Crashworthiness Analysis of Topology-Optimized Jeepney Chassis Frame Through Explicit Dynamics Simulations

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Abstract: The Philippine jeepney remains to be the backbone of the country's public transportation network. However, as of today it still harbors safety concerns due to high crash rates and poor crashworthiness due to the lack of dedicated crumple zones. This leads to undesirable transfer of extreme frontal impact energies directly into the passenger cabin. To address this issue, topology optimization is applied to the front-end of the chassis with the goal of increasing energy-absorbing efficacy while also aiding in cost by removing material. Using frontal impact explicit dynamics simulations, baseline and optimized designs were evaluated under two distinct impact energies: 12.9 kJ and 29.4 kJ. Results show that the baseline traditional chassis suffered mid-section buckling implicating cabin intrusion while the optimized chassis developed front crumple zones. These redesigns consistently increased energy absorption, reduced peak forces by 46.66% up to 59.18%, and increased crumple distances by 32% up to 276.55%.

Keywords: Topology Optimization, Crashworthiness, Philippine Jeepney, Public Utility Vehicle Modernization Program

1. INTRODUCTION

Public transportation remains as the leading mode of transportation in the Philippines, covering 80% of overall urban trips. Of this, 40% are provided by jeepneys, solidifying its significant role as the backbone of urban and rural transportation for Filipinos (Dimalanta et al., 2023). However, their involvement in road crashes has raised serious safety concerns. Data from the Metro Manila Accident Recording and Analysis System (MMDA, 2019) reported 10,296 public utility jeepney crashes in the region alone in 2019, resulting in 40 fatalities and 1,817 injuries. An existing assessment of jeepneys revealed that its structural design is not adequately engineered to effectively absorb crash energy, which significantly increases the risk of fatal injuries for passengers (Rogelio et al., 2018).

To address these safety deficiencies, the Philippine government implemented the Public Utility Vehicle Modernization Program (PUVMP) where it mandates strict adherence to the Philippine National Standards (PNS) for dimensions and environmental compliance. While ideally sounding, various stakeholders have shown resistance to the said program. Estipular (2020) and Dimalanta et al. (2023) mentioned that the initiative has been fixated on replacing old jeepneys with modern jeepneys fueled by EURO 4-compliant diesel engines, liquefied petroleum gas, and electric or hybrid sources. These replacements financially burden the drivers

and operators as modern vehicles cost around 10 times more than the existing ones despite local assembly.

In modern vehicle design, passive safety systems such as airbags, seatbelts, and crumple zones are widely employed to mitigate crash consequences and protect the vehicle occupants (ACEA, 2019; Lopes et al., 2024). Central to these systems is the concept of crashworthiness that focuses on designing vehicle structures capable of protecting occupants during a crash by absorbing collision kinetic energy and minimizing occupant cabin intrusion (Li et al., 2017). To achieve this, the crumple zones play a crucial role in absorbing energy during impact by undergoing controlled deformation governed by structural design (Du Bois et al., 2004; Lopes et al., 2024). Fundamentally, the function of a crumple zone is grounded in the Work-Energy Theorem, which states that the kinetic energy of a moving vehicle must be dissipated as work during a crash event. This is expressed by:

$$W = KE = Fd \tag{1}$$

where,

W : work,
KE : kinetic energy,
F : force, and
d : displacement.

Because work is the product of impact force and deformation distance, increasing the distance over which the structure safely deforms is critical. A rigid chassis forces energy dissipation over a near-zero interval, causing a spike in reaction force and fatal deceleration levels. Conversely, well-engineered crumple zones facilitate controlled plastic deformation over an extended distance, reducing the magnitude of the force transmitted to occupants (Catalan, 2017). Unfortunately, traditional jeepney designs completely lack dedicated crumple zones (Clavecillas, 2017), with front bumpers and grilles serving purely aesthetic purposes rather than safety mechanisms (Bacero, 2009).

A promising computational approach for enhancing structural safety is topology optimization. This design tool maximizes structural performance by optimizing material distribution within a given design domain while satisfying specific load, performance, and volume constraints. Unlike size and shape optimization, which are limited to modifying predefined geometries, topology optimization is not constrained by an initial design, allowing for a significantly expanded design space (Rong et al., 2022; Tang et al., 2024).

Topology optimization has also been proven effective in improving the crashworthiness of modern vehicles. A study by Silva et al. (2024) focusing on heavy passenger vehicles utilized topology optimization to redesign the frontal crash box component. The resulting optimized geometry demonstrated superior energy absorption capabilities and minimized intrusion during frontal impacts, validating the method's ability to improve passenger safety metrics even within complex, non-linear crash environments.

Despite these international advancements and the push for modernization under the PUVMP, applying topology optimization to Philippine jeepneys remains unexplored. This leaves a critical gap in developing passive safety features tailored to the local context. A more grounded approach to modernization is urgently required, as outright vehicle replacement imposes severe financial strains on drivers and operators. Furthermore, local builders like Sarao Motors face material and manufacturing constraints that preclude complex safety systems like advanced polymer foams. Given that jeepneys will realistically remain on the road for years and heavily rely on surplus materials rather than standardized parts production, topology optimization offers a pragmatic, cost-effective pathway. It maximizes the structural safety of

accessible local materials without requiring an overhaul of the existing manufacturing ecosystem.

This study then applies topology optimization to increase the energy-absorbing efficacy of the front-end of the chassis and evaluate its crashworthiness against the baseline traditional design and accepted standards. Using ANSYS Structural Optimization, the optimization process is applied across the traditional and modern chassis designs retrieved from an existing experimental study literature and field measurements. These optimized structures are subjected to ANSYS Explicit Dynamics to analyze their performance metrics, specifically energy absorption, peak impact force, crumple distance, total deformation, and equivalent stress and strain relative to the baseline frame on different impact energies.

2. METHODOLOGY

This study utilizes a comparative, simulation-based design to evaluate and enhance the crashworthiness of the Philippine jeepney chassis structure. The evaluation is conducted using topology optimization and explicit dynamics simulations within the ANSYS Workbench environment.

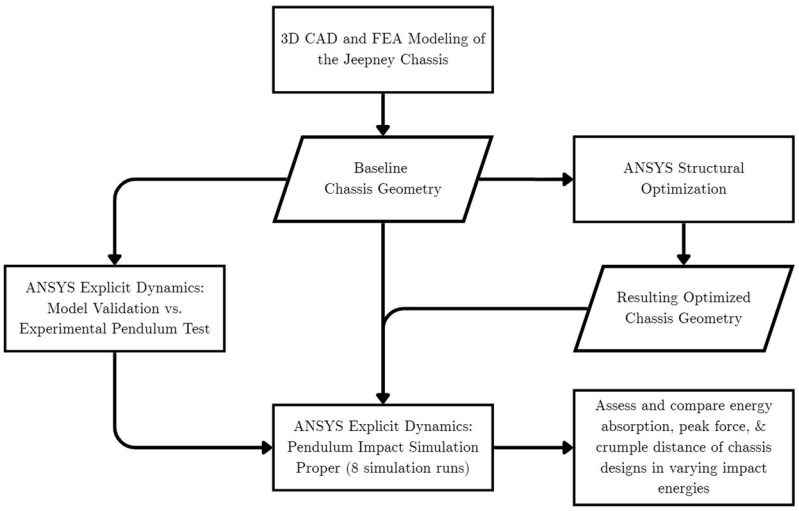
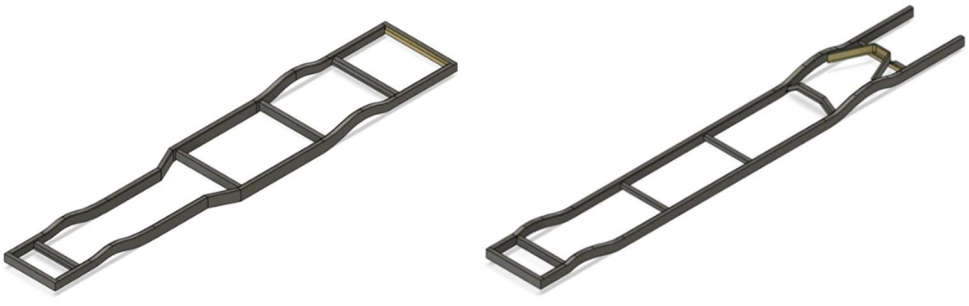


