

Article

Topology Optimization-Based Design Roadmap and Fatigue Life Evaluation of a 4 × 4 Independent Suspension Special-Purpose Electric Scooter Stub Axle

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Abstract

This study presents a topology optimization-based design methodology for a fail-safe stub axle of a lightweight 4 × 4 electric scooter with independent suspension, with the objective of developing a structural design roadmap. Topology optimization was performed under five critical load conditions: vertical, longitudinal, and lateral impacts, as well as braking and cornering under braking, representing standard driving scenarios defined in the literature. The final geometry was built by combining the topology optimization results from each load case, and it was evaluated using finite-element analysis, showing that it is safe under all critical loading conditions with respect to yield strength. The fatigue life assessment was performed using the Goodman–Haigh approach, based on load-condition pairs recommended in the literature, and it was found that the stresses in critical regions remain within the infinite fatigue life region. In addition, based on literature data, the proposed lightweight design approach indicates potential benefits in terms of both energy consumption and manufacturing cost. Overall, the findings suggest that the presented methodology can serve as a fail-safe design roadmap for the development of electric vehicle components.

Keywords: chassis system engineering; electromobility; Goodman–Haigh method; fail-safe design; small electric vehicle (SEV); structural optimization; topology optimization; product design

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1. Introduction

The severe environmental impacts of global warming have accelerated the transition from fossil-fuel vehicles to electric vehicles. Within this transition, it is generally assumed that the electrical energy utilized is sourced from renewable and non-fossil-based energy systems. Studies reported in the literature indicate that, over time, electric vehicles are expected to achieve a 40–50% reduction in energy consumption and a 45–55% decrease in production costs [1].

In parallel with population growth, the increasing demand for transportation further exacerbates critical issues such as environmental pollution, insufficient parking capacity, and the saturation of existing transportation infrastructure [2,3]. One of the proposed

approaches to mitigate these challenges is the adoption of smaller and lighter electric vehicles, along with their integration into multimodal transportation systems such as public transit networks [3]. One of the primary objectives of the ten-year transportation roadmap established by the European Commission is to ensure clean and sustainable urban mobility. In line with this objective, the promotion of smaller, lighter, and purpose-specific passenger vehicles is strongly emphasized [4].

The environmental impact of the transportation sector is significant on a global scale; for instance, more than 30% of total greenhouse gas emissions in the United States originate from this sector. To this end, vehicle lightweighting has emerged as an effective approach to reducing energy consumption. According to the open literature, for a vehicle with an average mass of 1500 kg, each 50 kg reduction in weight can lead to a decrease of approximately 4–5 g in CO₂ emissions [1]. In addition to lightweighting strategies achieved through material substitution, reducing vehicle size and promoting vehicle-sharing concepts are also among the key approaches contributing to emission reduction. When these strategies are implemented collectively, it is estimated that the total life-cycle emissions of a single vehicle can be reduced by up to 57% [1].

This category of vehicles can be best exemplified by those referred to as small electric vehicles (SEVs). In the literature, various definitions and classifications exist for SEVs, and these designations may vary depending on regional regulations and applicable legislation. In general, vehicles are classified based on their intended use, number of wheels, and design characteristics. Within the European Union, vehicle classification is carried out in accordance with the standards established by the United Nations Economic Commission for Europe (UNECE), and the corresponding classification is presented in Table 1 [5].

Table 1. Motor vehicle classes and their main characteristics [According to 5].

Category	Subclass	Description (Simplified)
L	L1–L7	Two-, three-, and light four-wheeled vehicles (≤ 350 – 550 kg, ≤ 15 kW depending on class)
M	M1–M3	Passenger vehicles (up to >8 seats, $\leq$ or >5 tons)
N	N1–N3	Goods transport vehicles (≤ 3.5 t to >12 t)
O	O1–O4	Trailers (≤ 0.75 t to >10 t)
T	-	Agricultural/forestry tractors (tractive vehicles)
R	-	Trailers for load transport/processing (ratio ≥ 3.0)
S	-	Specialized agricultural/forestry trailers (ratio < 3.0)
G	-	Off-road vehicles

In Europe, it is estimated that approximately 35 million users regularly prefer SEVs [2]. The fact that these vehicles are shorter than conventional vehicles offers a significant advantage in terms of superior maneuverability, particularly in congested traffic conditions. SEVs constitute a subset of both electric vehicles and micro-mobility solutions. Generally designed as single- or two-seat vehicles, they exhibit lower mass compared to conventional electric vehicles due to the use of smaller battery systems and predominantly composite body structures [6]. SEVs are typically classified as light motor vehicles with four or fewer wheels within the L-category. In contrast, M1 category vehicles are defined as passenger cars with at least four wheels and primarily intended for passenger transport. In Europe, L-category vehicles (L2e, L5e, L6e, L7e) are regulated under Regulation (EU) No 168/2013, and under certain conditions, four-wheeled micro and small electric vehicles may also be classified within the M1 category [7–9].

