

Research Article

Design and Crash Safety Assessment of an Autonomous Electric Minibus Superstructure

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Abstract:

This study presents the structural design and crash safety assessment of an autonomous electric minibus superstructure. A systematic approach combining material selection with simulation-based refinement enabled a balance of safety, efficiency, and manufacturability. ASTM A36 steel was selected for its favorable mechanical and practical properties. Targeted reinforcement of door pillars and floor cross-members improved lateral-impact resistance, reducing displacement by 65.9%, while front-end geometry adjustments enhanced pedestrian protection, keeping HIC and chest accelerations within safety limits. The final design also achieved a 7.7% reduction in structural mass without compromising integrity. These results demonstrate that combining material-informed design with parametric safety evaluation can guide the development of autonomous minibuses that are both safe and efficient for low-speed urban environments.

Keywords: *Autonomous electric minibus, Superstructure design, Side-impact performance, Pedestrian safety, Finite element analysis*

1. Introduction

Autonomous electric minibuses have emerged as a promising sustainable transport solution for short- to medium-distance routes in controlled environments such as campuses, industrial sites, and residential zones. These vehicles combine electric propulsion with autonomous driving technologies to reduce emissions, enhance energy efficiency, and improve urban mobility management [1,2]. Real-world pilot projects—such as those led by EasyMile and NAVYA—demonstrate their practical implementation across Europe and Asia, showcasing their potential to meet evolving transportation needs [3,4].

A critical challenge in deploying autonomous electric minibuses is the structural design to achieve lightweight efficiency and high crash safety. Reducing vehicle mass is essential for extending electric driving range, decreasing energy consumption, and improving acceleration and maneuverability—especially in low-speed, start-stop operational scenarios typical of such transport applications [5–6]. However, weight reduction must not compromise occupant protection, structural integrity, or pedestrian safety. Therefore, structural design must carefully balance stiffness, strength, and crash energy absorption while complying with international safety regulations, such as UNECE R107 for passenger accommodation [7] and accessibility, ECE R95 for side-impact protection [8], and ECE R66 for structural deformation control in rollover scenarios [9].

Recent research on electric vehicle structural design has largely concentrated on large-scale buses and conventional passenger cars [10–13], while the specific requirements of low-speed autonomous shuttles have received comparatively little attention. Several studies have demonstrated substantial potential for weight reduction through

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the use of advanced materials and structural optimization techniques [14]. Jung et al. [15] achieved lightweight bus frame designs using multi-material topology and cross-sectional optimization under combined torsional and bending loads. Zhou et al. [16] performed multi-objective optimization of aluminum alloy electric bus frame connectors to improve deformation during rollover. Advanced composite materials have also shown considerable promise; for instance, Jongpradist et al. [17] demonstrated superior energy absorption in sandwich composite bus components, although their high cost remains a barrier to large-scale adoption. Despite these advancements in structural and crash performance, limited research has addressed pedestrian safety considerations.

The integration of pedestrian safety considerations into autonomous vehicle design has emerged as an important area of research, particularly given the close interaction between these vehicles and vulnerable road users in urban environments. Rashid et al. [18] employed human body models to investigate pedestrian injury mechanisms in collisions involving autonomous buses, highlighting the strong influence of vehicle front-end geometry on injury severity. Han et al. [19] conducted detailed kinematic analyses of pedestrian–vehicle interactions using explicit finite element simulations, demonstrating that front-end inclination and local structural configurations significantly affect the Head Injury Criterion (HIC) and thoracic acceleration. Similarly, Martin et al. [20] reported that inclined front-end profiles can redirect pedestrian motion, mitigating direct head impacts and contributing to safer vehicle geometries. However, studies focusing on low-speed autonomous minibuses remain limited. These vehicles operate in confined or mixed-use environments where pedestrian interactions are frequent and impact speeds are low, demanding design strategies that jointly address structural integrity and pedestrian protection under realistic service conditions.

This study presents the development and evaluation of a crashworthy superstructure for an autonomous electric minibus. ASTM A36 steel was selected based on a performance-to-cost assessment using Ashby material selection charts. The structure was analyzed using finite element simulations to assess its response under service loading, side-impact, and pedestrian collision scenarios. A parametric investigation of front-end geometry was conducted to evaluate its influence on the Head Injury Criterion (HIC) and chest acceleration of pedestrians. The results provide engineering insight into front-end structural design parameters that enhance pedestrian safety while ensuring compliance with vehicle strength and crashworthiness requirements for autonomous urban transport applications.

2. Methodology

2.1 Design Specifications

The autonomous electric minibus was developed specifically for operation in controlled environments. The vehicle design integrated functional, safety, and regulatory requirements while optimizing passenger capacity and maneuverability for its intended operational settings.

The minibus configuration accommodates nine passengers with a curb weight of 800 kg and a gross vehicle weight (GVW) of 2,000 kg. Note that although the minibus is intended for nine passengers, the structure was conservatively designed to withstand occasional overload conditions (up to fifteen passengers with baggage) to ensure adequate safety margin under crowded conditions. Performance specifications include a maximum operational speed of 30 km/h and climbing capability up to 15-degree gradients, suitable for typical enclosed-environment topography. The overall vehicle dimensions are 2,000 mm width, 3,800 mm length, and 2,600 mm height, established to balance structural performance, safety compliance, and spatial efficiency within confined operational areas.

Table 1 summarizes the key design specifications, while Fig. 1 illustrates the baseline layout of the vehicle platform and superstructure, including all major structural components and sub systems.