Figure 2.1. Methodology process flowchart

The research evaluates two primary baseline models: the traditional jeepney chassis and the modern jeepney chassis. Both these chassis were modeled from actual chassis retrieved from a well-known jeepney manufacturer in the Philippines called Sarao Motors. Figure 2.2 shows the final 3D CAD models of both configurations side-by-side.



(a) (b)

Figure 2.2. CAD models of Baseline (a) Traditional and (b) Modern Chassis

As visually evident in the geometric models, a critical design modification in the modern chassis is the complete elimination of the outward width offset profile in the middle of the chassis, at the passenger area. In the traditional chassis, this portion has been identified as the weakest structural area with high stress concentration, highly susceptible to catastrophic rupture and permanent yielding under impact loads. By eliminating this bend and straightening the profile, the designer aims to improve structural integrity of the chassis.

The geometries were then imported into the ANSYS Workbench environment, where the chassis was assigned to the material properties of mild structural steel. The foundational material properties were obtained from the experimental data of Catalan (2017) on the jeepney chassis material from Sarao Motors whose tested material was identified to be highly similar to the standard AISI A36 mild steel. To establish the baseline elastic behavior of the chassis, general mechanical properties were retrieved from readily available, established data for AISI A36.

Additionally, to accurately capture energy dissipation through plastic deformation during a crash event, a multilinear isotropic hardening model was utilized. The critical plasticity parameters, specifically the yield strength and ultimate tensile strength, were directly extracted from the physical tensile testing results published by Catalan (2017). Furthermore, the specific tabular data used to define the multilinear isotropic hardening model within ANSYS was mathematically derived from the experimental true stress-strain plot obtained from those same tensile tests. A summary of these assigned material properties is detailed in Table 3.1.

Table 2.1. Material Properties for AISI A36 Chassis

Material Property	**Value**	**Unit**
Density	7850	kg/m^3
Young's Modulus	200	GPa
Poisson's Ratio	0.26	-
Bulk Modulus	138.89	GPa
Shear Modulus	79.37	GPa
Yield Strength	275.59	MPa
Ultimate Strength	639.40	MPa

These initial designs are then processed through ANSYS Structural Optimization to generate optimized geometries driven by a specified loading condition, objective function, and response constraints. The loading condition used is based on an equivalent static load derived from the average force the chassis will experience upon pendulum impact, adopting the experimental conditions established by Catalan (2017).

To govern the material removal process, the chassis geometry was meticulously divided into inclusion (design) and exclusion (non-design) regions using the named selection feature in ANSYS. In accordance with modern automotive safety principles, the design space was strictly limited to the front-end chassis structure extending in front of the engine compartment. The objective in this region was to synthesize an energy-absorbing crumple zone.



Figure 2.3. Jeepney Chassis Frame installed with Main Body Panels

Conversely, the remaining chassis structure, particularly under the passenger cabin shown in Figure 2.3, was defined as the exclusion region to prevent the solver from removing material in critical assembly areas, thereby maintaining the integrity of the occupant survival space (Du Bois et al., 2004). These design and exclusion regions within ANSYS can be seen applied in Figure 2.4.

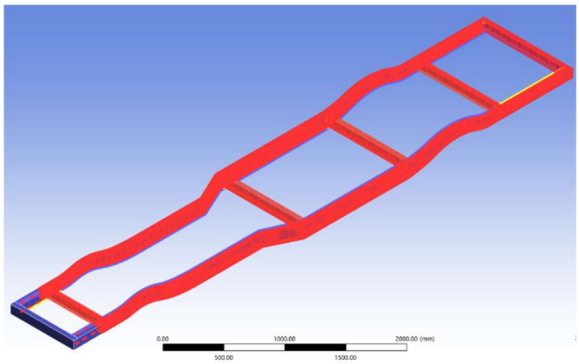


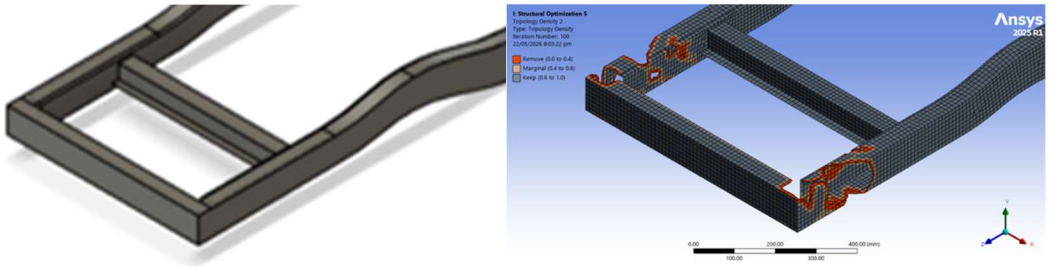
Figure 2.4. Chassis Design Region and Exclusion Region

The primary objective of the solver was to minimize the volume of the design space, constrained by a volume retention range of 50% to 90%. Specifying this range provides the solver with sufficient flexibility to iteratively remove material, while ensuring that enough structural mass remains to absorb and dissipate the kinetic energy of an impact (Rong et al., 2022). Furthermore, a symmetry design constraint was applied along the YZ plane of the inclusion region to contribute to manufacturability and balanced energy dissipation during a frontal collision.

To further guarantee occupant safety, a local von-Mises stress response constraint was implemented specifically within the exclusion region. The stress was capped at 275.59 MPa, corresponding directly to the yield strength of the AISI A36 mild structural steel. Applying response constraints to non-design spaces is heavily supported by the framework of De Leon et al. (2015) for stress-based topology optimization. In their methodology, boundary constraints are mathematically managed by defining input and output ports. By analogy, the front-end design region acts as the input port where the external force is applied, while the outward width offset of the traditional chassis at its middle section serves as the output port where the force is expected to react. Therefore, applying the stress constraint to the chassis exclusion region optimizes the structure to prevent plastic deformation to the region of the chassis within the passenger section. A summary of the topology optimization settings is provided in Table 2.2.

