Driver-Seat Subsystem with Active PID Actuation

The driver seat adopts a two-stage isolation structure consisting of the seat frame mass m_{se} supported by a seat suspension with stiffness K_{se} and damping C_{se} , and the driver mass m_{Dr} interacting through a viscoelastic cushion characterized by K_{Dr} and C_{Dr} .

An electromechanical actuator is mounted in parallel with the seat suspension to generate an active control force F_a . A PID controller modulates F_a based on the deviation between the driver's measured acceleration and the desired value (0) to suppress vertical vibration.

2.1.5. Passenger-Seat Masses

Four additional DOFs represent the vertical motions of the passenger-seat masses. Each mass is mounted directly on its cushion without auxiliary suspension elements, reflecting typical middle- and rear-row seating structures. These DOFs introduce additional loading on the car body and affect how vibration energy propagates across the cabin.

2.1.6. Luggage Mass Representation

The luggage is modeled as a lumped mass m_{lug} located at horizontal and vertical offsets: f and e from the CG. Luggage is placed on the vehicle floor via rigid bumpers, represented as K_{lug} and C_{lug} . Its presence alters axle load distribution, shifts the effective CG location, and modifies pitch-related vibration transmission.

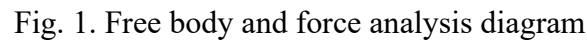
2.1.7. Integrated Multi-Subsystem Dynamics

All subsystems are coupled through suspension forces, tire-road interactions, geometric offsets, and distributed masses. The vertical forces arise from tire deflections (K_{t1} , K_{t2}), suspension compression (K_{s1} , K_{s2}), and seat-human interactions (K_{se} , C_{se} , K_{Dr} , C_{Dr}). The parameters a , b , d_1 , d_2 , d_{p1} , and d_{p2} dictate the pitch-roll coupling of these forces. The force analysis diagram for the entire system is illustrated in Fig. 2.

2.1.8. Differential Equations

Driver vertical displacement Z_{Dr}

$$m_{Dr}\ddot{Z}_{Dr} = -K_{Dr}(Z_{Dr} - Z_{Se}) - C_{Dr}(\dot{Z}_{Dr} - \dot{Z}_{Se}) \quad (1)$$


$$m_{Se}\ddot{Z}_{Se} = -K_{Se}(Z_{Se} - Z_f) - C_{Se}(\dot{Z}_{Se} - \dot{Z}_f) - K_{Dr}(Z_{Dr} - Z_{Se}) - C_{Dr}(\dot{Z}_{Dr} - \dot{Z}_{Se}) - K_{Se}(d\varphi_x + c_1\varphi_y) - C_{Se}(d\dot{\varphi}_x + c_1\dot{\varphi}_y) - F_a \quad (2)$$
$$m_{Pi}\ddot{Z}_{Pi} = -K_{Pi}(Z_{Pi} - Z_f) - C_{Pi}(\dot{Z}_{Pi} - \dot{Z}_f) - K_{Pi}(d_i \varphi_x + c_i \varphi_y) - C_{Pi}(d_i \dot{\varphi}_x + c_i \dot{\varphi}_y) \quad (3)$$
$$m_f \ddot{Z}_f = F_{KSe} + F_{CSe} + \sum_{i=1}^4 (F_{KPi} + F_{C Pi}) + \sum_{j=1}^2 (F_{KSjL} + F_{CSjL} + F_{KSjR} + F_{CSjR}) + F_{lug} + F_a \quad (4)$$
$$I_x \ddot{\phi}_x = d(F_{KSe} + F_{CSe}) + \sum_{j=1}^2 d_j (F_{KSjR} + F_{CSjR} - F_{KSjL} - F_{CSjL}) + d(F_{KP4} + F_{CP4}) + d(F_{KP1} + F_{CP1}) + fF_{lug} + F_a \quad (5)$$

Floor angular displacement (roll, lateral axis): φ_y

$$I_y \ddot{\phi}_y = c_1[(F_{KS1R} + F_{CS1R}) - (F_{KS1L} + F_{CS1L})] + c_2[(F_{KS2R} + F_{CS2R}) - (F_{KS2L} + F_{CS2L})] + \sum_{i=1}^4 c_i (F_{KPi} + F_{CPi}) + eF_{lug} + F_a \quad (6)$$

Front unsprung mass displacement $x = L, R$: Z_{u1x}

$$m_{u1} \ddot{Z}_{u1x} = K_{t1}(Z_{rx} - Z_{u1x}) - K_{s1}(Z_{u1x} - Z_{sx}) - C_{s1}(\dot{Z}_{u1x} - \dot{Z}_{sx}) \quad (7)$$

Rear unsprung mass displacement (dependent axle): Z_2

$$m_{u2} \ddot{Z}_2 = k_{t2}(Z_{r2} - Z_2) - k_{s2}(Z_2 - Z_{s2}) - c_{s2}(\dot{Z}_2 - \dot{Z}_{s2}) \quad (8)$$

Angular displacement of the rear unsprung mass: ϕ_2

$$I_{u2} \ddot{\phi}_2 = l_{s2}[k_{s2}(Z_{s2R} - Z_{s2L}) + c_{s2}(\dot{Z}_{s2R} - \dot{Z}_{s2L})] \quad (9)$$

2.1.9. Determination of Technical Parameters of the Seat Suspension System

According to ISO 2631 [1] and numerous experimental studies, ride comfort for occupants in passenger vehicles is most sensitive within the vibration frequency range of 1-2 Hz. Therefore, the design objective is to select the seat suspension stiffness (K) and damping coefficient (C) such that the natural frequency of the driver's seat suspension system falls within this optimal range, while maintaining adaptability to varying driver body masses and loading conditions. Accordingly, the key parameters are determined as:

$$K = (2\pi f_n)^2 m, \quad C = 2\zeta \sqrt{Km} \quad (10)$$

where f_n is the desired natural frequency (Hz); m is the combined seat-driver mass (kg); and ζ is the damping ratio (typically 0.2-0.8, with $\zeta = 0.8$ adopted in this study) [16].

2.2 Apply Methodologies

2.2.1. Polynomial Regression Model

A second-order polynomial regression model was established to predict the RMS acceleration and displacement responses at the driver's seat and vehicle floor. The general quadratic form is expressed as in Equation 11 [17].

$$y = \beta_0 + \beta_1 x_1 + \beta_2 x_2 + \beta_3 x_1^2 + \beta_4 x_2^2 + \beta_5 x_1 x_2 \quad (11)$$

In this study, the regression model is formulated as Equation 12:

$$Y = \beta_0 + \beta_1 m_{Dr} + \beta_2 C_1 + \beta_3 m_{Dr}^2 + \beta_4 C_1^2 + \beta_5 (m_{Dr} \cdot C_1) \quad (12)$$

where m_{Dr} is the driver's body mass (kg), C_l is the longitudinal seat coordinate (m), and β_i is the regression coefficients are estimated via least squares.

The model performance is evaluated using statistical indices, including the coefficient of determination R^2 , root mean square error (RMSE), as defined in Equation 13, 14.

$$R^2 = 1 - \frac{\sum_{i=1}^n (y_i - \hat{y}_i)^2}{\sum_{i=1}^n (y_i - \bar{y})^2} \quad (13)$$

$$RMSE = \sqrt{\frac{1}{n} \sum_{i=1}^n (y_i - \hat{y}_i)^2} \quad (14)$$

These metrics collectively assess model accuracy, robustness, and generalization capability [17, 18].

Tab. 1

Regression model evaluation

Metric	Good Threshold	Interpretation
R^2	> 0.95	Indicates an excellent model fit
RMSE	< 0.01-0.03 m/s ²	Reflects high prediction accuracy

2.2.2. TOPSIS Method

According to the eight-step procedure for calculating the C_i^* index presented in [19], the performance indicators, including the RMS acceleration, RMS displacement, and the driver's settling time, were determined using the assigned weighting factors of 0.35, 0.35, and 0.3, respectively.