SEVs require less space both during parking and operation. For example, at a speed of 30 km/h, it has been reported that the Renault Twizy occupies approximately 20 m² less area compared to the Mercedes-Benz B-Class model [3]. This situation may contribute to

considerable space savings in urban transport, thereby reducing travel times by up to 48% [2]. In addition, it improves land-use efficiency, as a single conventional parking space can accommodate approximately five electric vehicles. From an environmental perspective, it is reported that electric vehicles produce approximately 50% less CO₂ emissions than conventional automobiles for the same trip [10]. Cost is a key determinant in transportation choices, and SEVs generally offer lower purchase and operating costs compared to conventional electric vehicles [3]. However, the current adoption rate of SEVs has not yet reached a level sufficient to deliver a significant environmental improvement [2]. For this reason, the importance of improvement studies aimed at enhancing the performance of SEVs and expanding their application areas is growing. The SEV-class electric scooter shown in Figure 1 incorporates fundamental vehicle subsystems such as steering, suspension, and braking. However, the presence of critical electrical components such as the electric motor and battery package further emphasizes the importance of structural design in terms of safety.

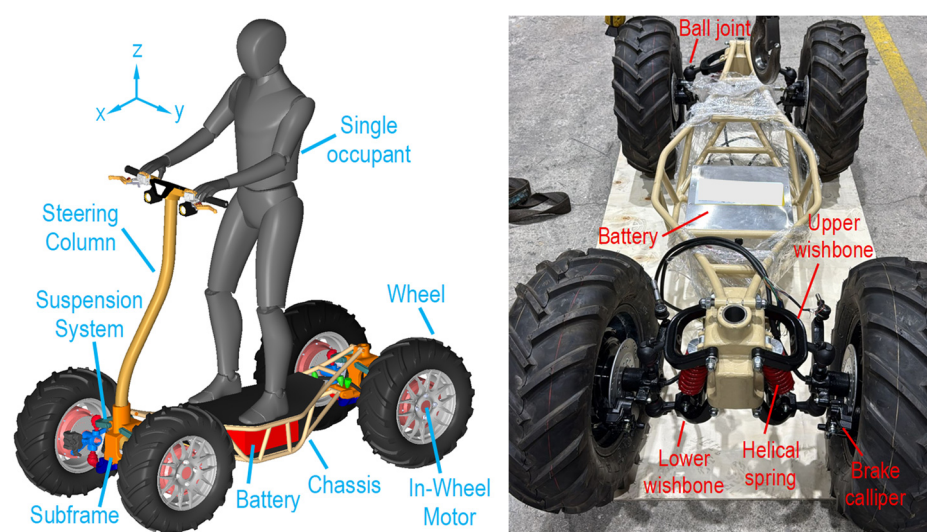


Figure 1. Schematic representation of the SEV-class electric scooter.

Vehicle components must be designed to safely and efficiently transmit the torque generated by the electric motor. Increasing acceleration values, higher maximum speeds, and improved braking performance have significantly increased the demand for more durable and safer chassis systems. Consequently, the use of independent suspension systems has become more widespread. In particular, the double-wishbone suspension system has emerged as an effective solution, enabling precise control of wheel position via upper and lower wishbones. In the system shown in Figure 2a, increasing the effective distance (c) between the upper and lower wishbones reduces the forces acting on the suspension arms and their connection points (A, B, C and D). Consequently, component deformations and bushing loads decrease, resulting in more precise wheel control. In this way, such suspension systems offer a wide range of design flexibility [11]. The stub axle, an integral part of this structure, is a critical component that operates in conjunction with the steering and suspension systems and is subjected to time-varying loads throughout its service life. The stub axle serves as the primary mounting interface between the wheel assembly and the suspension system, and subsequently the chassis. It also provides structural support for both the steering and braking systems. The stub axle is subjected to a wide range of loading conditions [8]. The stub axle is a fundamental load-bearing component that transmits the forces and moments generated between the chassis and the road surface. Furthermore, it plays a critical role in the positioning and steering of the wheel assembly, as well as in the integration of the drive and braking systems [7].