Table 2.2. Summary of Topology Optimization Settings

Parameter	Setting
Optimization Method	Density-Based Topology Optimization
Objective Function	Minimize Volume
Response Constraint 1 at Design Region	Volume (50% Min – 90% Max Volume Retained)
Response Constraint 2 at Exclusion Region	Local von-Mises Stress (≤275.59 MPa)
Design Constraint 1 at Design Region	Symmetry (YZ Plane)



(a) (b)

Figure 2.5. Traditional Chassis (a) baseline and (b) topology optimized

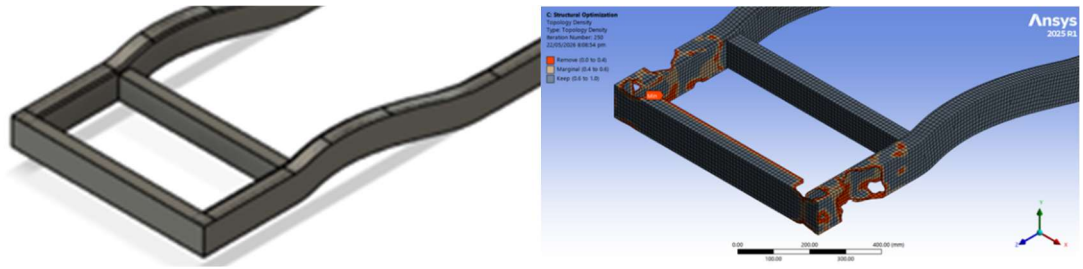


Figure 2.6. Modern Chassis (a) baseline and (b) topology optimized

Figure 2.5 and Figure 2.6 illustrate the raw, software-generated topology optimization results for the traditional and modern chassis, respectively. As illustrated, the optimization algorithm drastically altered the structural profile of the front-end design regions while leaving the rest completely untouched. However, while these raw generated structures are mathematically optimal, they cannot be directly utilized for final crashworthiness analysis. As is common in density-based topology optimization, the raw output is generated as a faceted, jagged mesh rather than a smooth geometric solid. These jagged boundaries and numerical artifacts must be smoothed to maintain excellent mesh quality for finite element analysis.

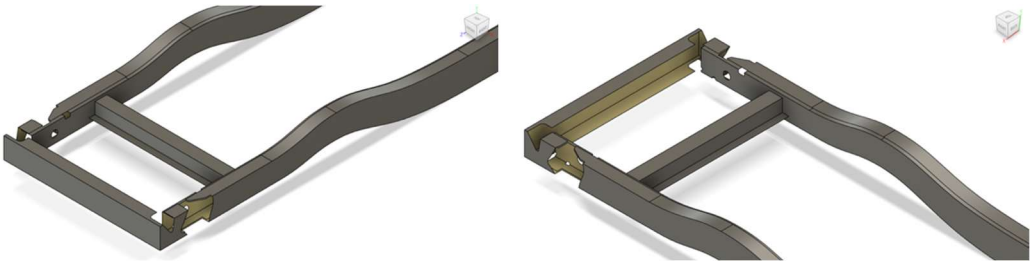


Figure 2.7. Idealized Optimized Traditional Chassis

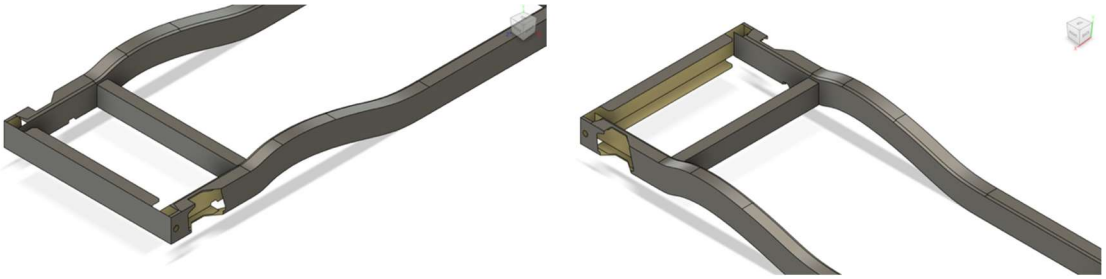


Figure 2.8. Idealized Optimized Modern Chassis

To resolve this, the raw faceted model from ANSYS was exported to Autodesk Fusion for geometric idealization and surface reconstruction. Figures 2.7 and 2.8 present the final 3D CAD versions of the optimized traditional and modern chassis, respectively. This clean-up process provides a clean geometry that can be properly meshed for the final explicit dynamic crashworthiness evaluation.

To assess the efficacy of these design improvements, each chassis design is analyzed in both its baseline and topology-optimized states, yielding four (4) distinct configurations. Frontal-impact simulations are performed on these configurations using ANSYS Explicit Dynamics, specifically modeled as a pendulum impact test where the chassis frame were on fixed supports. The test only modeled the main chassis frame itself, and no additional substructures that led to the passenger cabin or seats were included. To ensure robustness and reliability, the finite element analysis (FEA) parameters are validated against an experimental physical pendulum impact test conducted by Catalan (2017) at the Philippine Department of Science and Technology – Metals Industry Research and Development Center (DOST-MIRDC), that yielded accurate acceleration results on impact. Following this validation, the FEA pendulum setup is applied to each of the four chassis configurations at two (2) distinct impact energies–12.9 kJ and 29.4 kJ– adapted from the test of Catalan (2017) and the protocol of the United Nations Economic Commission for Europe (UNECE) Regulation No. 29, representing varying severity levels. This yields a total of eight (8) independent simulation runs.