2.2 Material Selection

A systematic material selection methodology was implemented to achieve balance between structural strength, crashworthiness, manufacturability, and economic feasibility. The superstructure was divided into four functional zones based on dominant loading conditions: roof structure for primary bending loads, wheel arch reinforcements for impact and torsional loads, front/rear sections for crash energy absorption, and side panels for lateral impact protection.

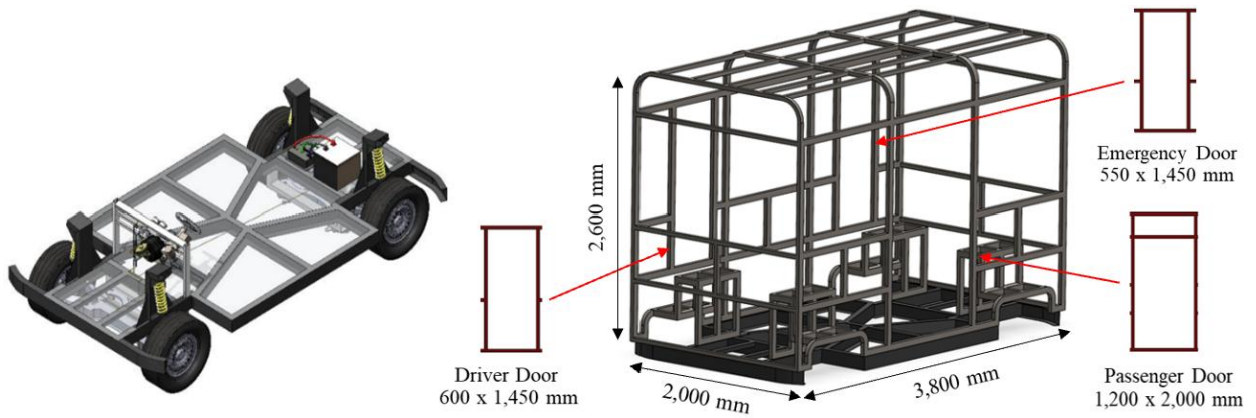


Fig. 1. Baseline platform and superstructure configuration.

Table 1: Electric autonomous minibus specification.

Specification	Value
Number of passengers	9 people
Curb weight	800 kg
Gross vehicle weight	2,000 kg
Maximum speed	30 km/h
Battery capacity	1.92 kWh
Operational range	20 km
Wheelbase	1,550 mm
Trackwidth	2,740 mm

Material performance evaluation employed Ashby methodology using two critical performance indices: $P_1 = E^{1/2}/\rho$ for stiffness-critical components, and $P_2 = \sigma^{3/2}/E$ for energy absorption applications, where E is Young's modulus, σ denotes yield strength and ρ indicates material density. The performance index P_1 is particularly appropriate for components where stiffness under bending is critical (roof structure, wheel arches), while P_2 is suitable for crash-critical zones requiring energy absorption capacity (side panels, front/rear structures).

Fig. 2 presents the Ashby plots correlating Young's modulus with density and specific modulus with specific strength for candidate materials. The lines P_1 and P_2 represent design guideline trends indicating equivalent performance and serve as reference lines for identifying materials with higher stiffness- and strength-to-density ratios, respectively. Candidate materials include aluminum alloys, conventional steel, glass fiber-reinforced polymers (GFRP), and carbon fiber-reinforced polymers (CFRP). To ensure economic viability, performance indices were normalized by cost. Table 2 summarizes the resulting performance-to-cost ratios.

Despite superior mechanical performance characteristics of CFRP and GFRP, their elevated material costs significantly limited feasibility for mass production applications. Steel grades demonstrated the most favorable performance-to-cost balance and were selected for primary structural components.

Table 2: Performance indices and cost efficiency.

Material	Cost (Baht/kg)	$P_1 = E^{1/2}/\rho$ (GPa ^{1/2})/ (kg/m ³)	P_1/Cost	$P_2 = \sigma^{3/2}/E$ (kg ^{1/2})/(m ² .s)	P_2/Cost
Al Alloy	125	98.5	1.10	61.6	0.49
Steel	30	58.4	1.94	14.0	0.54
GFRP	320	158.6	0.50	45.4	0.14
CFRP	1,700	214.0	0.13	133.6	0.08

Further steel grade evaluation incorporated manufacturability criteria including weldability, fracture toughness, and corrosion resistance. Table 3 presents the multi-criteria evaluation using a weighted scoring system (3 = excellent, 2

= good, 1 = poor) for ASTM A36, ST52-3, and DP600. ASTM A36 achieved the highest composite score due to excellent weldability and fracture toughness characteristics, despite moderate corrosion resistance. While its corrosion resistance is limited, protective treatments such as electrophoretic coating or galvanization can ensure long-term durability. Shot-peened or coated ASTM A36 steel exhibits fatigue endurance limits of approximately 200 MPa under cyclic bending [21], which is sufficient for low-speed EV superstructures where operational stresses are substantially lower. Consequently, ASTM A36 was selected as the primary material for the minibus superstructure. The mechanical properties of selected materials for subsequent finite element analysis are presented in Table 4.