2.2.3. PID Control Method

A PID controller is implemented to regulate the hydraulic actuator for attenuating road-induced vibrations at the driver's seat. Three tuning approaches are considered:

Ziegler-Nichols Method [20]

PID gains are determined as:

Tab. 2

Ziegler-Nichols tuning rule for PID

Controls	K_p	K_i or T_i	K_d or T_d
P	$0.5K_u$	—	—
PI	$0.45K_u$	$T_i = T_u/1.2$	—
PID	$0.6K_u$	$T_i = T_u/2$	$T_d = T_u/8$

MATLAB PID Tuner

Provides automatic PID tuning via linearization at the operating point; requires user expertise.

PSO-Based PID Optimization

The Particle Swarm Optimization (PSO) algorithm was employed to optimize the proportional-integral-derivative (PID) controller gains K_p , K_i , K_d for the active driver-seat suspension system. In this study, an electromechanical actuator mounted beneath the driver seat generates an active control force F_a to reduce seat vibration and improve ride comfort.

In the PSO framework, each particle represents a candidate PID parameter set:

$$Particle_i = [K_p, K_i, K_d] \quad (15)$$

The velocity and position of each particle are updated iteratively according to:

$$v_{i,j}^{k+1} = wv_{i,j}^k + c_1r_1(pbest_{i,j} - x_{i,j}^k) + c_2r_2(gbest_j - x_{i,j}^k) \quad x_{i,j}^{k+1} = x_{i,j}^k + v_{i,j}^{k+1} \quad (16)$$

where w is the inertia weight, c_1 and c_2 are the acceleration coefficients, and $r_1, r_2 \in [0,1]$ are random numbers. The inertia weight was dynamically updated to improve convergence performance:

$$w = w_{max} - \frac{(w_{max} - w_{min})k}{MaxIt} \quad (17)$$

The fitness evaluation was implemented in the MATLAB Simulink environment. The optimization objective was to minimize the weighted RMS value of the vertical driver-seat acceleration (a_w) while constraining the actuator force within the allowable limit of 500 N. The fitness function is defined as:

$$Fitness = RMS(a_w) + Penalty \quad (18)$$

with the penalty function expressed as:

$$Penalty = \begin{cases} 0, & \max |F_a| \leq 500 \\ (\max |F_a| - 500) \times 200, & \max |F_a| > 500 \end{cases}$$

The optimization process continued until the maximum number of iterations was reached, and the optimal PID parameters were obtained from the global best solution.

2.3. Ride Comfort Evaluation Standard

The ride comfort evaluation in this study follows the ISO 2631-1:1997 standard [1], in which the root mean square (RMS) acceleration in the vertical direction is employed as the primary indicator of whole-body vibration (WBV). The RMS acceleration is computed as:

$$a_w = \sqrt{\frac{1}{T} \int_0^T a_w^2(t) dt} \quad (19)$$

The ride comfort is classified based on the magnitude of whole-body vibration (WBV) exposure, with predefined thresholds corresponding to different comfort levels.

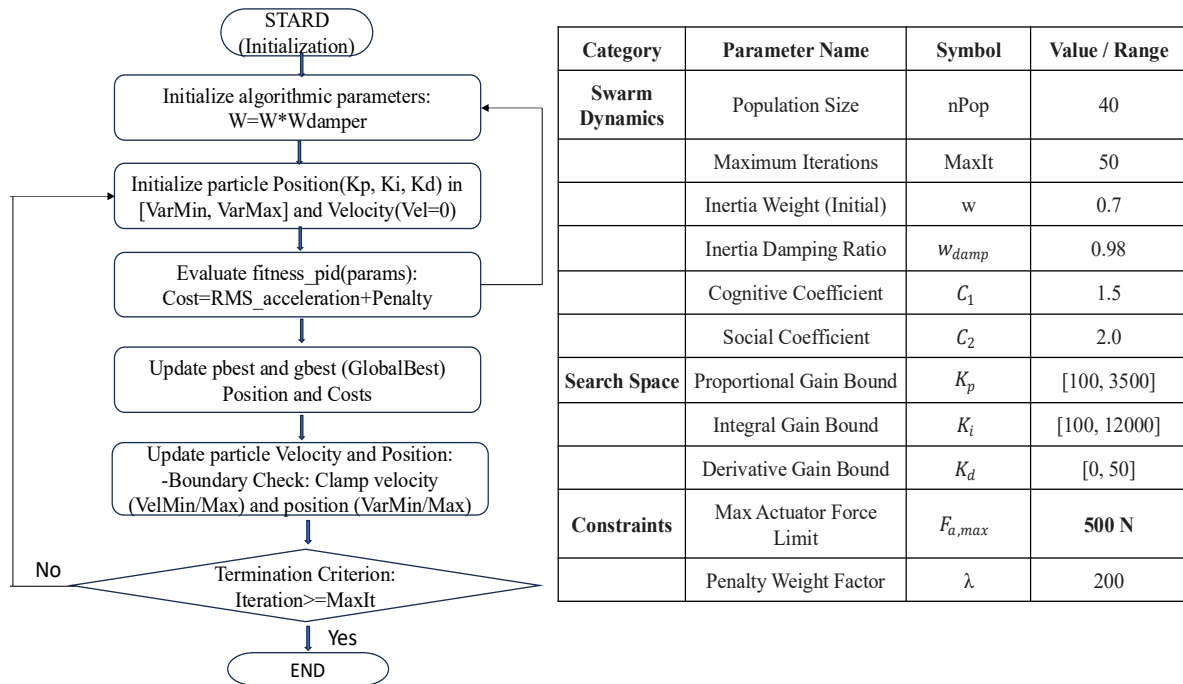


Fig. 3. Flowchart of the PSO algorithm and the values of its variables

Tab. 3

RMS acceleration assessment standard

Vibration Emission	Comfort Scale Rating
$<0.315 \text{ m/s}^2$	Not uncomfortable
$0.315 \text{ m/s}^2 - 0.63 \text{ m/s}^2$	A little uncomfortable
$0.5 \text{ m/s}^2 - 1 \text{ m/s}^2$	Fairly uncomfortable
$0.8 \text{ m/s}^2 - 1.6 \text{ m/s}^2$	Uncomfortable
$1.25 \text{ m/s}^2 - 2.5 \text{ m/s}^2$	Very uncomfortable
$>2.5 \text{ m/s}^2$	Extremely uncomfortable

2.4. Simulation

CD Road Profile Modeling: The road surface was modeled following ISO 8608:1995[23], using a Class C-D Power Spectral Density (PSD) profile at reference spatial frequency $n_0 = 0.1$ cycle/m with $G_q(n_0) = 256 \times 10^{-6} \text{ m}^3$ and slope exponent $w = 2$. The spatial PSD was converted to the time domain using $f = n \cdot v$ (vehicle velocity $v = 20 \text{ m/s}$). The road elevation signal $q(t)$ was synthesized as:

$$q(t) = \sum_{i=1}^N \sqrt{2G_q(n_i)\Delta n} \cos(2\pi n_i v t + \varphi_i) \quad (20)$$

where $\varphi_i \sim U(0,2\pi)$ is a random phase and Δn is the spatial frequency increment. The synthesized profile (band $n \in [0.01,10]$ cycle/m, $N = 1000$) was sampled at 1000 Hz for 10 s and implemented in Simulink via a From Workspace block to excite the vehicle model. The CD Road profile is described in Fig. 4.