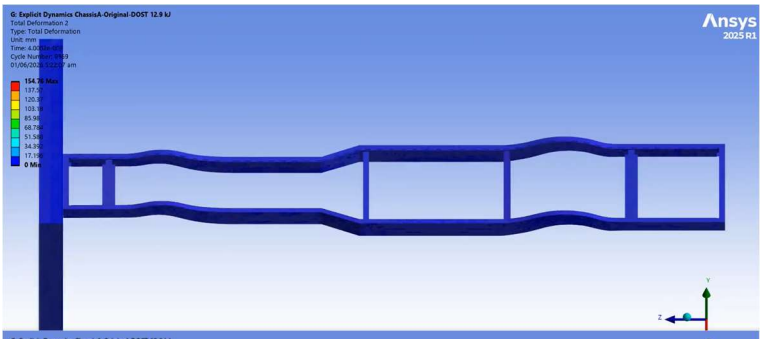
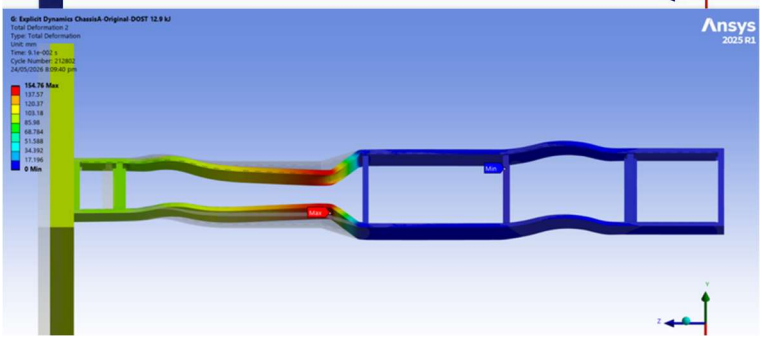
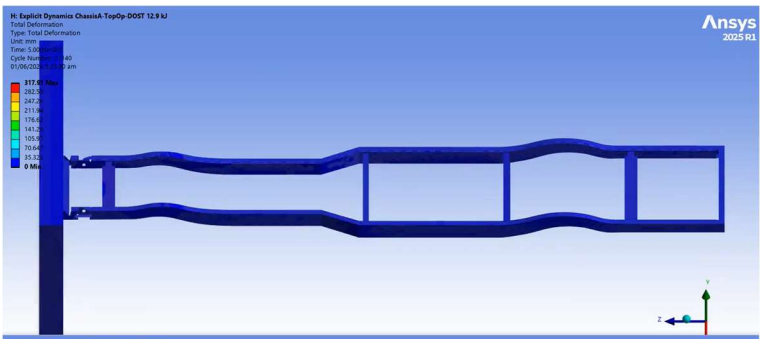
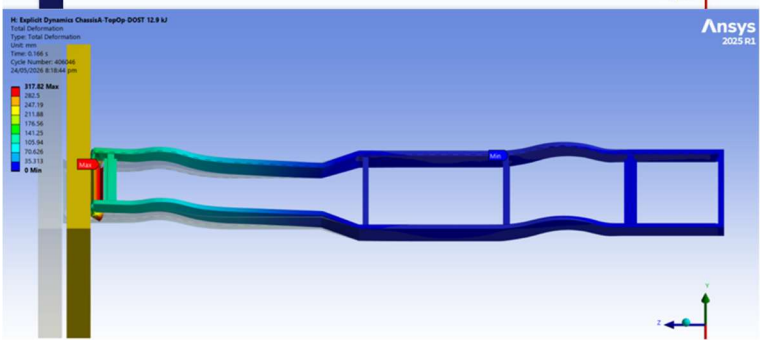
Finally, structural performance is evaluated using core crashworthiness metrics: energy absorption, peak impact force, and crumple distance. To further support the dynamic analyses, secondary parameters including total deformation, equivalent von-Mises stress, equivalent plastic strain, and crash pulse duration are also quantified. Ultimately, by comparing the baseline models against their optimized counterparts, the study aims to determine whether stress-based topology optimization significantly improves the passive safety capabilities of the jeepney chassis.

3. RESULTS AND DISCUSSION

3.1 Total Deformation

Frontal impact explicit dynamics were performed to evaluate chassis deformation behavior and isolate critical structural vulnerabilities under loading, with the optimization objective focused on confining deformation strictly to the front-end region.

Table 3.1 Total Deformation Contour Plots of Traditional Chassis

Chassis Design	Total Deformation Contour Plot
12.9 kJ Impact Energy **Baseline** t = 0.000 s	
t = 0.091 s	
Optimized t = 0.000 s	
t = 0.166 s	
29.4 kJ Impact Energy **Baseline** t = 0.000 s	