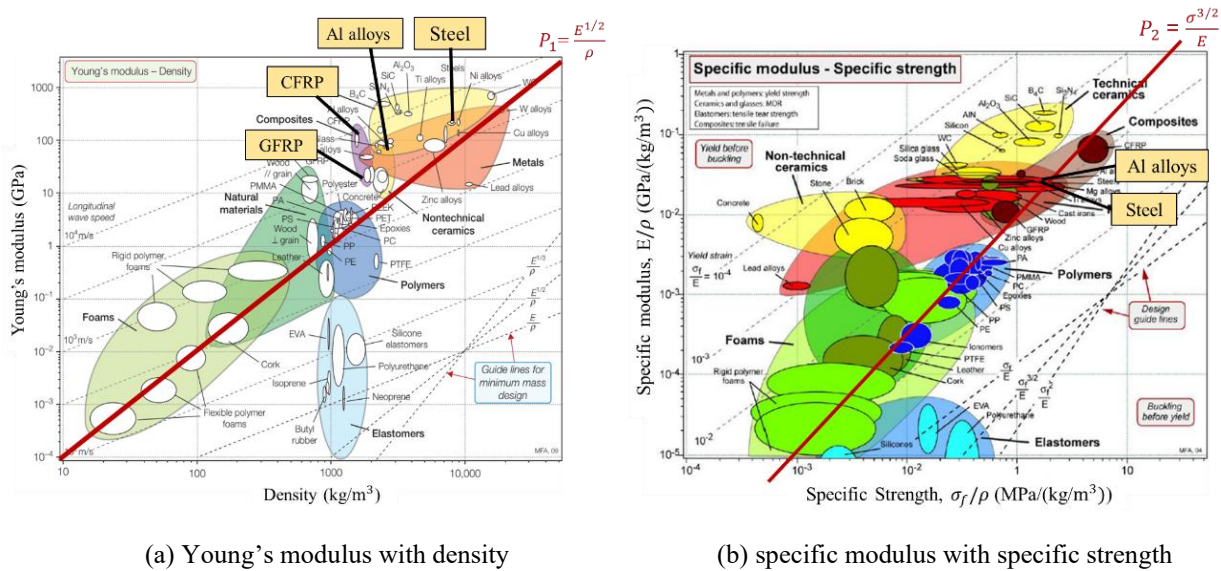


Fig. 2. Ashby plots for material selection.

Table 3: Multi-criteria evaluation of steel grades for superstructure application.

Material	Corrosion Resistance (weight = 0.33)	Weldability (weight = 0.33)	Fracture Toughness (weight = 0.33)	Total Score
A36	1	3	3	2.31
ST52-3	2	2	2	1.98
DP600	3	1	1	1.65

Table 4: Material properties of the minibus.

Material Properties	Steel A36	ST52-3	Polypropylene	Tempered Glass
Density (kg/m ³)	7,800	7,800	950	2,600
Young's modulus (GPa)	210	200	1.325	70
Yield strength (MPa)	250	350	31	210
Ultimate tensile strength (MPa)	350	531		

2.3 Operational Load Cases

To ensure structural robustness under typical operating conditions, four primary load scenarios were evaluated: bending stiffness, torsional stiffness, lateral loading from cornering maneuvers, and longitudinal load from braking. Finite Element Analysis (FEA) was conducted using HyperWorks software, with results as shown in Fig 3. The structural model was discretized primarily using four-node Belytschko–Tsay shell elements, which provide an efficient balance between computational cost and accuracy for thin-walled components. The nominal shell element size was 20 mm, determined from a mesh convergence study on global deformation and maximum intrusion criteria with 65,755 nodes and 64,442 elements. Boundary conditions were defined to mimic real operational constraints, including wheel-ground interactions and joint fixities. During quasi-static loading, the minibus was supported at the suspension mounting points with translational constraints along the global z-axis. Material constitutive behavior for

ASTM A36 steel was modeled using MAT_24 Piecewise Linear Plasticity using the properties described in Table 4. Note that strain-rate effects were not included in this study, as the simulated impact velocities (≤ 30 km/h) correspond to relatively low strain rates ($<10^1$ s⁻¹), for which the mechanical response of mild steel remains nearly rate-insensitive. This assumption yields a conservative estimate of deformation [22, 23].

For the evaluation of bending stiffness, a concentrated load of 5 kN is applied to the platform to quantify the maximum deflection. Torsional stiffness is assessed through the application of asymmetric wheel loads of 5 kN to the left and right wheels in opposing directions, with the resulting angular deformation measured to determine the torsional rigidity. Lateral loading conditions are simulated by applying cornering forces equivalent to a lateral acceleration of 1g. Longitudinal loading scenarios are examined by imposing forces corresponding to a maximum braking deceleration of 1.5g. Displacement contours and stresses induced under these loading conditions are illustrated in Fig. 3. Model validity was verified through a mesh convergence study, in which further refinement beyond the selected 20 mm element size changed the global deformation by less than 2%