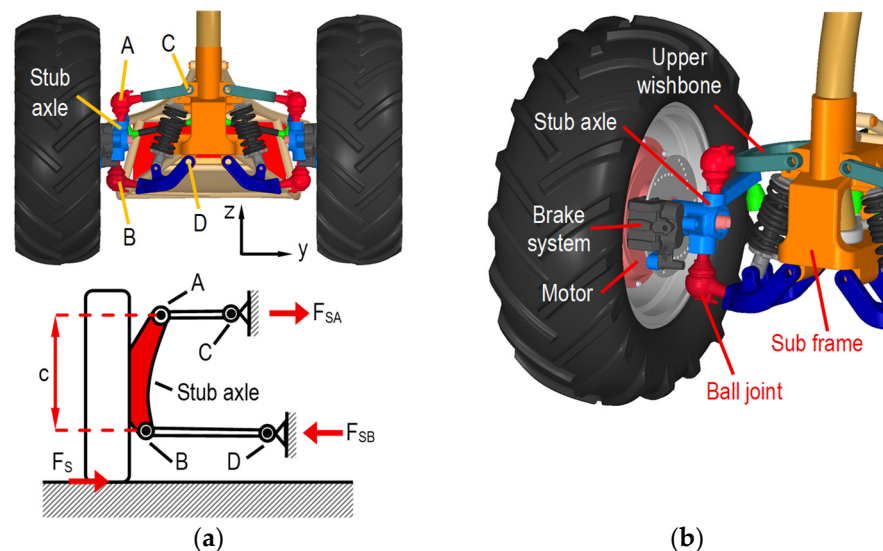


Figure 2. (a) Schematic representation of the double-wishbone suspension system (The schematic adapted from [11]); (b) Detailed view of the stub axle assembly, including the upper wishbone, ball joint, brake system, motor, and subframe connections.

As shown in Figure 2b, particularly in scooters, the stub axle serves as a platform to which sub-components such as the engine drive shaft, tie rod, and brake caliper are mounted. Therefore, in stub axle design, the precise definition of load transfer scenarios, the real-life evaluation of load conditions, and ensuring structural compatibility with all subsystems are of critical importance [7].

The stub axle operates under constantly varying and dynamic loads during normal service conditions and is at risk of plastic deformation or cracking due to high stresses in the event of overloading or incorrect use [7]. For this reason, fatigue behavior is considered a critical design criterion in stub axle design [12]. The literature indicates that spheroidal graphite (SG) cast iron is the most used material in the manufacture of stub axles, particularly for light-duty vehicle applications [12]. However, depending on the vehicle classification and design requirements, alternative materials such as cast iron, forged steel, carbon steel, aluminum alloys, and composite structures may also be preferred [7,13]. However, the limited fatigue strength of aluminum alloys brings with it certain design constraints [14].

There are several studies in the literature that examine different material choices and optimization approaches with the aim of reducing the weight of the stub axle and suspension system components while simultaneously enhancing their mechanical behavior. In a study in the literature, a suspension component for a Formula Society of Automotive Engineers (FSAE) vehicle was redesigned using a metal matrix composite (MMC) material, ensuring that the safety factor (FOS) was greater than 2. Subsequent analytical calculations and finite-element analysis (FEA) results were evaluated together, and it was reported that a mass reduction of 69.28% was achieved while meeting the design constraints [15]. Similarly, a topology optimization study was carried out for a suspension component used in a different FSAE vehicle, based on Al 6061-T6 and Ti-6Al-4V materials, and the results obtained were evaluated comparatively. According to the analysis results, a safety factor of 2.6237 was obtained for the Ti-6Al-4V material, while this value was determined to be 1.554 for the Al 6061-T6 material. It was reported that the optimized design achieved a 68.737% improvement while meeting the design criteria [16]. In a study comparing cast iron, low-carbon steel, and EN 47 forged steel, it was reported that, as a result of the improvements achieved, EN 47 forged steel provided a 5.64% reduction in stub axle mass compared with low-carbon steel [17]. In another study, it was reported that a stub axle

designed using 6061 aluminum alloy provided approximately 30% higher stiffness compared to FCD600 cast iron [18]. In addition, in a study where structural steel was taken as the reference material, 4130 steel and AISI 1020 steel were examined as alternative materials; a comparison of the analysis results revealed that 4130 steel delivered the highest performance under static loading conditions. It was stated that a mass reduction of 42.70% was achieved through the optimization process applied in the same study [19].