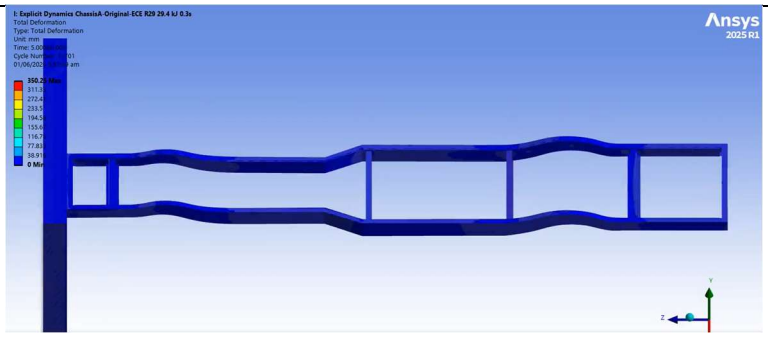
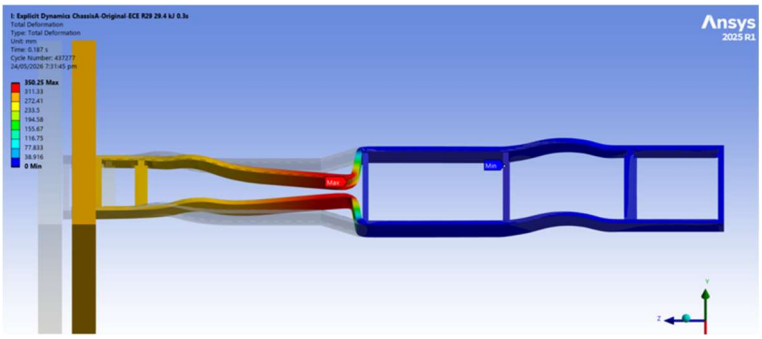
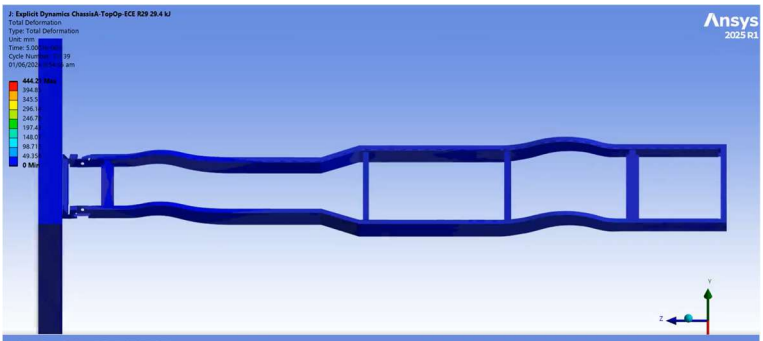
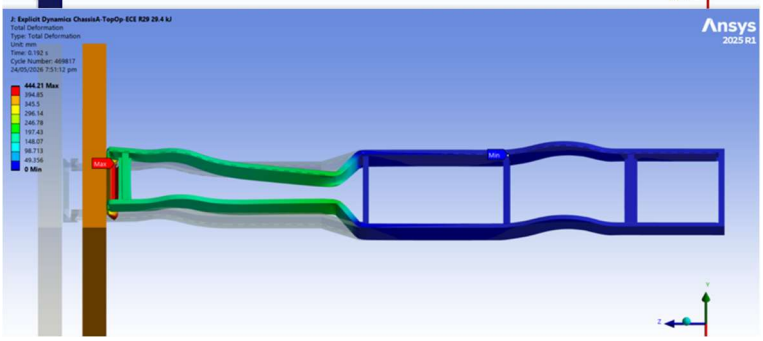
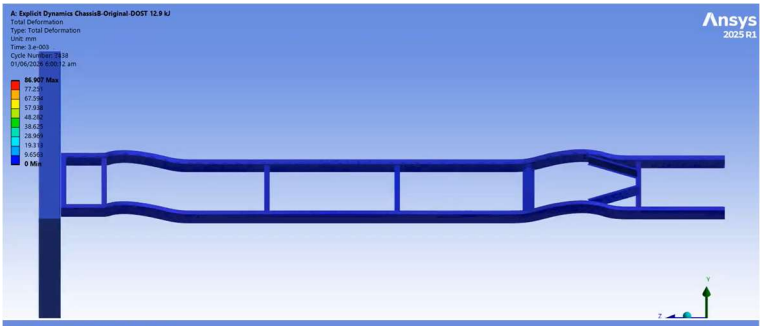
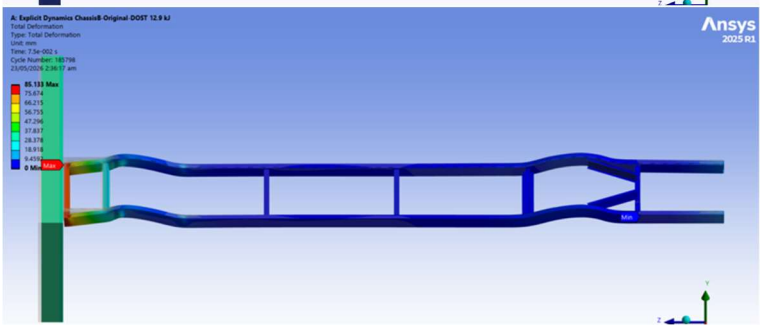
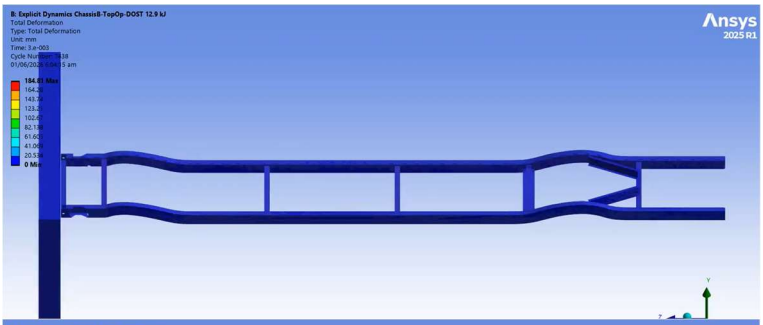
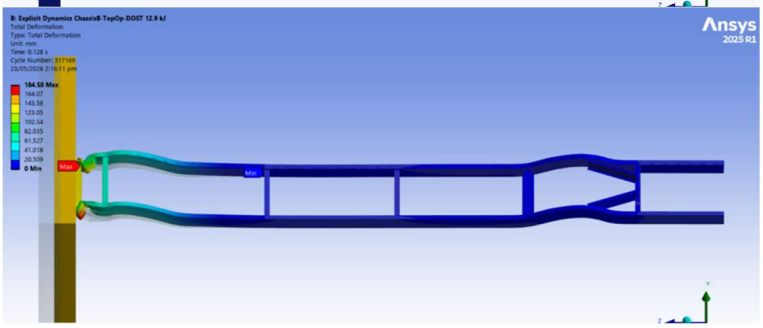
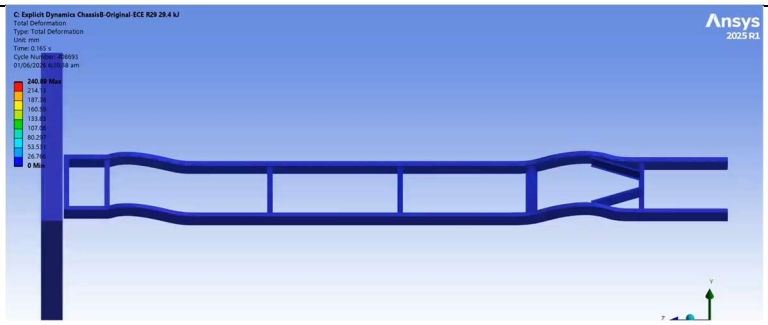
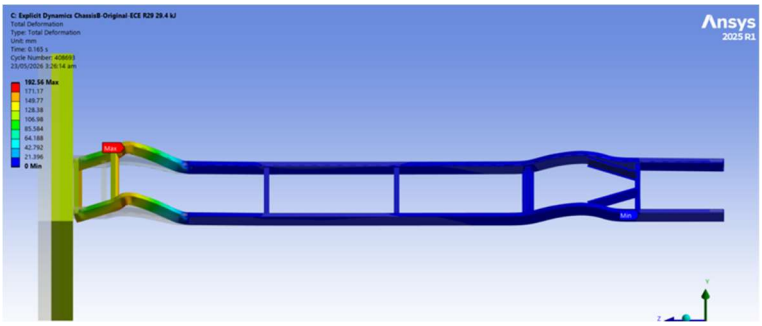
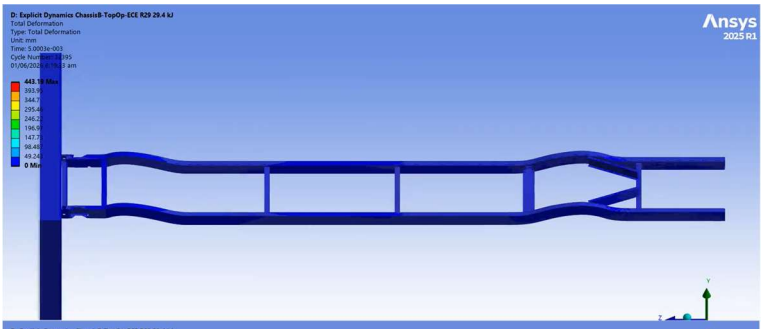
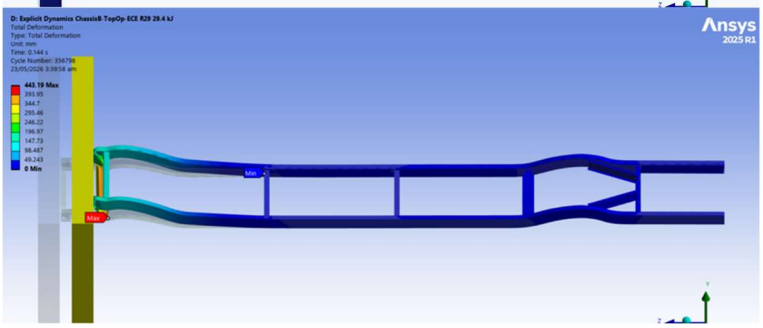
Chassis Design	Total Deformation Contour Plot
t = 0.187 s	
	
Optimized t = 0.000 s	
t = 0.192 s	

Table 3.1 shows deformation contour plots of the traditional chassis. The vertical bars on the left-hand side of the figures indicate the color scale representing the amount of total deformation the chassis undergoes throughout the dynamic event, measured in millimeters. Moreover, the particular time steps showing the deformed chassis were specifically selected to show the significant deformation the chassis has experienced. For the traditional configuration, the baseline chassis under 12.9 kJ and 29.4 kJ loads initially succumbed to severe buckling at the middle offset width portion. This confirms the historical vulnerability of traditional jeepneys, where the bent middle section acts as a critical stress concentration leading to passenger cabin

intrusion (Clavecillas, 2017). In contrast, the optimized traditional chassis eliminated middle-section deformation across both energy levels and successfully restricted the maximum deformation strictly to the front-end region.

Table 3.2 Total Deformation Contour Plots of Modern Chassis

Chassis Design	Total Deformation Contour Plot
12.9 kJ Impact Energy **Baseline** t = 0.000 s	
t = 0.075 s	
Optimized t = 0.128 s	
	
29.4 kJ Impact Energy	

Chassis Design	Total Deformation Contour Plot
Baseline t = 0.000 s	
t = 0.165 s	
Optimized t = 0.000 s	
t = 0.144 s	

A similar trend was observed for the modern chassis configuration shown in Table 3.2. Under both 12.9 kJ and 29.4 kJ impacts, the modern baseline chassis localized deformation at the curved geometry around the front wheel well beneath the engine compartment. While this modern layout eliminates the mid-chassis buckling found in traditional designs, the wheel-well yielding presents a severe intrusion risk to the driver's footwell. Upon applying topology optimization, the modern optimized chassis successfully shifted this yielding forward to the front-end. While secondary load paths remained near the wheel well, the optimized layout effectively prevented progressive buckling and preserved driver cabin integrity.