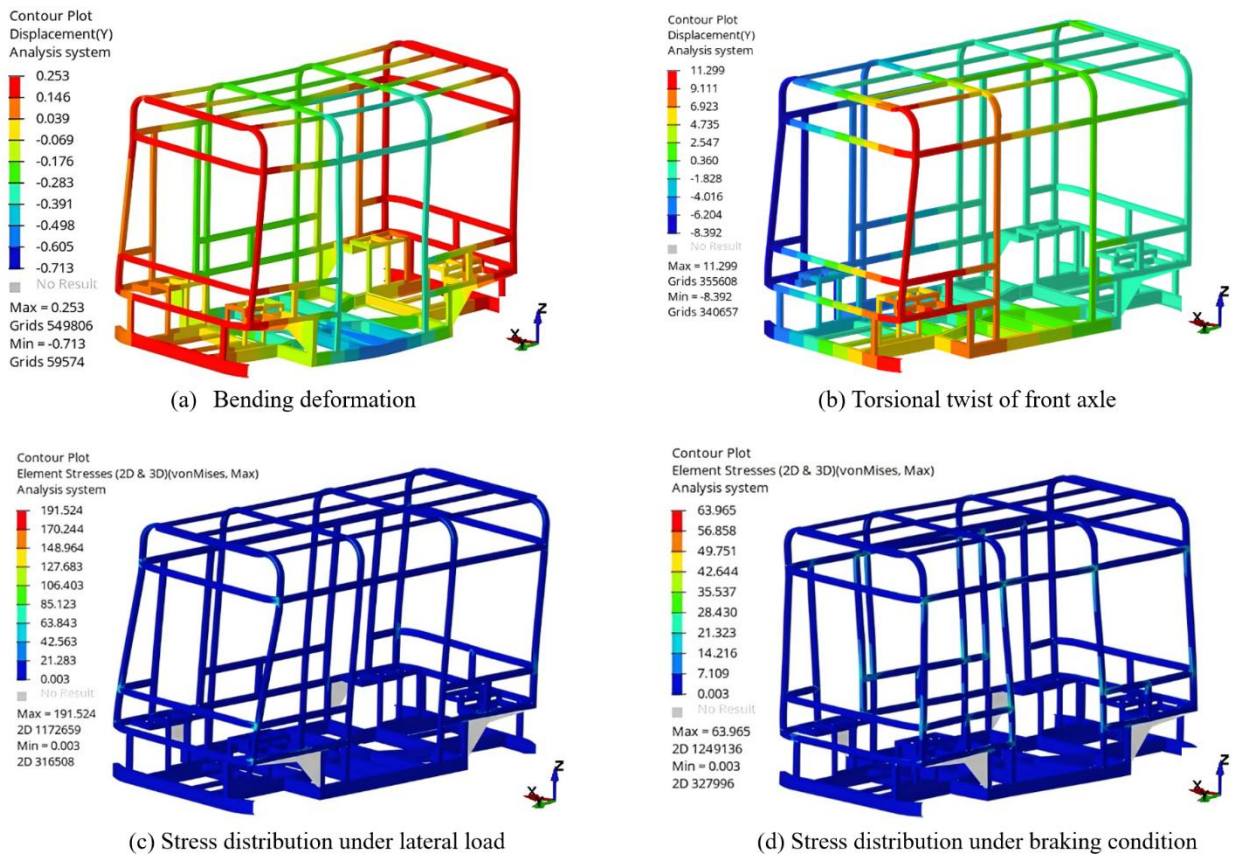


Fig. 3. FEA results under operational load cases

The FEA results for the baseline configuration exceeded all performance thresholds:

- Bending stiffness: 8,264.5 N/mm, (Minimum requirement: 5,000 N/mm)
- Torsional stiffness: 8,915.8 N·m/degree (Requirement: $> 7,200$ N·m/degree)
- Lateral and braking safety factors: 3.6 and 10.2, respectively (Minimum: 2.0)

These results indicate that the baseline structure exhibits sufficient rigidity and durability to withstand routine service loads without excessive deflection or instability. The substantial safety margins provide confidence for subsequent optimization procedures while maintaining structural reliability.

3. Crashworthiness and Injury Evaluation

The crashworthiness of the autonomous minibus was evaluated under two critical scenarios: side impact from another vehicle and frontal collision with a pedestrian. All simulations were performed using explicit finite element analysis via LS-DYNA to investigate nonlinear and dynamic crash simulations.

3.1 Side impact performance

Side impact performance was evaluated to assess occupant protection during lateral collisions, representing one of the most critical crash scenarios for passenger vehicles. The simulation employed a validated Toyota Yaris finite element model from the National Crash Analysis Center (NCAC) database [24] as the striking vehicle. The Yaris was positioned to impact the minibus laterally at the door centerline, identified as the most structurally vulnerable region due to the presence of large door openings and reduced load-path continuity. To ensure realistic boundary conditions, the minibus model incorporated wheel and floor contacts to maintain stability and accurately replicate vehicle support conditions.

Contact interactions between all components were defined using the AUTOMATIC_SURFACE_TO_SURFACE algorithm with a penalty-based formulation. A friction coefficient of 0.2 was assigned for steel-to-steel interfaces, while a higher coefficient of 0.7 was applied to the wheel–floor contact to represent tire–ground friction. The crash simulation of the passenger car model was validated against the results reported by Kongwat et al. [25], confirming the model’s reliability.

For the side-impact scenario, the striking vehicle was configured to travel at 30 km/h and aligned to impact the center of the side-door structure. Fig. 4a illustrates the overall simulation setup, showing the relative positioning and impact trajectory of both vehicles. Mesh convergence has been confirmed. Energy balance and hourglass energy were monitored to ensure numerical stability, maintaining non-physical energy below 5% of total internal energy.

The residual space, defined per ECE R66, was monitored using internal reference nodes and planes to measure maximum inward deformation. Finite element results revealed maximum lateral deformation of 270 mm at the impact zone, resulting in 120 mm intrusion into the defined residual space. This intrusion level exceeded acceptable limits for passenger safety, indicating insufficient lateral structural stiffness in the baseline configuration.

Fig. 5 presents the stress contour showing structural damage and energy absorption characteristics of individual structural components. Plastic deformation patterns were concentrated primarily in the platform side beams and door pillar structures, with these components providing the majority of energy absorption during impact. The platform absorbed the largest portion of internal energy due to its alignment with the striking vehicle’s bumper height, effectively capturing the dominant energy absorption mechanism in the minibus structure. The energy dissipation mechanisms included plastic hinge formation at beam connections and progressive crushing of structural elements. The analysis revealed that while the side structure absorbed substantial energy, deformation patterns indicated inadequate load distribution and insufficient structural stiffness to meet safety requirements.