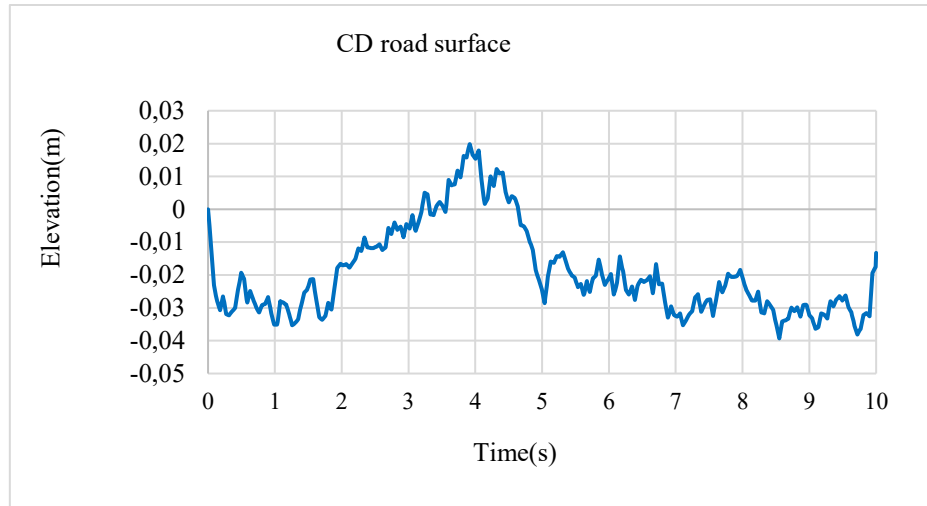


Fig. 4. Standard CD Road profile

The simulation block diagram developed in Simulink software is presented in Figure 5. The simulation data used in Table 4 are referenced from document [24]. In addition, the seat displacement parameter C_l was directly measured on a 2021 Mitsubishi XPander car.

Refer to the simulation parameter table reported in [24] as the simulation data.

Tab. 4

Simulation parameters

Symbol	Value	Unit	Symbol	Value	Unit
a	1.011	m	I_y	2460	Kg.m ²
b	1.803	m	I_x	460	Kg.m ²
c_1	0.3-0.5	m	I_{u2}	250	Kg.m ²
d	0.25	m	m_{u1}, m_2	40; 85	Kg
c_2	0.65	m	m_{se}	35	Kg
d_1, d_2	0.761; 0.711	m	m_s	1460	Kg
d_{p1}, d_{p2}	0.761; 0.761	m	m_{lg}	50	Kg
K_{se}	31000	N/m	V	16.67	m/s
C_{se}	830	N.s/m	g	9.8	m/s ²
K_{s1}, K_{s2}	19960; 17500	N/m	f	0	m
C_{s1}, C_{s2}	1290; 1620	N.s/m	e	1.2	m
K_{t1}, K_{t2}	175500	N/m	K_{Dr}	18000	N/m
C_{t1}, C_{t2}	20	N.s/m	C_{Dr}	200	N.s/m

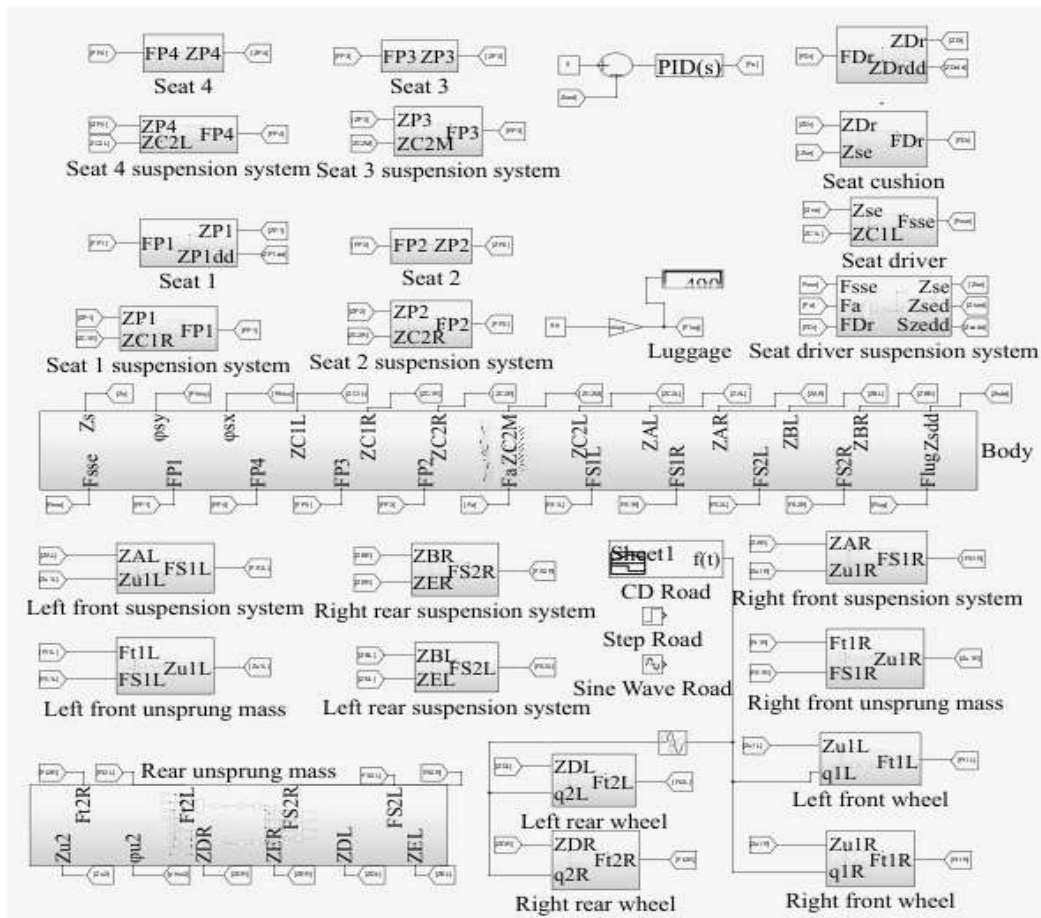


Fig. 5. Full-vehicle simulation model

3. RESULTS AND DISCUSSION

3.1. Effects of Driver Mass and Seat Position on Ride Comfort of Driver and Vehicle Floor

The regression analysis was conducted to predict the vibration responses at the driver's seat and vehicle floor under varying load and seating conditions. Numerical simulations were performed by varying the driver's body mass (40-100 kg) and longitudinal seat position (0.20-0.60 m) on a Mitsubishi Xpander 2021, yielding 117 cases.

RMS Acceleration at the Vehicle Floor (Floor a_{RMS})

As the driver's mass increases from 40 to 100 kg, the floor RMS acceleration slightly decreases from 0.411 to 0.391 m/s², indicating greater dynamic stability under higher loads. This trend agrees with the forced vibration relation $a = F/m$.

Seat position has a negligible influence, with variations below 0.002 m/s² across all masses. The high stiffness and uniform load distribution through the vehicle body ensure that the floor vibration remains largely unaffected by driver configuration. Overall, the fluctuation of less than 5% confirms that the floor maintains stable dynamic behavior under CD-type road excitation.