3.2 Equivalent von-Mises Stress and Plastic Strain

Table 3.3. Summary of Maximum Equivalent von-Mises Stress and their Location

Chassis Design	Overall Max Stress (MPa)	Location
Traditional (12.9 kJ)		
Baseline	600.77	Mid-section
Optimized	719.01	Front
Traditional (29.4 kJ)		
Baseline	813.46	Mid-section
Optimized	728.55	Front
Modern (12.9 kJ)		
Baseline	900.34	Mid-section
Optimized	925.23	Front
Modern (29.4 kJ)		
Baseline	982.32	Mid-section
Optimized	817.32	Front

Table 3.4. Summary of the Local Maximum Plastic Strain

Chassis Design	Max Plastic Strain (Front Section)	Max Plastic Strain (Middle Section)
Traditional (12.9 kJ)		
Baseline	0.082	0.326
Optimized	0.495	0.072
	+503.7%	*–77.9%*
Traditional (29.4 kJ)		
Baseline	0.119	0.564
Optimized	0.483	0.430
	+305.9%	*–23.8%*
Modern (12.9 kJ)		
Baseline	0.298	0.405
Optimized	0.420	0.084
	+40.9%	*–79.3%*
Modern (29.4 kJ)		
Baseline	0.538	0.482
Optimized	0.782	0.124
	+45.4%	*–74.3%*

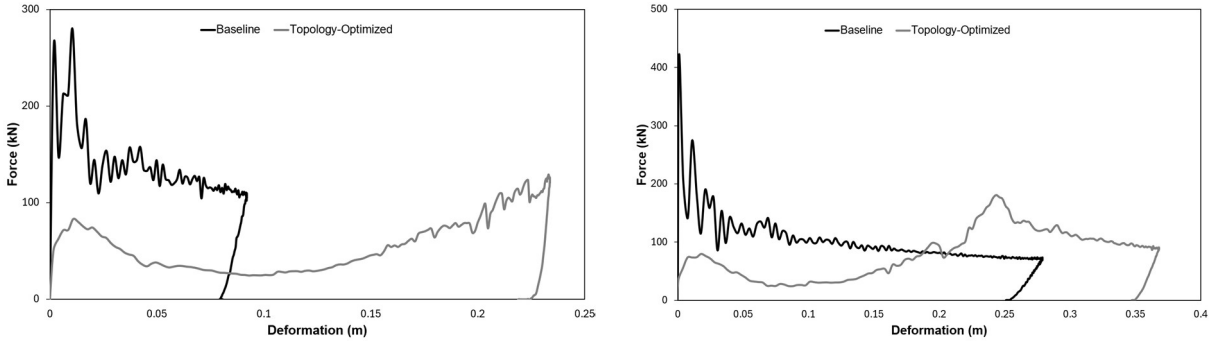
While deformation plots track macroscopic collapse, investigating internal structural integrity and permanent yielding requires evaluating equivalent von-Mises stress (Table 3.3) and local plastic strain (Table 3.4). For the traditional chassis configurations, the unoptimized baseline under both 12.9 kJ and 29.4 kJ loads experienced severe stress concentrations and plastic strain spikes localized at the middle width offset section, confirming heavy vulnerability beneath the

passenger cabin. Topology optimization successfully forced the absolute peak stress and primary yielding out of the cabin and into the front end. However, under the higher 29.4 kJ load, the optimized traditional chassis still retained significant residual mid-section stress and experienced a lower strain reduction (–23.8%). This indicates that even with optimization, the inherent geometry of traditional longitudinal width offsets acts as an unavoidable stress concentrator under severe loading.

Conversely, the straight longitudinal rail architecture of the modern chassis yielded highly superior optimization performance. In the modern baseline, the straight rails successfully shifted macro-stress distributions away from the mid-section toward the front wheel arch, though absolute peak stresses and maximum plastic strains were still recorded in the mid-section due to localized buckling. The optimized modern configurations completely resolved this issue across both energy levels, systematically forcing absolute peak stresses forward and dropping mid-section plastic strains by 79.3% (12.9 kJ) and 74.3% (29.4 kJ). By eliminating the legacy middle-bend geometry, the optimized modern front-end acts as a highly effective, controlled passive safety mechanism that prevents the transmission of critical impact energy to the driver footwell and passenger spaces.

3.3 Energy Absorption

The objective of structural crashworthiness is to dissipate the kinetic energy of a collision through controlled plastic deformation before that energy can reach the occupants. Quantitatively, the total Energy Absorption (EA) of a structure is mathematically defined as the integral of the impact force with respect to the deformation displacement; graphically, this is calculated as the total area under the Force versus Deformation curve. To evaluate the overall energy-absorbing capacity of the jeepney structures, Force vs. Deformation responses were extracted for all configurations.



(a) (b)

Figure 3.1. Force vs. Deformation Response of Traditional Baseline and Topology-Optimized Chassis under (a) 12.9 kJ and (b) 29.4 kJ Impact Energy

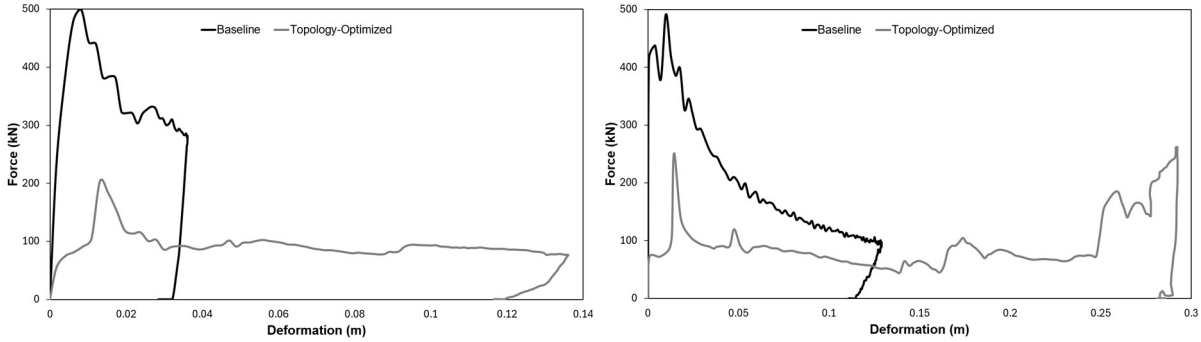


Figure 3.2. Force vs. Deformation Response of Modern Baseline and Topology-Optimized Chassis under (a) 12.9 kJ and (b) 29.4 kJ Impact Energy

As visually shown in Figures 3.1 and 3.2, the baseline graphs are dominated by an abrupt, massive spike in force with minimal longitudinal deformation. Because the rigid chassis refused to yield, it abruptly stopped the impactor without properly absorbing the kinetic energy into the metal. By sustaining a controlled force over a much longer deformation distance, the optimized chassis visibly maximizes the progressive area under the curve. Once the kinetic energy is fully transferred and maximum deformation is reached, an unloading phase occurs. The force steeply drops as the metal acts like a stiff spring, releasing its stored elastic energy. This elastic recovery causes the total deformation to physically decrease, forming the steep return curve. Ultimately, the point where this curve reaches the horizontal axis defines the actual, permanent plastic deformation sustained by the frame.