The literature also includes studies focusing on the durability and fatigue performance of the stub axle. The fatigue behavior of a stub axle manufactured from C45 steel was investigated using FEA under specific loading conditions. A comparison of constant-amplitude loading (CAL) and variable-amplitude loading (VAL) conditions revealed that safety criteria were met within appropriate stress ranges [20]. In another study conducted on a cast iron stub axle, driving scenarios involving acceleration, braking, cornering, and different road conditions were considered. It was reported that the fatigue life predictions made using the CAL and VAL methods contained an error of approximately 21% [13].

In a few studies, fatigue life has been assessed not only using FEA but also by utilizing data obtained from test conditions. In some studies, fatigue life has been assessed not only using FEA but also by utilizing data obtained from road test conditions. In this context, fatigue life has been estimated based on the S-N approach, using experimental tests conducted with road load data obtained from the road tests [9]. For example, in the axle design of a vehicle developed for the BAJA SAE India competition, it was found that the initial design failed under sudden loading. Considering this finding, the design process was revisited, optimization studies were carried out, and the final design was validated under real-world terrain conditions [21]. However, there are also studies in the literature that are based on standard load conditions [22].

An examination of existing literature shows that research into stub axle design for electric scooters is limited. To the best of the authors' knowledge, no study in open literature has been found that combines topology optimization with fatigue life assessment for the stub axle component used in electric scooters. The aim of this study is to design a cast iron stub axle for a prototype electric scooter using a topology optimization method and to evaluate the fatigue life performance of the final design. The term 'fail-safe', as used in this study, refers to a design approach whereby the component is designed to prevent sudden failure under critical loading conditions and to ensure that the stresses arising under operating conditions remain within the safe life zone.

2. Materials and Methods

2.1. Materials

When selecting stub axle design materials, density was not the only criterion considered; mechanical performance, cost, and operating conditions were also evaluated together. Although aluminum alloys are widely preferred due to their lightweight advantage, it is known that the environmental impacts arising from production processes are greater compared to metals with higher density, such as iron and steel. Furthermore, they were not selected in this study due to their lower mechanical properties and higher cost compared to iron-based materials [1]. However, spheroidal graphite cast iron (GGG series) can be used in thin-walled components thanks to appropriate alloying and production methods. This enables them to achieve a competitive weight while retaining their exceptional mechanical properties.

The lightweight electric 4 × 4 scooter platform prototype is intended for use in various off-road conditions, primarily for military applications. Due to these operating conditions, the stub axle component is subjected to variable-amplitude and impact-induced loads. Consequently, the component's high fatigue strength has been identified as a critical

design requirement. An examination of the literature reveals that the fatigue strength of the Al 7075-T6 alloy is lower than that of GG60 spheroidal graphite cast iron. For this reason, spheroidal graphite cast iron has been selected as the material for the axle component in this study. High-strength aluminum alloys such as 7075-T6 do not exhibit a distinct fatigue limit, unlike iron-based materials [23–27]. Consequently, fatigue damage may continue to accumulate even at low stress levels under sufficiently high cycle counts, which can reduce structural reliability under high-cycle and variable-amplitude loading conditions. This limitation is considered a critical design concern, particularly for components subjected to fluctuating and impact-dominated service loads [14].

For these reasons, EN-GJS-600-3 material, belonging to the spheroidal graphite cast iron class, was selected for the manufacture of the stub axle (Tibet Makine Sanayi ve Ticaret A.Ş., Izmir) in this study. The mechanical properties of the material are presented in Table 2 [28–30]. Here, “R”, “R_p”, “HB”, and “A” denote the tensile strength, yield strength, Brinell hardness, and elongation values, respectively.

Table 2. The mechanical properties of EN-GJS-600-3 (According to [27–29]).