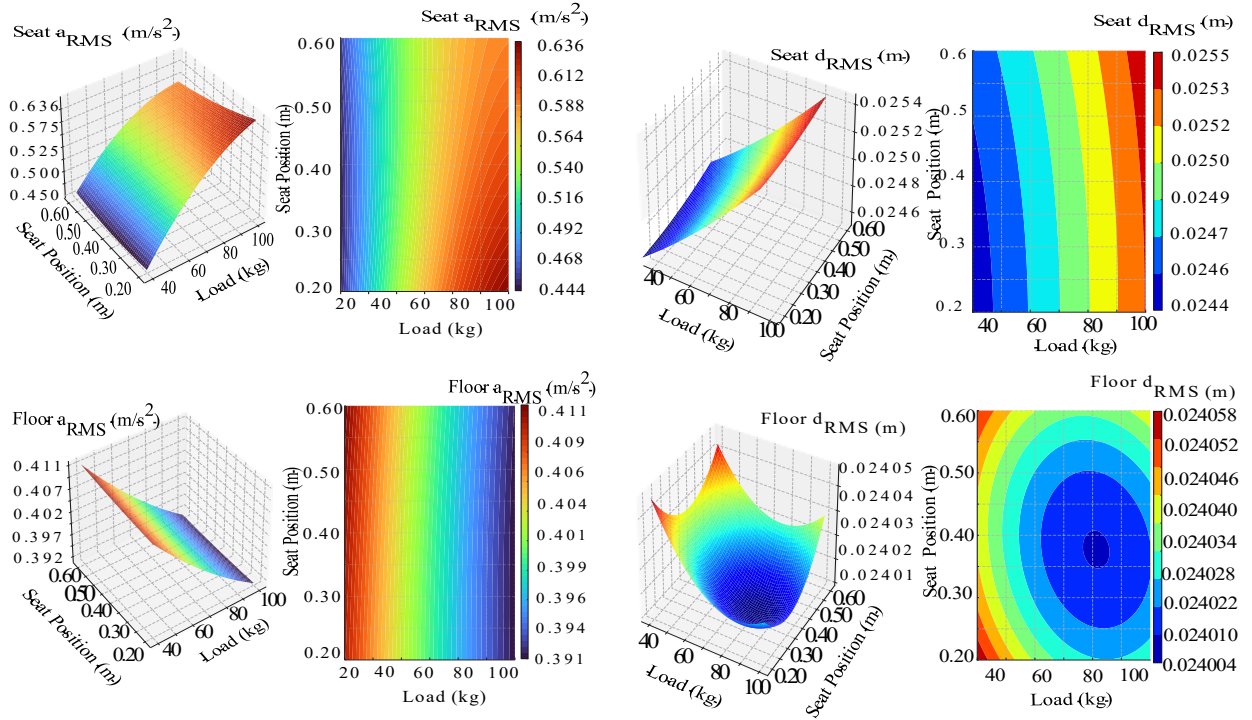


Fig. 2. The regression of RMS acceleration and displacement

RMS Displacement at the Vehicle Floor (Floor d_{RMS})

The floor RMS displacement remains nearly constant at 0.0240 ± 0.0001 m, showing that its motion primarily reflects the vehicle body's global vibration, governed by suspension and road input rather than seat-specific effects.

RMS Acceleration at the Driver's Seat (Seat a_{RMS})

Unlike the floor, seat RMS acceleration increases significantly with driver mass. As m_{Dr} rises from 40 to 100 kg, $a_{RMS,seat}$ grows by approximately 43% ($0.444 \rightarrow 0.636$ m/s² at $C_1 = 0.2$ m). This occurs due to resonance coupling: increasing mass reduces the seat's natural frequency $f_n = \frac{1}{2\pi} \sqrt{k_s/m_s}$. When f_{road} approaches f_n (≈ 2.3 Hz), resonance amplifies seat acceleration. Seat position causes only secondary effects – RMS acceleration slightly decreases at front positions (0.2–0.4 m) and stabilizes toward the rear, with overall variation within $\pm 5\%$. This is mainly due to changes in excitation energy related to the vehicle's center-of-gravity distance.

RMS Displacement at the Driver's Seat (Seat d_{RMS})

Seat displacement varies slightly between 0.02445 and 0.0255 m, increasing by about 4% with driver mass but remaining insensitive to seat position. The seat structure thus maintains stiffness and small oscillations around equilibrium without excessive deflection.

Regression Model Performance

RMS acceleration at the floor:

$$\begin{aligned} Floor_{a_{RMS}} = & 0.42986 - 0.00054 \cdot m_{Dr} + 0.00303 \cdot C_1 + 0.000002 \cdot m_{Dr}^2 \\ & - 0.00006 \cdot m_{Dr} \cdot C_1 + 0.00496 \cdot C_1^2 \end{aligned} \quad (21)$$

RMS displacement at the floor:

$$\begin{aligned} Floor_{d_{RMS}} = & 0.02420572 - 0.00000313 \cdot m_{Dr} - 0.00036603 \cdot C_1 \\ & + 0.00000002 \cdot m_{Dr}^2 + 0.00000073 \cdot m_{Dr} \cdot C_1 + 0.00040626 \\ & \cdot C_1^2 \end{aligned} \quad (22)$$

RMS acceleration at the driver's seat:

$$\begin{aligned} Seat_{a_{RMS}} = & 0.1781 + 0.00853 \cdot m_{Dr} - 0.00375 \cdot C_1 - 0.00003 \cdot m_{Dr}^2 \\ & - 0.00241 \cdot m_{Dr} \cdot C_1 + 0.15056 \cdot C_1^2 \end{aligned} \quad (23)$$

RMS displacement at the driver's seat:

$$\begin{aligned} Seat_{d_{RMS}} = & 0,0241479 + 7,755 \cdot 10^{-6} \cdot m_{Dr} - 1,4702 \cdot 10^{-4} \cdot C_1 + 4,7064 \cdot 10^{-8} \\ & \cdot m_{Dr}^2 - 1,7949 \cdot 10^{-6} \cdot m_{Dr} \cdot C_1 + 7,1262 \cdot 10^{-4} \cdot C_1^2 \end{aligned} \quad (24)$$

The second-order polynomial regression accurately characterizes the nonlinear relationship between driver mass, seat position, and vehicle vibration responses. The coefficients of determination demonstrate high predictive accuracy, with $R^2 = 0.9787$ for seat acceleration, 0.9572 for floor acceleration, and 0.9228 for seat displacement. In contrast, the floor displacement exhibits a very low coefficient of determination ($R^2 = 0.0392$), indicating minimal sensitivity to driver mass and seat position. This suggests that floor displacement remains nearly constant and is primarily governed by global vehicle body motion rather than local seat – occupant dynamics, making polynomial regression ineffective for predicting this response based on these variables alone. The corresponding root-mean-square errors (RMSE = 0.0055 m/s^2 for seat acceleration, 0.00135 m/s^2 for floor acceleration, and $7.48 \times 10^{-5} \text{ m}$ and $5.42 \times 10^{-5} \text{ m}$ for seat and floor displacements, respectively) confirm the regression model's strong interpolation capability and physical consistency. These results validate the model as a reliable surrogate for numerical simulation, enabling a substantial reduction in computational cost while maintaining high predictive precision. Extrapolation remains applicable for small variations in driver mass within fixed seat position limits. For instance, at $m_{Dr} = 120 \text{ kg}$ and $C_l = 0.3 \text{ m}$, the model predicts RMS accelerations of 0.695 m/s^2 (seat) and 0.393 m/s^2 (floor), both consistent with the observed nonlinear trend.

3.2. Integrated Evaluation Using Polynomial Regression and TOPSIS

Polynomial regression results were integrated into a multi-criteria decision-making (MCDM) framework to identify the optimal seat position balancing comfort and dynamic stability.

The analysis identifies the most comfortable configuration at $m_{Dr} = 40 \text{ kg}$ and $C_l = 0.35 \text{ m}$ ($C_l^* = 0.989$), while the least comfortable configuration occurs at $m_{Dr} = 100 \text{ kg}$ and $C_l = 0.20 \text{ m}$ ($C_l^* = 0.003$). These results provide a quantitative basis for seat position optimization under varying driver loads.

The analysis indicates that the optimal seat position (C_l) generally increases with driver mass from 40 kg to approximately 90 kg ($0.35\text{-}0.60 \text{ m}$) and slightly decreases at 100 kg . This trend reflects the shifting center of mass of the driver-seat system and its influence on vibration transmission, allowing improved comfort for moderate loads.