Table 3.5. Energy Absorption Summary

Chassis Design	Impact Energy (kJ)	Total Energy Absorbed (kJ)
Traditional		
Baseline	12.9	12.268
Optimized	12.9	12.444
		+1.43%
Traditional		
Baseline	29.4	28.252
Optimized	29.4	28.519
		+1.16%
Modern		
Baseline	12.9	12.116
Optimized	12.9	12.257
		+0.94%
Modern		
Baseline	29.4	26.183
Optimized	29.4	26.959
		+2.97%

To quantitatively verify this structural behavior, the total Energy Absorbed (EA) was calculated by integrating the area under each respective Force vs. Deformation curve. Table 3.5 summarizes the final energy absorption values for all models. As detailed in the table, the optimized structures yielded a slight but consistent mathematical increase in overall energy absorption, ranging from 0.94% to 2.97% across the various configurations.

3.4 Peak Impact Force

In evaluating crashworthiness design, it is imperative that structures that generate lower instantaneous forces are highly desirable. A massive peak force transferred to the vehicle could compromise the safety of the passengers. From the impulse-momentum theory, this can be achieved by lengthening the impulse duration. To evaluate this behavior, the maximum force responses for the traditional and modern chassis configurations were obtained from the simulation along with the collision duration.

Table 3.6 Peak Impact Force and Collision Duration Summary

Chassis Design	Peak Impact Force (kN)	Impact Duration (s)
Traditional (12.9 kJ)		
Baseline	280.255	0.086
Optimized	129.014	0.162
	-53.97%	*+88.37%*
Traditional (29.4 kJ)		
Baseline	418.953	0.185
Optimized	180.813	0.190
	-56.84%	*+2.70%*
Modern (12.9 kJ)		
Baseline	498.810	0.034
Optimized	203.609	0.124
	-59.18%	*+264.7%*
Modern (29.4 kJ)		
Baseline	491.852	0.106
Optimized	262.352	0.142
	-46.66%	*+33.96%*

As detailed in Table 3.6, baseline configurations exhibit severe, instantaneous force spikes, while topology-optimized models consistently lower peak forces by up to 59.18% and extend collision durations by up to 264.70%. This inverse relationship validates that the optimized front-end geometries crush progressively to maximize deceleration time, reducing the impact force transferred to the cabin.

The modern chassis under 12.9 kJ impact energy provides the most critical evidence of this structural improvement. The modern baseline behaves as an overly rigid structure, arresting the impactor in a mere 0.034 s and generating a massive peak force of nearly 499 kN—a load that would incur major injuries to occupants. In contrast, the optimized configuration deforms systematically, extending the crash duration to 0.124 s and slicing the peak force down to roughly 203 kN. This confirms that the optimized designs successfully convert violent, instantaneous structural shocks into manageable, survivable deceleration events.

3.5 Crumple Distance

The objective of a vehicular crumple zone is to dissipate energy through controlled physical collapse while strictly preventing structural intrusion into the passenger survival space (Li et al., 2017). In crashworthiness analysis, while we want to increase crumple distance, it is imperative to quantitatively verify that impact deformations do not compromise the engine and passenger space.

To evaluate this, the longitudinal crumple distance, measured as the total axial displacement of the chassis' front face during impact, was extracted for all configurations across both the 12.9 kJ and 29.4 kJ impact energy levels. For both traditional and modern chassis design, a critical maximum crumple limit of 362 mm and 380 mm was established respectively. Any axial deformation exceeding these thresholds indicates a critical structural intrusion into the engine region. Table 3.8 summarizes the final crumple distances for all evaluated models.

Table 3.8. Crumple Distance and Safety Limits Summary

Chassis Design	Crumple Distance (mm)	Safety Limits Exceeded?
Traditional (12.9 kJ)		
Baseline	92.06	No
Optimized	233.7	No
	+153.86%	
Traditional (29.4 kJ)		
Baseline	279.02	No
Optimized	368.11	Yes (>362mm)
	+31.93%	
Modern (12.9 kJ)		
Baseline	36.15	No
Optimized	136.11	No
	+276.55%	
Modern (29.4 kJ)		
Baseline	129.01	No
Optimized	292.04	No
	+126.37%	

As detailed in the table, all optimized configurations exhibited a significantly increased longitudinal crumple distance compared to their respective baselines. The optimized structures were able to reach up to 276.55% increase in crumpling distance under 12.9 kJ energy and up to 126.37% increase under 29.4 kJ. From a crashworthiness perspective, this increase is a highly successful and intended outcome. The fundamental goal of structural crashworthiness is to lengthen crash through controlled, progressive vehicle deformations (crumple zones) while simultaneously preventing those deformations from intruding into the passenger survival space. By deliberately introducing deformable structures via topology optimization, the front-end portion were successfully allowed to crush and lengthen the collision duration. For both the traditional and modern chassis under the 12.9 kJ baseline impact, as well as the modern chassis under the 29.4 kJ impact, this controlled crumple was highly effective and safely remained below the set critical intrusion threshold.

However, under the severe 29.4 kJ impact, the traditional optimized chassis failed the 362 mm safety criterion. The >362 mm measured displacement at the front face is a compounded

geometric response, a combination of the localized front-end crushing plus the global structural shifting caused by the offset width profile at the middle of chassis. Because of this mid-chassis yielding, the entire front half of the vehicle was forced to shift and fold backward, ultimately pushing the front face past the intrusion limit.

4. CONCLUSION

The primary objective of this study was to enhance the passive safety of traditional Philippine jeepneys by addressing their inherently rigid, non-energy-absorbing chassis structures. Through explicit dynamics simulations evaluated at baseline (12.9 kJ) and severe commercial vehicle standards (29.4 kJ UNECE R29), the study demonstrated that applying stress-based topology optimization successfully transforms the front chassis into a progressive, energy-absorbing crumple zone. The optimized structures significantly prolonged collision durations by up to 264.70%, reducing lethal peak impact forces by up to 59.18% while maximizing total energy absorption. Furthermore, the optimization effectively redistributed dangerous von-Mises stresses and severe plastic strains away from the passenger cabin, safely trapping permanent material yielding at the extreme front end of the vehicle.

Despite these significant crashworthiness improvements, a critical limitation was observed under extreme kinetic energy, where the topology-optimized traditional chassis failed the 362 mm safety criterion by breaching the passenger intrusion limit. This structural failure highlights that while computational algorithms can efficiently generate localized crumple zones, they cannot overcome fundamentally flawed global geometries like the mid-chassis bend of the traditional jeepney, which acts as a geometric weak point. Consequently, this study mathematically justifies the necessity for local manufacturers to transition away from historical bended chassis designs and adopt modernized, straight-rail geometries to eliminate catastrophic structural shifting and successfully align with the safety objectives of the Public Utility Vehicle Modernization Program (PUVMP).