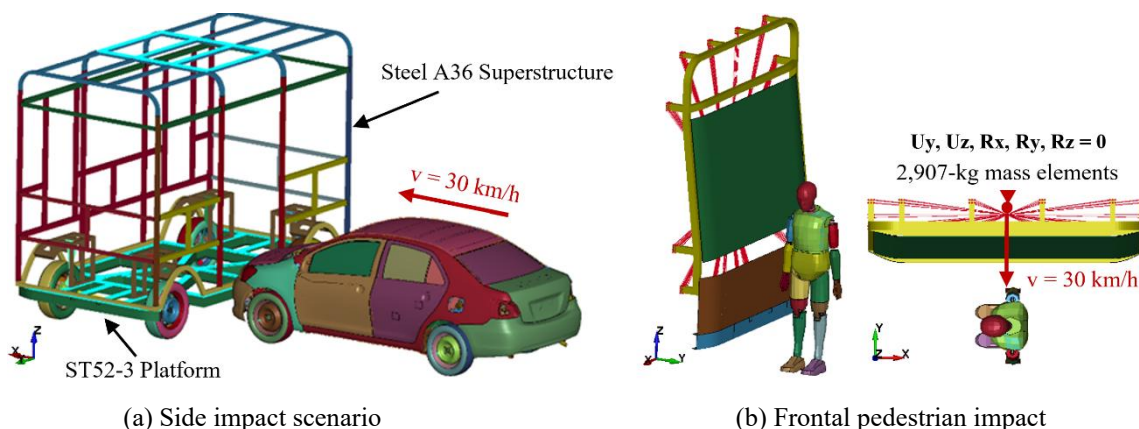


Fig. 4. Crash simulation setup configurations.

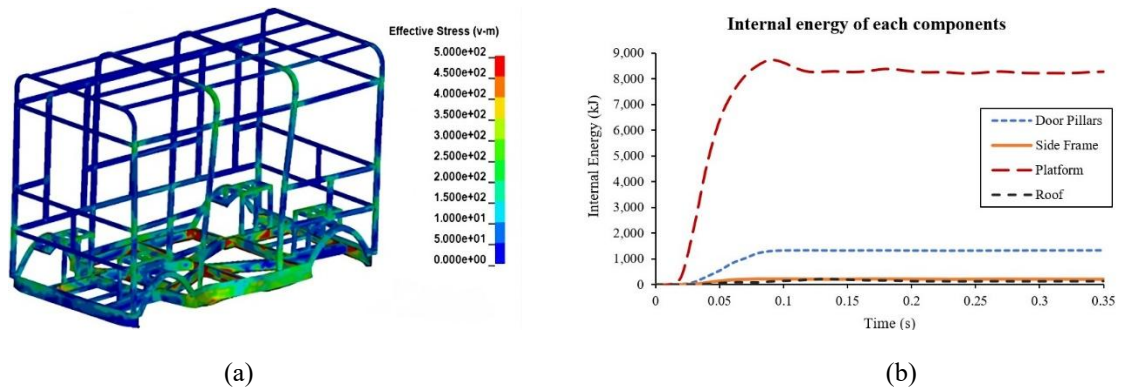


Fig. 5. (a) Side impact stress contour and (b) energy absorption by structural components

3.2 Front impact to pedestrian

Pedestrian safety evaluation was conducted to assess injury risk in frontal collision scenarios, addressing the critical requirement for vulnerable road user protection in autonomous vehicle applications. The analysis utilized a validated Hybrid III 50th percentile male anthropomorphic test device (ATD) model in standing posture, positioned centrally in the vehicle's forward trajectory path.

The minibus model was simplified by representing only the front structural section while maintaining appropriate mass distribution to enhance computational efficiency. The total vehicle mass was modeled using lumped mass elements of 2907 kg, corresponding to the structure excluded from the finite element model, and connected to the front structure through rigid links (Fig. 4b). This approach preserved accurate inertial characteristics while significantly reducing computational cost. The validity of the simplified model was confirmed by verifying the equivalence of impact energy at the onset of collision and the consistent conservation of total energy throughout the crash event. The reference point was constrained to translate solely in the frontal direction (u_x), simulating a minibus traveling at its maximum operational speed of 30 km/h toward the stationary pedestrian dummy.

Pedestrian injury assessment was conducted using two primary biomechanical criteria: Head Injury Criterion (HIC) at the dummy head's CG and chest acceleration. These metrics were obtained from accelerometer sensors embedded within the dummy's head and thoracic regions, providing direct measurement of impact severity.

The HIC_{36} value was calculated using the standard formulation based on the resultant acceleration-time history over the most critical 36-millisecond interval:

$$HIC_{36} = \left[\frac{1}{t_2 - t_1} \int_{t_1}^{t_2} a dt \right]^{2.5} (t_2 - t_1) \quad (1)$$

where a represents the resultant head acceleration (g), and t_1 and t_2 define the time interval that maximizes the HIC value. Chest acceleration was determined by identifying the peak resultant acceleration recorded at the thoracic sensor throughout the impact duration. Acceleration-time histories and dummy kinematics were analyzed to capture the dynamic response of the head and thorax (Fig. 6). Acceleration signals were processed using a SAE 108 Hz filter before HIC_{36} calculation.

In accordance with established safety criteria, the HIC_{36} should not exceed 1,000, while the peak chest acceleration should remain below 60g. These thresholds are widely recognized in pedestrian safety assessments and serve as benchmarks for determining the severity of head and thoracic injuries. The baseline results exceeded these limits with a HIC_{36} of 1,336 and chest acceleration of 65.9g, indicating a high likelihood of severe injuries and potentially life-threatening consequences. The analysis suggests that the baseline front-end geometry induces excessive impact severity due to inadequate energy absorption and unfavorable contact conditions. A brief sensitivity evaluation confirmed that refining the mesh size from 20 mm to 15 mm altered the maximum deformation, chest acceleration of the passenger dummy by less than 5.2%, validating the adequacy of the applied discretization and the reliability of the simulation results.