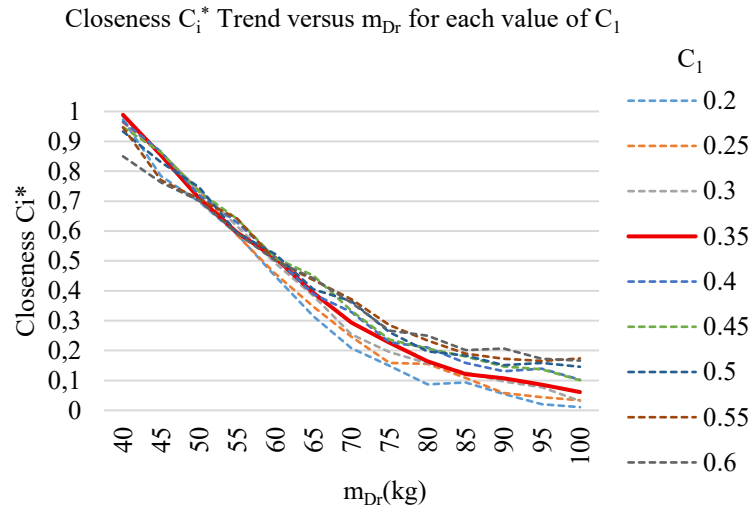


Fig. 3. The ride comfort index based on TOPSIS

3.3. Evaluation of Passenger and Luggage Effects on Driver Ride Comfort

Simulation Conditions: Passenger count varies randomly from 0 to 4, each weighing 60 kg, with seating positions randomly assigned (P1-P4). One luggage item of 50 kg is placed randomly in the trunk.

Tab. 5

Driver vibration under varying passenger and luggage configurations

Passengers & Luggage	Min C_i^* Position	Interpretation	Max C_i^* Position	Interpretation
0 passenger	—	No passengers or luggage, due to light floor load.	—	-
1 passenger	P2	The load is biased to the right	P1	Load uniform on the front axle
2 passengers	P2, P3	The load is biased to the right	P2, P4	Load uniform on the rear axle
3 passengers	P2, P3, P4	The load is biased to the right and rear	P1, P2, P4	Load uniform between front and rear
4 passengers & luggage	—	Heavy, uniform load	All positions + luggage	The absolute minimum occurs with a heavy, uniform load.

Tab. 5 indicates the randomly distributed passenger positions that result in the most uncomfortable and the most comfortable ride conditions for the driver, based on the evaluation of displacement and acceleration criteria using the TOPSIS.

The driver's RMS acceleration increases significantly under asymmetric loading, particularly when the mass is biased toward the right or rear positions (P2-P4). The lowest acceleration occurs when the vehicle carries four passengers and luggage, where the load is

heavy but evenly distributed; however, such loading also increases the seat and floor displacements, which, if excessive, may cause bump-stop contact and directly affect the driver's comfort and safety.

Tab. 6

Most and least comfortable passenger positions on the vehicle

Passengers	Max C_i^* Position	Min C_i^* Position	Ride comfort in descending order
1 passenger	P1	P2	P1>P4>P3>P2
2 passengers	P1_P2	P2_P3	P1_P2>P1_P3>P1_P4>P4_P2>P4_P3>P2_P3
3 passengers	P1_P2_P4	P2_P3_P4	P1_P2_P4>P1_P2_P3>P1_P3_P4>P2_P3_P4
4 passengers & luggage	P1	P2	P1>P4>P3>P2

Tab. 6 confirms that passenger seat selection directly affects individual ride comfort. Central positions, particularly P1, are subjected to lower vibration levels because they are closer to the vehicle's longitudinal centerline, where dynamic load transfer remains more balanced. In contrast, peripheral seats such as P2 experience higher accelerations due to asymmetric loading and increased lateral vibration. This variation is explained by the manner in which oscillatory forces are transmitted through the suspension and vehicle body, depending on seat location. In practical terms, passengers seated in less favorable positions are more susceptible to motion-related discomfort such as motion sickness, headaches, or nausea. This influence becomes even more critical in multi-passenger vehicles, where the choice of seating arrangement plays a decisive role in overall ride comfort.

3.4. Solutions for Improving Ride Comfort of Vehicle Occupants

3.4.1 Adjustment of Seat Suspension Parameters

Equation 10 was applied to recalculate the seat suspension parameters based on the data in Tab. 4. Subsequently, a comparison was carried out to evaluate the improvement achieved by adjusting the seat suspension characteristics to match the occupant's weight and the vehicle's ride comfort frequency, in comparison with the conventional passive seat suspension system. The calculation results indicate that the seat suspension stiffness was recalculated to $K_{se} = 2803 \text{ N/m}$ and the damping coefficient to $C_{se} = 714 \text{ N.s/m}$. The calculated parameters of the seat suspension system under different loading conditions were compared with the results reported in the literature [26, 27] to assess the validity of the numerical results.

The results in Figure 8 indicate that the comfort level of the seat suspension system strongly depends on the accurate tuning of the spring and damper parameters. In the non-optimized case (a_{ZDrdd_none}), the driver's RMS acceleration reached 0.5 m/s^2 , which is 19.6% higher than the floor acceleration: a_{Zsdd} (0.402 m/s^2), corresponding to a "quite uncomfortable" condition. After optimization, this value decreased to 0.33 m/s^2 (a_{ZDrdd_CT}), which is 17.9% lower than the floor acceleration (0.436 m/s^2) and 34% lower than that of the passive suspension system, demonstrating a significant improvement in ride comfort.

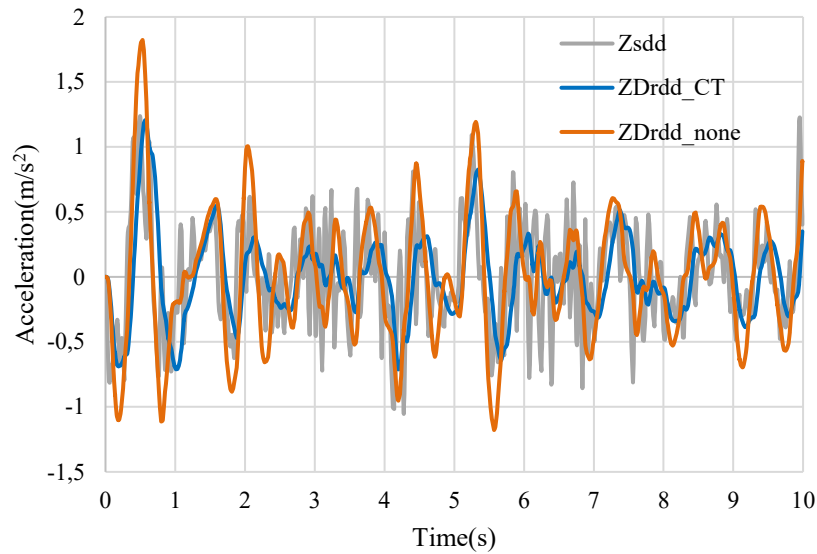


Fig. 8. Floor and driver acceleration graph

3.4.2 Implementation and Evaluation of PID Tuning Strategies

The PID controller tuned using the Ziegler-Nichols method reduced the RMS acceleration to approximately 0.35 m/s^2 , corresponding to a reduction of nearly 30%. According to ISO 2631 ride comfort evaluation criteria, this improvement changes the driver's perception from fairly uncomfortable to a little uncomfortable. ($K_p = 600$; $K_i = 10909$; $K_d = 8.25$).

The PID controller optimized using the Particle Swarm Optimization (PSO) algorithm further reduced the RMS acceleration to approximately 0.30 m/s^2 , equivalent to a 40% reduction ($K_p = 1927.4690$; $K_i = 8776.4448$; $K_d = 6.6063$).

Meanwhile, the MATLAB PID Tuner achieved the minimum RMS value of approximately 0.29 m/s^2 , corresponding to a 42% reduction, resulting in a ride condition classified as not uncomfortable ($K_p = 2683$; $K_i = 2738$; $K_d = 0$).