To fully realize these safety benefits within the local transport sector, future research must expand upon this foundational structural analysis. Key recommendations include conducting high-speed, full-vehicle crash simulations with anthropomorphic test devices to evaluate biomechanical occupant loads, as well as physically prototyping the optimized front-end for destructive pendulum testing at DOST-MIRDC to validate the computational models. Additionally, future studies should investigate multi-directional impact scenarios like side collisions, explore multi-material designs for further weight reduction, and develop modular, replaceable bolted crash boxes tailored for jeepney architectures to ensure both passenger safety and post-collision economic viability.

5. RECOMMENDATIONS

This study isolated the dynamic analysis to the bare chassis to evaluate foundational structural behavior. Future research should integrate the complete vehicle body, powertrain mass, and passenger payload into the finite element model. Conducting high-speed full-vehicle simulations, such as the ASEAN NCAP 64 km/h full-frontal or 40% offset tests, using anthropomorphic test devices (crash dummies) will provide a more comprehensive evaluation of the Head Impact Criterion (HIC) and the exact biomechanical loads experienced by the occupants.

Additionally, while the computational models in this study were rooted in verified physical pendulum baselines, the newly optimized chassis geometries must be physically tested for more accurate validation. A scaled or full-size prototype of the topology-optimized front-end may be fabricated and tested to practically correlate and validate the simulated force-displacement curves.

Lastly, the study acknowledges that frontal collisions are only one aspect of road accidents. To ensure holistic passenger safety, future studies should subject the optimized jeepney framework to side-impact analyses and dynamic roll-over tests. Modifying the structural pillars and side-impact beams using topology optimization could further mitigate injuries in complex, multi-vehicle crash scenarios.

REFERENCES

ACEA (European Automobile Manufacturers Association). (2019). *Road safety: safe vehicles, safe drivers, safe roads*. https://roadsafetyfacts.eu/themes/ACEA-Road-Safety-Facts/img/ACEA_Road_Safety.pdf

ANSYS, Inc. (n.d.). Topology optimization of a bell crank. ANSYS Innovation Space. Retrieved May 27, 2026, from https://innovationspace.ansys.com/courses/courses/topology-optimization-using-ansys-mechanical/lessons/topology-optimization-of-a-bell-crank-lesson-1/

Bacero, A. (2009). The state of jeepney assembly in the Philippines. NCTS, UP Diliman. Available at: https://ncts.upd.edu.ph/tssp/wp-content/uploads/2018/08/Bacero09.pdf

Catalan, F. A. (2017). Simulation and experimental jeepney chassis crash impact analysis [Master's thesis, Technological University of the Philippines].

Clavecillas, R. C. (2017). Crash simulation of a customized local road vehicle (CLRV) – Jeepney chassis front impact structure using LS-DYNA non-linear solutions [Master's thesis, Technological University of the Philippines].

De Leon, D. M., Alexandersen, J., Fonseca, J. S. O., Sigmund, O. (2015). Stress-constrained topology optimization for compliant mechanism design. *Structural and Multidisciplinary Optimization*, 52, 929–943. https://doi.org/10.1007/s00158-015-1279-z

Dimalanta, R., Atienza, J. M., Samonte, E. (2023). Putting transport workers and commuters first: The route to just transition in public transport modernization. UP CIDS Policy Brief 2023-01. University of the Philippines Center for Integrative and Development Studies.

Du Bois, P., Chou, C. C., Fileta, B. B., Khalil, T. B., King, A. I., Mahmood, H. F., Mertz, H., Wismans, J. (2004). *Vehicle Crashworthiness and Occupant Protection*. American Iron and Steel Institute, Detroit, MI.

Duddeck, F., Volz, K. (2012). A new topology optimization approach for crashworthiness of passenger vehicles based on physically defined equivalent static loads. In Proceedings of the International Crashworthiness Conference, Milan, Italy.

Emmendoerfer, H., Jr., Fancello, E. A., Silva, E. C. N. (2020). Stress-constrained level set topology optimization for compliant mechanisms. *Computer Methods in Applied Mechanics and Engineering*, 362, 112777. https://doi.org/10.1016/j.cma.2020.112777

Estipular, J. L. (2020). *Looking into the Implementation of Public Utility Vehicle Modernization Program*. Congressional Policy and Budget Research Department. http://serp-p.pids.gov.ph/publication/public/view?slug=looking-into-the-

implementation-of-public-utility-vehicle-modernization-program. Accessed 16 Jun 2026.

Fancello, E. A. (2006). Topology optimization for minimum mass design considering local failure constraints and contact boundary conditions. *Structural and Multidisciplinary Optimization*, 32, 229–240.

Li, Z., Yu, Q., Zhao, X., Yu, M., Shi, P., Yan, C. (2017). Crashworthiness and lightweight optimization to applied multiple materials and foam-filled front end structure of auto-body. *Advances in Mechanical Engineering*, 9(8), Article 168781401770280. https://doi.org/10.1177/1687814017702806

Lopes, R. F. F., Pinto, S. M., Parento, M. P. L., Moreira, P. M. G. P. (2024). Crashworthiness optimisation and environmental impact assessment of a redesigned passenger coach integrating lean design-for-X framework. *Journal of Engineering Design*, 35, 390–429. https://doi.org/10.1080/09544828.2024.2318992

Manvi, P., Nizami, F., Anandtheertha, M. R., Srikanth, H. V. (2024). Stress based topology optimization of axial tubes under impact to improve energy absorption. *Journal of Polymer & Composites*, 12(Special Issue 4), S210–S220. https://doi.org/10.37591/JoPC

Metropolitan Manila Development Authority (MMDA). (2019). MMARAS annual report 2019. TEC-Road Safety Unit.

Rogelio, J. P., Clavecillas, R. C., Maglantay, R. S., Bandala, A., Munsayac, F. E. T., Baldovino, R. G., Liza, F. P., Puerto, J. Q. (2018). Design of the Philippine jeepney for crashworthiness analysis: A finite element analysis approach. In 2018 IEEE 10th International Conference on Humanoid, Nanotechnology, Information Technology, Communication and Control, Environment and Management (HNICEM). IEEE.

Rong, Y., Zhao, Z.-L., Feng, X.-Q., Xie, Y. M. (2022). Structural topology optimization with an adaptive design domain. *Computer Methods in Applied Mechanics and Engineering*, 389, Article 114382. https://doi.org/10.1016/j.cma.2021.114382

Silva, C. J. G., Lopes, R. F. F., Domingues, T. M. R. M., Parente, M. P. L., Moreira, P. M. G. P. (2024). Crashworthiness topology optimisation of a crash box to improve passive safety during a frontal impact. *Structural and Multidisciplinary Optimization*, 68(3). https://doi.org/10.1007/s00158-024-03924-6

Tang, T., Wang, L., Zhu, M., Zhang, H., Dong, J., Yue, W., Xia, H. (2024). Topology optimization: A review for structural designs under statics problems. *Materials*, 17(23), 5970. https://doi.org/10.3390/ma17235970

United Nations Economic Commission for Europe. (2012). Regulation No. 29 (Revision 3): Uniform provisions concerning the approval of vehicles with regard to the protection of the occupants of the cab of a commercial vehicle. https://unece.org/fileadmin/DAM/trans/main/wp29/wp29regs/R029r2e.pdf