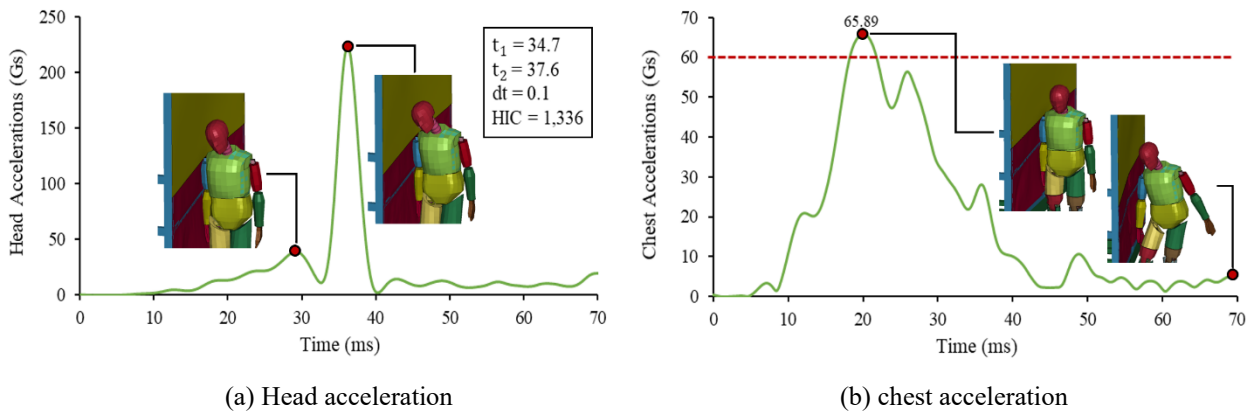


Fig. 6. Time histories of pedestrian injury indices.

4. Structural Redesign and Improvements

4.1 Design modification strategy

Following the baseline assessment, targeted design modifications were implemented to enhance structural performance, improve crash safety, and reduce vehicle mass as displayed in Fig. 7a. These revisions addressed failure mechanisms observed during crash simulation impact resistance and structural integrity. The superstructure base was removed to integrate the main frame directly with the platform. This reduced structural complexity, improved load transfer efficiency, and lowered mass. Wheel arch structures were redesigned with continuous integration to the side frame assemblies, significantly improving structural continuity and lateral stiffness. This integrated design eliminated stress concentration points at traditional joint locations while providing enhanced load distribution during impact scenarios.

A central floor cross member was strategically added to increase lateral structural stiffness and improve energy absorption during side impact events. Side pillar thickness was increased from 2 mm to 3 mm based on stress analysis results from the baseline configuration. This modification significantly enhanced deformation resistance during lateral impacts while maintaining structural weight efficiency. The increased thickness provided improved energy absorption capacity and reduced peak stress concentrations in critical impact zones.

The front structure's geometry was parametrically studied to minimize dummy head and thoracic injuries during frontal collisions (Fig. 7b). This study focused on varying the rake transition height (h) defined as the vertical offset where front-end curvature begins, and the incline angle of the front section (α) which is known to influence the dummy's kinematic response during impact. The selection of these parameters was informed by established pedestrian safety research, highlighting their significant impact on injury severity.

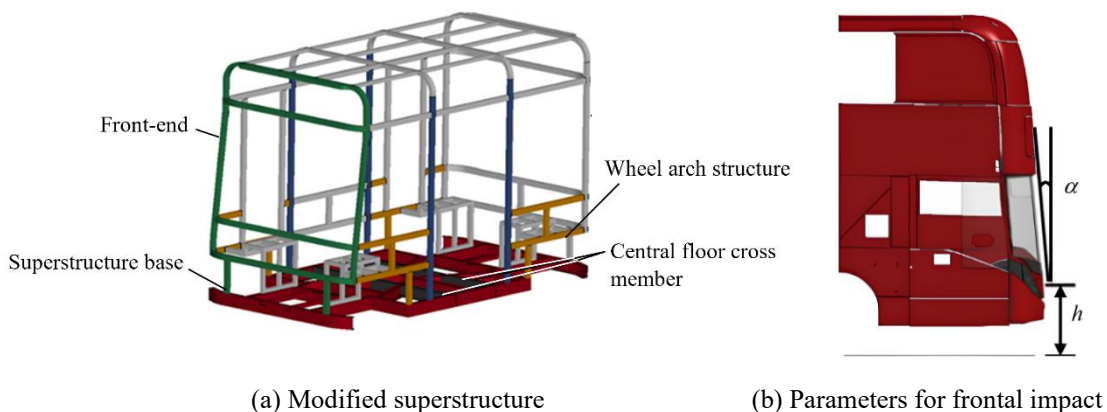


Fig. 7. Structural design improvements.

4.2 Side impact performance of the improved model

The improved structural configuration demonstrated significant performance improvements in side impact scenarios. Maximum lateral deformation was reduced to 92.2 mm, representing a 65.9% improvement compared to the baseline configuration (270 mm). This deformation level satisfies ECE R66 residual space requirements and provides adequate passenger protection.

The structural modifications resulted in a 7.7% reduction in total superstructure mass (from 312.5 kg to 300 kg) while achieving enhanced safety performance. The weight reduction was primarily attributed to elimination of redundant structural elements and improved material distribution in high-stress regions.