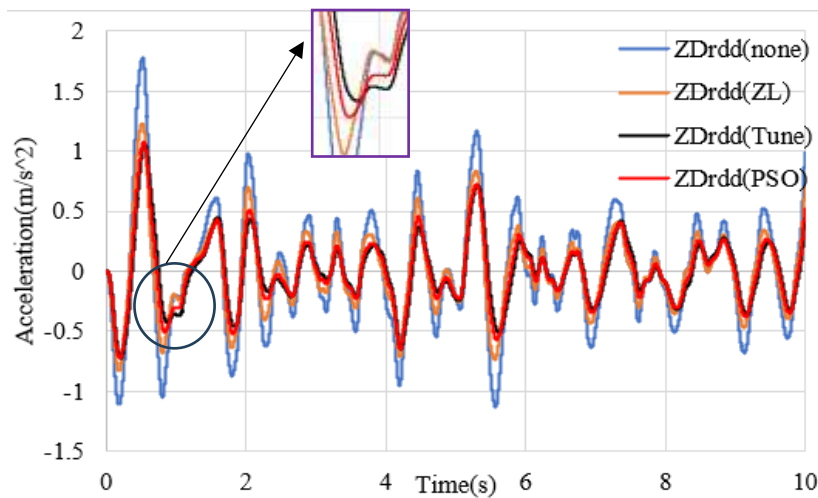


Fig. 9. Seat Acceleration when applying 3 PID turning methods

At approximately $t \approx 0.5\text{s}$, the vehicle encountered a sudden downward road disturbance immediately after the start of motion. This excitation generated the largest negative acceleration peak in the passive suspension response, exceeding -1 m/s^2 . To counteract this rapid downward motion, the active controllers generated compensating actuator forces. In particular, the PSO-based PID controller produced a large positive actuator force of approximately 430 N, which effectively lifted the seat upward and mitigated the “free-fall” effect induced by the road profile.

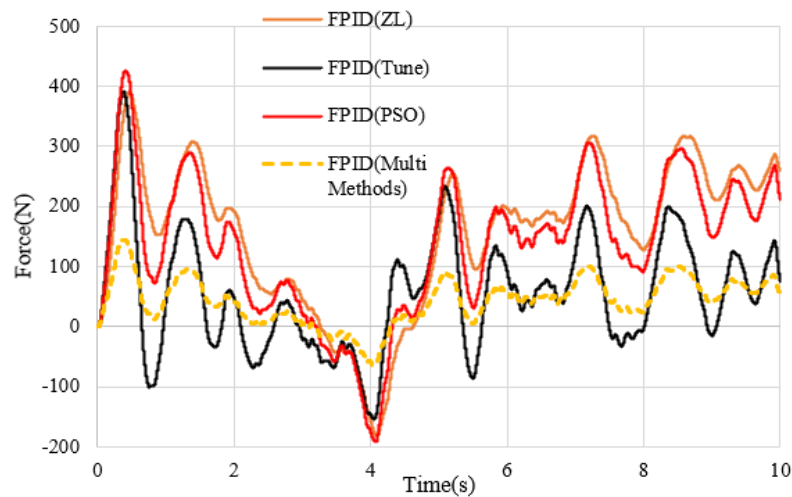


Fig. 10. Force generated by actuator

The PSD analysis indicates that the ISO Class C-D road profile contains most of its energy concentration within the low-frequency range. Consequently, the dominant PSD peaks of the driver-seat acceleration also appeared mainly in the frequency band of 0.5-1.5 Hz, which is highly sensitive to human body vibration. The active suspension controllers, especially the PSO-PID strategy, significantly attenuated these PSD peaks, demonstrating strong effectiveness in suppressing large and long-wavelength road irregularities associated with Class C-D road conditions.

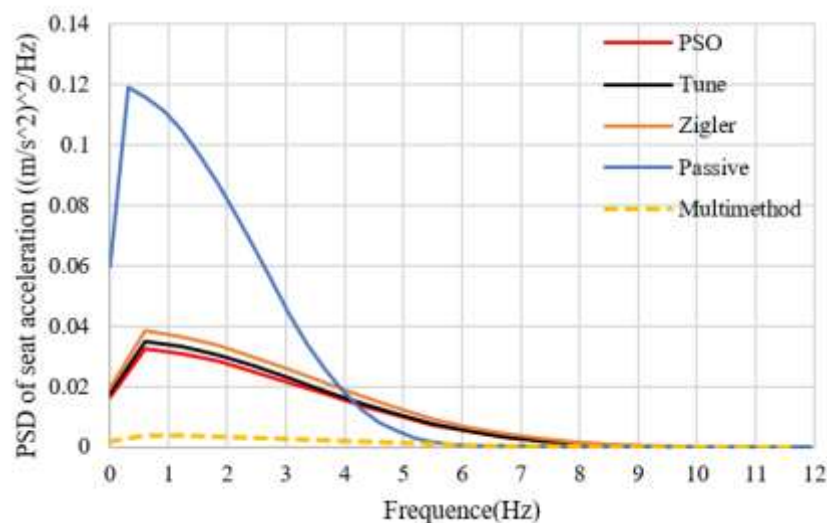


Fig. 11. PSD of seat acceleration

The PSD curves also reveal an important trade-off characteristic of the passive suspension system. After the major excitation peak occurring around 0.5s, the PSD magnitude of the passive system decreased rapidly and eventually became lower than the three active PID controllers after approximately 4s. This behavior reflects the inherent compromise of passive suspension systems, where high damping characteristics help the vehicle return more quickly to equilibrium following severe disturbances, albeit at the expense of transmitting larger initial accelerations to the driver.

3.4.3. The Multi-Method Integration

The Multi-Method Control Framework was developed based on the decision-making process of a fundamental Artificial Intelligence algorithm.

As illustrated in Fig. 8, the procedure begins with the real-time acquisition of key parameters, including driver and passenger loads, seat positions, and vehicle velocity (m_{Dr} , m_{Pi} , C_i , V), along with road excitations assumed as Step, SW, and CD profiles. After calibration and signal filtering, a regression model combined with a multi-criteria decision-making (MCDM) approach is applied to determine the most comfortable seat position for both passengers and luggage.

Once the load conditions are established, Equation 10 is employed to compute and adjust the suspension stiffness and damping coefficients (k_{se} and C_{se}) before vehicle departure. During operation, when the vehicle encounters uneven road surfaces and external disturbances, the Hybrid-PSO algorithm is activated to suppress real-time vibrations transmitted to the driver.

Finally, the measured vehicle responses are compared with predicted values, ISO reference limits, and the observed deviations are used to update model parameters continuously. This closed-loop learning mechanism enables the controller and prediction model to self-adapt and progressively enhance ride comfort performance through successive driving cycles.

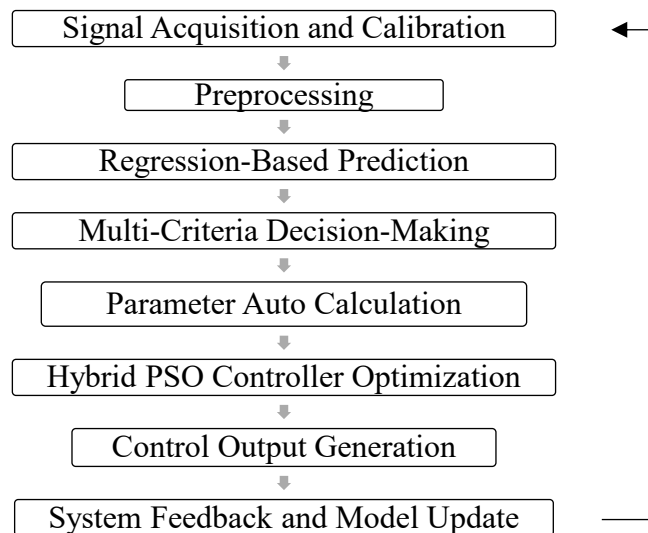


Fig. 12. Multi-method integration process

As illustrated in Fig. 12, polynomial regression is first employed to capture the characteristics of driver acceleration and displacement as functions of mass distribution and the longitudinal distance between the seat and the vehicle center of gravity. These results are then evaluated using the TOPSIS method to identify the optimal seating position. Subsequently, Equation 10 is employed to compute and adjust the suspension stiffness and damping coefficients (k_{se} and C_{se}) before vehicle departure. Finally, a PSO-PID controller is applied to attenuate vibration during vehicle operation

The evaluation scenario assumes a driver with a body mass of 56 kg accompanied by a passenger weighing 60 kg, both randomly positioned within the vehicle cabin. An additional luggage load of 50 kg is placed randomly in the trunk compartment. The vehicle is simulated on an ISO Class CD Road surface under a constant forward velocity of 16.67 m/s. The results of applying the multimethod calculation procedure are shown in Fig. 13 and 14.