The Mechanical Properties			
R (MPa)	R _p (MPa)	HB	A
458 ± 15	363 ± 8	198 ± 4	2.1 ± 0.5%

2.2. Methods

2.2.1. Topology Optimization

Topology optimization is one of the most widely used approaches among structural optimization methods. This method aims to determine the optimal material distribution within a given design domain. In this context, the structural topology is represented through distribution functions defined over a fixed design space, enabling the achievement of an optimal material layout in accordance with the targeted performance criteria [31]. The fundamental objective is to identify the most efficient load transfer path under given loading and boundary conditions [32].

Among commonly used approaches in topology optimization, the SIMP (Solid Isotropic Material with Penalization) method enables the determination of an optimal material distribution under various loading conditions, boundary constraints, and manufacturing restrictions. In this method, the material density (ρ) is defined for each finite element within the design domain, ranging from 0 (no material) to 1 (full material). The relationship between material density and material properties in the SIMP approach is expressed by Equations (1) and (2) [33], where p denotes the penalization factor, ρ the density, and E the elastic modulus.

$$E_{ijkl}(x) = \rho(x)^p E_{ijkl}^0 \quad (1)$$

$$\int_{\Omega} \rho(x) d\Omega \leq V ; 0 \leq \rho(x) \leq 1, x \in \Omega \quad (2)$$

When the penalization factor is set to $p = 1$, stiffness varies linearly with density. However, selecting $p > 1$ suppresses intermediate density values, driving the solution toward a discrete 0–1 distribution. In the literature, values of $p \geq 3$ are generally reported to yield appropriate results. For the penalization to be effective, additional constraints such as a volume fraction constraint must be defined. The SIMP method is initiated with a uniform initial density distribution. During the iterative process, equilibrium equations are solved, sensitivity analyses are performed, and density values are updated using filtering techniques. These steps are repeated until convergence is achieved, resulting in the optimal material distribution [31–34].

2.2.2. Goodman–Haigh Approach

Accurate evaluation of the fatigue life of mechanical components is a critical design requirement, particularly for structures subjected to variable loading conditions. Among the various fatigue prediction methods developed for this purpose, the Goodman–Haigh approach defines the fatigue-safe region of a design by considering the relationship between mean stress and stress amplitude. In the Goodman–Haigh diagram, an example of which is presented in Figure 3, the stress amplitude (σ_a) is represented on the vertical axis, while the mean stress (σ_m) is shown on the horizontal axis. In the first stage, the yield strength is defined on both axes, and the yield envelope is obtained by connecting these points. Negative mean stress values correspond to compressive loading conditions, whereas positive mean stress values represent tensile loading conditions.

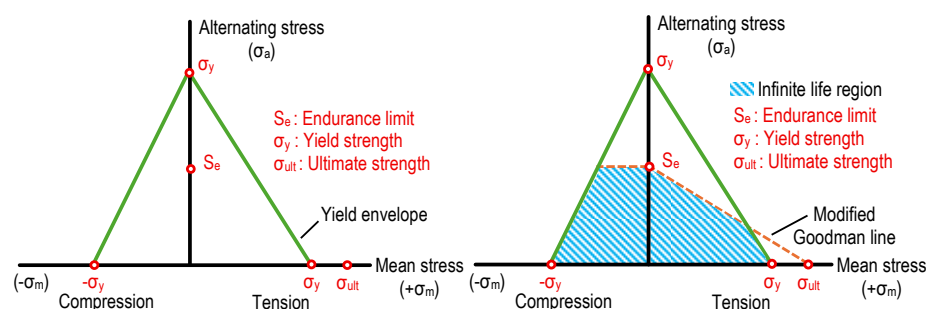


Figure 3. Infinite life region defined by the Modified Goodman line (According to [35]).

Subsequently, the endurance limit of the material, represented by the S_e value, is defined on the stress amplitude axis. This S_e value is then connected to the ultimate tensile strength (σ_{ult}) in the tensile region to construct the modified Goodman line. In the compressive region, the diagram is bounded by a linear curve originating from the S_e value. The region enclosed between the yield envelope and the modified Goodman line is considered the infinite life region [35–37]. Within the scope of this study, the Goodman–Haigh diagram was employed using loading scenarios reported in the literature for fatigue assessment of axle components. The method was used to evaluate whether the cyclic stress states obtained in the final stub axle design remained within the infinite life region.