Energy absorption characteristics were significantly improved through the enhanced structural design. The increased side pillar thickness and integrated wheel arch design provided more effective energy dissipation mechanisms, with plastic deformation distributed more uniformly across structural elements rather than concentrated at weak points.

4.3 Pedestrian-friendly front-end design

A pedestrian-friendly front-end structure was developed to address injury risks associated with potential autonomous system failures in obstacle detection. The design improvement focused on reducing direct impact forces through geometric modifications that promote pedestrian deflection and controlled energy absorption. The parametric study evaluated four rake transition heights ($h = 1,250\text{ mm}, 1,300\text{ mm}, 1,350\text{ mm}, 1,400\text{ mm}$) and four incline angles ($\alpha = 0^\circ, 3^\circ, 5^\circ, 7^\circ, 9^\circ$), resulting in 20 design configurations to determine optimal parameter combinations.

Fig. 8 presents the relationship between HIC_{36} and variations in h and α . The 0° configuration was excluded from Figure 8 because its HIC and chest acceleration values were substantially higher than the other cases, obscuring the comparative differences between the inclined configurations. The results demonstrate that HIC values are significantly elevated at low α ($0^\circ, 3^\circ, 5^\circ$), regardless of rake height configuration. Conversely, increasing the α to 7° consistently reduces HIC values across all height configurations. Moreover, lower h (1,250 mm) generally demonstrate improved performance, though the α effect proves more pronounced in HIC reduction.

In the baseline configuration ($\alpha = 0^\circ$), the front structure was nearly vertical with a ground clearance of about 60 cm, causing direct head impact against the rigid superstructure and resulting in high HIC values. In contrast, the improved designs reduced the ground clearance to 45 cm and introduced an inclined front profile, which redirected the pedestrian dummy’s motion upon contact. This geometric modification promoted body rotation and glancing impacts, effectively reducing both HIC and chest acceleration.

The inclined front structure helped prevent the vehicle’s front frame from directly striking the dummy at the initial moment of impact, unlike the baseline design with a vertical front profile. The increased alpha angle also allowed a longer rotational path for the upper body, resulting in a more oblique and less severe impact to the head and chest. In the final stage of the impact sequence, the dummy remained in contact with the vehicle’s front structure before being subsequently projected forward.

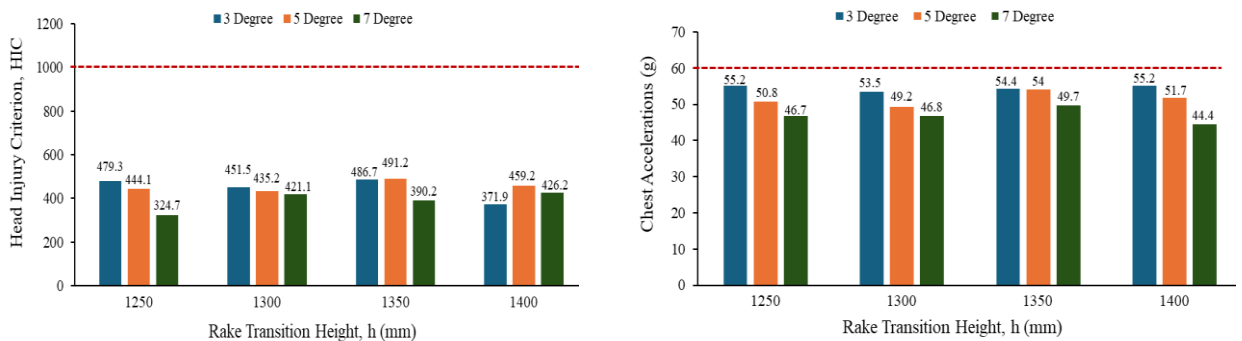


Fig. 8. HIC_{36} and chest acceleration as functions of alpha angle and rake transition height.

The optimal configuration featuring a 7° alpha angle and 1,250 mm rake transition height achieved the lowest injury criteria among all tested configurations: $HIC_{36} = 324.7$ and chest acceleration = 46.7 g. Both values satisfy established pedestrian safety thresholds, representing a substantial improvement over the baseline design.

The parametric study evaluated three alpha angle configurations (3°, 5° and 7°) at a fixed rake transition height of 1,250 mm across multiple impact velocities, as exemplified in Fig. 9. Results demonstrate that inclination angle is the primary factor governing injury severity, with rake transition height providing a secondary influence. Figure 9 presents the variation of injury criteria with impact velocity for the three front-end configurations. The optimized 7° design consistently maintained HIC and chest acceleration within safety thresholds across all tested velocities, demonstrating stable and reliable model performance. This consistent trend across velocity variations supports the robustness and accuracy of the simulation results.

At low impact speeds (30 km/h), all configurations met the required safety thresholds. However, at higher speeds, the 3° and 5° configurations exceeded critical safety limits for chest acceleration, with HIC values also approaching or surpassing thresholds, indicating increased injury risk. The 3° configuration produced the most severe violations, with HIC values reaching 1,847.1 (84% above the 1,000 threshold) and chest acceleration peaking at 112 g (87% above the 60g limit) at 50 km/h. The 5° configuration showed moderate improvement but still exceeded safety limits, recording HIC of 1,045 and chest acceleration of 104 g at maximum speed.

Impact velocity significantly amplifies injury severity for shallow alpha angles. The 3° configuration exhibited sharp increases in both injury metrics with velocity due to its nearly vertical geometry, which restricts pedestrian kinematics and promotes direct head contact. The 5° configuration demonstrated reduced velocity sensitivity but maintained elevated injury levels due to insufficient geometric modification.