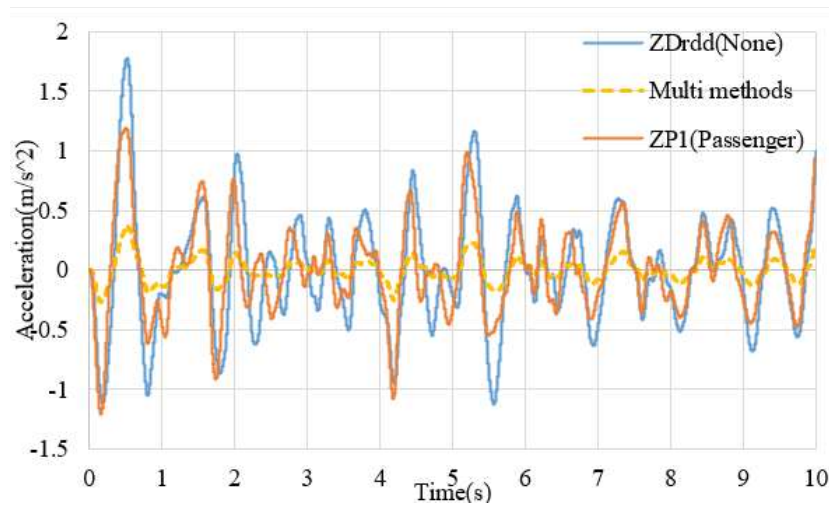


Fig. 13. Driver's acceleration

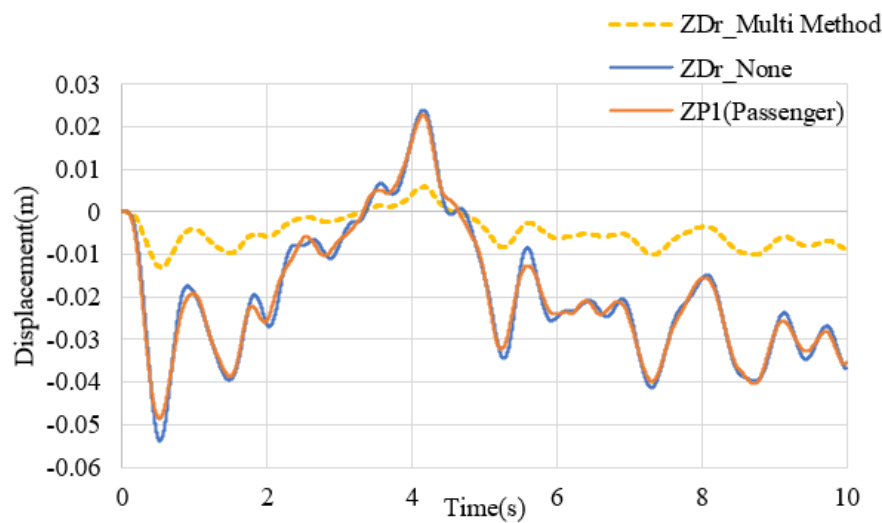


Fig. 14. Driver's displacement

The passenger seat position that provided the greatest comfort for both the driver and passenger was identified as P1, with $a_{(RMSP1)} = 0.401 \text{ m/s}^2$, corresponding to a “*slightly uncomfortable*” level according to ISO 2631-1, yet still within the acceptable range for short-term exposure. The optimal luggage position was determined at the center of the luggage compartment, aligned with the longitudinal axis of the vehicle.

The driver seat position that provided the greatest comfort was identified as $C_I = 0.35\text{m}$, the effective acceleration ($a_{(RMS)}$) was reduced by approximately 78%, from 0.50 m/s^2 to 0.11 m/s^2 , indicating a substantial attenuation of transmitted vibration and a significant improvement in the physiological perception of whole-body vibration.

The effective seat displacement ($d_{(RMS)}$) decreased by 76% (from 0.025 m to 0.006 m), reflecting enhanced postural stability, improved biomechanical coupling between the body and the seat, and reduced fatigue accumulation during prolonged driving. The obtained RMS acceleration and displacement values are consistent with previously reported results for vehicle suspension systems under similar operating conditions [15, 25].

The force generated by the actuator controlled by the PID controller using the PSO algorithm now only needs to reach a maximum value of 150 N (a reduction of 62.5%) while still ensuring high ride comfort for the driver. This improvement is achieved because the seat suspension stiffness has been adjusted appropriately according to the driver’s weight instead of using the default parameters listed in Table 4 (yellow dashed line in Fig. 10).

In particular, the PSO seat acceleration curve (yellow dashed line in Fig. 11) becomes smoother at low frequencies ($0.5\text{-}2 \text{ Hz}$) and approaches zero, demonstrating the effectiveness of combining multiple optimization methods.

The ride comfort position is defined as $C_I = 0.35 \text{ m}$. The seat suspension parameters are $K_{se} = 2803 \text{ N/m}$ and $C_{se} = 714 \text{ N.s/m}$. The passenger seating position is P_I , and the luggage position is at the center of gravity of the trunk ($f = 0 \text{ m}$).

4. CONCLUSIONS

This study demonstrates that ride comfort in a five-seat vehicle can be significantly improved through an integrated framework combining seat suspension stiffness tuning, PSO-PID control, regression-based prediction, and TOPSIS decision-making. The proposed approach reduced seat RMS acceleration by 78% and seat RMS displacement by 76%, while decreasing actuator force demand by 62.5% compared with the conventional PSO-PID strategy. These improvements effectively reduce vibration exposure, driver fatigue, and actuator workload, thereby enhancing both ride comfort and control efficiency.

The main novelty of this work lies in integrating seat-level suspension control with predictive passenger-load analysis and multi-criteria optimization. The results confirm that passenger seating position and luggage distribution strongly affect ride comfort, where the P1 seating configuration and centrally located luggage provide the best vibration performance. Unlike conventional full-vehicle suspension optimization, the proposed method improves local ride comfort at the driver seat without modifying the primary suspension system.

The proposed framework offers a scalable and intelligent solution for adaptive ride comfort optimization under varying occupancy and loading conditions. Future work will focus on experimental validation using magnetic or pneumatic seat suspension systems and the development of AI-based adaptive controllers for real-time vibration mitigation in multi-occupant vehicles.