It should be noted that this study employed only the Hybrid III 50th percentile male dummy as a baseline case. Future studies should consider a broader range of anthropometric models to evaluate the effects on other potential pedestrian groups.

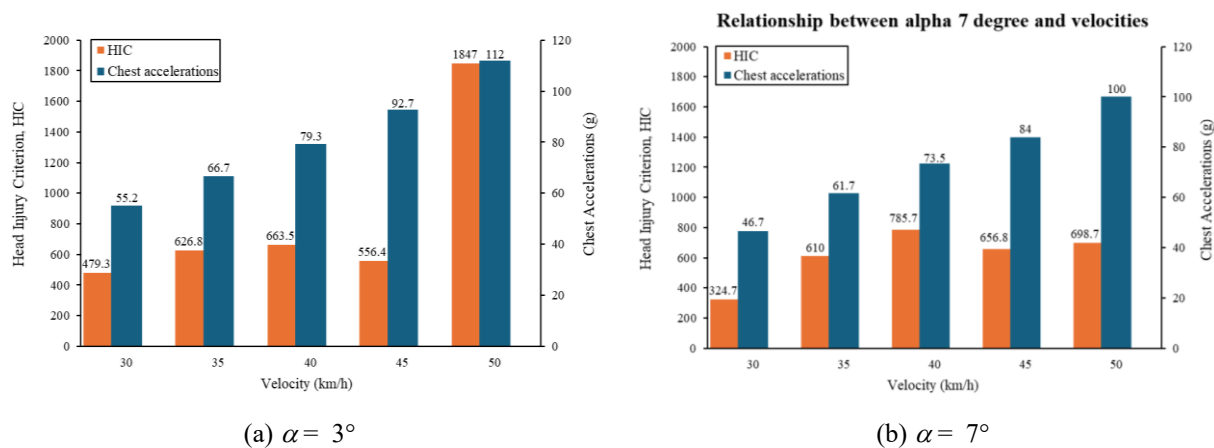


Fig. 9. Influence of front-end geometry on injury criteria across different impact velocities

Only the 7° configuration achieved consistent safety compliance, with HIC values ranging from 324.7 to 698.7 and chest acceleration below 100 g across all speeds. This performance results from enhanced pedestrian kinematics: the larger incline angle facilitates upper-body rotation and glancing head impacts, effectively distributing impact energy and preventing direct head-to-surface contact.

4.4 Integrated performance assessment

The improved superstructure design maintained adequate performance under operational loading conditions despite the 7.7% mass reduction. While bending stiffness decreased by 15.1% (from 8,264.5 to 7,018.2 N/mm), the value remained above the minimum requirement of 5,000 N/mm. Significant improvements were achieved in other operational performance metrics. Torsional stiffness increased by 73.2% (from 8,915.8 to 15,441 N·m/degree),

substantially exceeding the minimum requirement and providing enhanced vehicle stability during cornering maneuvers. Braking and lateral safety factors demonstrated substantial improvements of 38.5% and 172.4% respectively, indicating enhanced structural robustness under dynamic loading conditions. These improvements result from enhanced load distribution and increased structural continuity in the redesigned configuration.

The integrated design modifications delivered substantial safety improvements across both crash scenarios. Side impact performance showed a 65.9% reduction in maximum deformation (from 270 mm to 92.2 mm), bringing intrusion levels within acceptable safety limits.

Pedestrian safety performance demonstrated significant improvements with a 75.7% reduction in HIC_{36} (from 1,336 to 324.7) and 29.1% decrease in chest acceleration (from 65.9 to 46.7 g) at 30 km/h frontal crash. Both injury criteria now comply with established safety thresholds, representing a fundamental improvement in vulnerable road user protection.

The proposed superstructure is technically feasible to manufacture using conventional low-carbon-steel fabrication methods without specialized tooling. Based on average Thai market rates, the total manufacturing cost is estimated at approximately 19,000 THB. This includes the raw material of about 6,800 THB (≈ 22.7 THB/kg) [26], welding cost of around 7,300 THB for 30 hours at 243 THB/hour [27], and cutting, forming, and assembly cost of roughly 5,000 THB. These estimates indicate that the design is economically viable for prototype and low-volume production of electric minibuses.

The comprehensive structural design of the autonomous minibus successfully fulfilled the primary objective of enhancing safety performance for both passengers and pedestrians during crash events. The final configuration demonstrates that systematic modifications can simultaneously improve crashworthiness, reduce vehicle mass, and maintain operational performance requirements.

5. Conclusion

This study presents an improved superstructure design for an autonomous electric minibus that effectively integrates structural efficiency with comprehensive safety performance. By integrating systematic material selection with simulation-based structural refinement, the design achieved a 7.7% reduction in mass while substantially improving lateral-impact and pedestrian protection. Side-impact resistance was significantly enhanced through targeted reinforcement of the door pillars—increasing thickness from 2 mm to 3 mm—and the addition of floor cross-members, resulting in a 65.9% reduction in lateral displacement. Pedestrian protection was improved through parametric optimization of the front-end geometry. The adoption of a 7° alpha angle and a rake transition height of 1,250 mm led to a reduction in HIC and chest acceleration within international safety limits. These results demonstrate the potential of a performance-driven structural design approach to achieve safety compliance and material efficiency in autonomous electric minibuses intended for operation in low-speed, controlled environments.

Nomenclature

a	resultant head acceleration, m/s^2
E	Young's modulus, MPa
h	rake transition height, m
HIC_{36}	Head Injury Criterion
P_1, P_2	performance indices
t_1, t_2	starting and ending time interval, s
α	incline angle of the bus front, degree
ρ	material density, kg/m^3
σ	specific strength, MPa

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