References

1. ISO 2631-1. 1997. Mechanical Vibration and Shock – Evaluation of Human Exposure to Whole-Body Vibration – Part 1: General Requirements. *International Organization for Standardization*. Geneva, Switzerland.
2. Smith C. Malcolm, Fu-Cheng Wang. 2004. „Performance benefits in passive vehicle suspensions employing inerters”. *Vehicle System Dynamics* 42(4): 235-257. ISSN: 0042-3114. DOI: <https://doi.org/10.1080/00423110412331289871>.
3. Ikhsan N., R. Ramli, A. Alias. 2015. „Analysis of the kinematics and compliance of a passive suspension system using ADAMS Car”. *Journal of Mechanical Engineering and Sciences* 8: 1293-1301. ISSN: 2289-4659. DOI: <https://doi.org/10.15282/jmes.8.2015.4.0126>.
4. Cai Cheng, Guiyong Wang, Zhigang Wang, Raoqiang Li, Zhiwei Li. 2025. „Research on control strategy of semi-active suspension system based on fuzzy adaptive PID-MPC”. *Applied Sciences* 15(17): 9768. ISSN: 2076-3417. DOI: <https://doi.org/10.3390/app15179768>.
5. Zhao Weipeng, Liang Gu. 2023. „Hybrid particle swarm optimization genetic LQR controller for active suspension”. *Applied Sciences* 13(14): 8204. ISSN: 2076-3417. DOI: <https://doi.org/10.3390/app13148204>.
6. Liu Qianjie, Bo Hu, Wei Liu, Jiantao Li, Wenwen Yu, Gang Li, Guoliang Hu. 2024. „A fractional-order model predictive control strategy with Takagi-Sugeno fuzzy optimization for vehicle active suspension system”. *Fractal and Fractional* 8(10): 610. ISSN: 2504-3110. DOI: <https://doi.org/10.3390/fractalfract8100610>.
7. Zhang Shaobo, Mei Li, Jinsong Li, Jie Xu, Zelong Wang, Shuaihang Liu. 2024. „Research on ride comfort control of air suspension based on genetic algorithm optimized fuzzy PID”. *Applied Sciences* 14(17): 7787. ISSN: 2076-3417. DOI: <https://doi.org/10.3390/app14177787>.
8. Nagarkar P. Mahesh, Yogesh J. Bhalerao, Gahininath J. Vikhe Patil, Rahul N. Zaware Patil. 2018. „GA-based multi-objective optimization of active nonlinear quarter car suspension system—PID and fuzzy logic control”. *International Journal of Mechanical and Materials Engineering* 13: 10. ISSN: 2198-2791. DOI: <https://doi.org/10.1186/s40712-018-0096-8>.
9. Han Shi-Yuan, Tong Liang. 2022. „Reinforcement-learning-based vibration control for a vehicle semi-active suspension system via the PPO approach”. *Applied Sciences* 12(6): 3078. ISSN: 2076-3417. DOI: <https://doi.org/10.3390/app12063078>.
10. Parvez Yusuf, Nathi Ram Chauhan, Mahima Srivastava. 2025. „Vibration control and comparative analysis of passive and active suspension systems using PID controller with particle swarm optimization”. *Journal of the Institution of Engineers (India): Series C* 106(1): 1-19. ISSN: 2250-0545. DOI: <https://doi.org/10.1007/s40032-024-01038-y>.
11. Rao Gopala L. V. V., S. Narayanan. 2009. „Sky-hook control of nonlinear quarter car model traversing rough road matching performance of LQR control”. *Journal of Sound and Vibration* 323: 515-529. ISSN: 0022-460X. DOI: <https://doi.org/10.1016/j.jsv.2009.01.025>.
12. Tran Thanh An, Van Tuan Nguyen. 2024. „Optimization of the suspension system of passenger cars using the vibration model multi-objective method”. *Engineering, Technology & Applied Science Research* 14(5): 17019-17028. ISSN: 2241-4487. DOI: <https://doi.org/10.48084/etasr.8260>.

13. Sun Qihao, Changcheng Yin, Baohua Wang. 2024. „Experimental validation of truck cab suspension model and ride comfort improvement under various semi-active control strategies”. *Processes* 12(9): 1880. ISSN: 2227-9717.
DOI: <https://doi.org/10.3390/pr12091880>.
14. Deng Yaoji, Zhiyue Wang, Youqun Zhao, Junjie Gong, Hui Shen, Fen Lin. 2024. „Numerical prediction of ride comfort of tracked vehicle equipped with novel flexible road wheels”. *Chinese Journal of Mechanical Engineering* 37: 85. ISSN: 1000-9345.
DOI: <https://doi.org/10.1186/s10033-024-01070-0>.
15. Wu Difei. 2025. „Ride comfort prediction on urban road using discrete pavement roughness index”. *Applied Sciences* 14(7): 3108. ISSN: 2076-3417.
DOI: <https://doi.org/10.3390/app14073108>.
16. Vu Hai Quan, Quang Uy Duong. 2025. „Vehicle Body Vibration and Noise Impact on Driver and Passengers Analysis: The Case of the 29-Seat Thaco Garden TB79s Bus”. *Engineering, Technology & Applied Science Research* 15(1): 20049-20055. ISSN: 2241-4487. DOI: 10.48084/etasr.9200.
17. Montgomery C. Douglas, Elizabeth A. Peck, G. Geoffrey Vining. 2013. *Introduction to Linear Regression Analysis*, 5th ed. Hoboken, NJ, USA: John Wiley & Sons. ISBN: 978-0-470-54281-1.
18. Chai T., R. R. Draxler. 2014. „Root mean square error (RMSE) or mean absolute error (MAE)? – Arguments against avoiding RMSE in the literature”. *Geoscientific Model Development* 7: 1247-1250. ISSN: 1991-959X. DOI: <https://doi.org/10.5194/gmd-7-1247-2014>.
19. Vu Hai Quan, Anh Ngoc Nguyen, Minh Tien Nguyen, Thi Hong Hanh Nham, Van Hoan Hoang, Tung Duong Hoang. 2023. „The Research Design Calculations and Optimization of Control Arms in Mac Pherson Suspensions in Automobile”. In: *2023 International Conference on Control, Robotics and Informatics (ICCRI)*: 19-23. University of Danang, Danang, Vietnam. 7-9 April 2023, Danang, Vietnam.
DOI: 10.1109/ICCRI58865.2023.00011.
20. Ziegler J. G., N. B. Nichols. 1942. „Optimum settings for automatic controllers”. *Transactions of the ASME* 64(8): 759-765. ISSN: 2690-7038.
DOI: <https://doi.org/10.1115/1.4019264>.
21. Hu Guoliang, Qiangjie Li, Ruqi Ding, Gang Li. 2017. „Vibration control of semi-active suspension system with magnetorheological damper based on hyperbolic tangent model”. *Advances in Mechanical Engineering* 9(5): 15. ISSN: 1687-8140.
DOI: <https://doi.org/10.1177/1687814017694581>.
22. Vu Hai Quan, Huy Truong Nguyen, Trong Duc Nguyen. 2021. „Modeling and controlling a quarter-vehicle active suspension model”. In: *International Conference on Actual Issues of Mechanical Engineering (AIME 2021)*. 15-16 June 2021, Novorossiysk, Russia. ISSN: 1742-6588. DOI: <https://doi.org/10.1088/1742-6596/2061/1/012138>.
23. ISO 8608. 1995. *Mechanical vibration – Road surface profiles – Reporting of measured data*, 1st ed. Geneva, Switzerland: International Organization for Standardization.
24. Du Haiping, Li Weihua, Zhang Nong. 2013. „Semi-active control of an integrated full-car suspension with seat suspension and driver body model using ER dampers”. *International Journal of Vehicle Design* 63(3): 159-179. ISSN: 0143-3369.
DOI: <https://doi.org/10.1504/IJVD.2013.056133>.

25. Alfadhli Abdulaziz, Jocelyn Darling, Andrew J. Hillis. 2018. „The Control of an Active Seat Suspension Using an Optimised Fuzzy Logic Controller, Based on Preview Information from a Full Vehicle Model”. *Vibration* 1(1): 20-40. ISSN: 2571-631X. DOI: <https://doi.org/10.3390/vibration1010003>.
26. Zhao Yuli, Fengrong Bi, Hongli Shu, Linchuan Guo, Xu Wang. 2021. „Prediction of the driver’s head acceleration and vibration isolation performance of the seating suspension system using the time and frequency domain modeling”. *Applied Acoustics* 183: Art. no. 108308. ISSN: 0003-682X. DOI: <https://doi.org/10.1016/j.apacoust.2021.108308>.
27. He Haonan, Yuan Li, Lindsay Clare, Jason Zheng Jiang, Monzer Al Sakka, Miguel Dhaens, Steve Burrow, Simon Neild, and Andrew Conn, “A configuration-optimisation method for passive-active-combined suspension design”. *International Journal of Mechanical Sciences* 258: Art. no. 108560. DOI: <https://doi.org/10.1016/j.ijmecsci.2023.108560>.

Received 20.04.2026; accepted in revised form 11.07.2020



Scientific Journal of Silesian University of Technology. Series Transport is licensed under a Creative Commons Attribution 4.0 International License