3. Load Cases

In FEA studies conducted to investigate stress concentrations at critical areas of various vehicle components or to carry out optimization analyses, multiple load cases are generally considered. During the early stages of suspension design, wheel loads are often not directly available. Therefore, instead of real operating loads, standard load cases based on representative driving maneuvers and accepted as quasi-static are widely employed [7]. Similar standard load cases are commonly used by vehicle designers and have also been extensively adopted in the literature [7,18,20–22,38,39].

Wheel loads associated with these driving maneuvers can be defined in terms of acceleration values or as force and moment components. Since acceleration-based definitions are independent of axle load, they are considered more general parameters and can be reused even when the total vehicle mass changes. In the literature, standard load cases (1–10) and load case pairs used for fatigue and durability analyses, namely (11–12), (13–14), and (15–16), have been presented for different evaluation purposes [7].

Within the scope of this study, topology optimization was performed under loading conditions considered critical for the stub axle design, namely vertical impact (3 g) (Case 1), longitudinal impact (2.5 g) (Case 2), lateral impact (2.5 g) (Case 3), braking in a corner (Case 4), and braking (0.75 g) (Case 5). Different geometric configurations of the stub axle

were evaluated under these load cases. The coefficients corresponding to the selected load cases are presented in Figure 4.

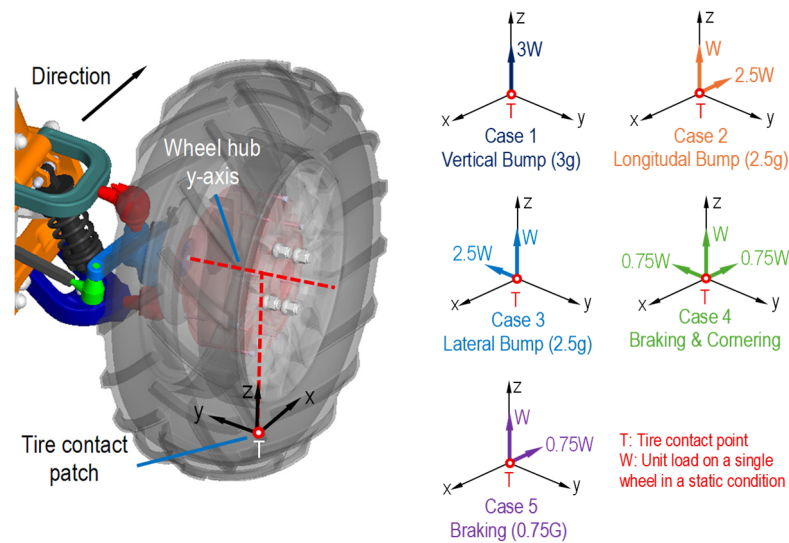


Figure 4. Schematic representation of the load distribution and coefficients for driving conditions (according to [7]).

The forces acting at the tire contact patch were determined assuming that the vehicle's total load is distributed equally between the front and rear axles, and that the front axle load is distributed symmetrically between the two wheels. The force distribution patterns at the tire–road contact patch for the selected load conditions are shown in Figure 5. The force components associated with load conditions are expressed as multiples of the gravitational acceleration (g) of the static load per wheel [7]. The total mass of the vehicle includes the mass of the driver, the payload, and the battery. In the diagram, W represents the load per wheel, while T represents the wheel contact point. In the case of a longitudinal impact assessed under Case 2, it is assumed that the scooter was driving in a scenario where visibility was clear and that the operator applied the brakes upon detecting the obstacle. The z parameter defined under Case 5 represents the acceleration coefficient. When braking, the braking force (B) acting on the center of mass (CM) creates a tipping moment due to the height of the CM , resulting in a load transfer. As a result of this load transfer, the vertical load on the front axle increases, while the vertical load on the rear axle decreases.

4. Definition of the Design Domain

Since vertical, lateral, and longitudinal (braking and acceleration) loads induced by the road surface are directly transmitted to the stub axle, this component is among the most critical parts of the vehicle, as it is subjected to the highest load levels within the system. However, the stub axle has a geometrically complex and variable structure, as it accommodates several sub-components belonging to the steering, suspension, and braking systems. The resulting geometry is governed by the kinematic characteristics of the steering and suspension systems, as well as assembly constraints and joint connection requirements associated with the other components.

In the first stage, the type of ball joint to be used in the vehicle was selected and integrated into the connection points of the upper and lower wishbones. Subsequently, the joint connecting the tie rod to the stub axle, as well as the brake caliper mounting interface, were incorporated into the model.

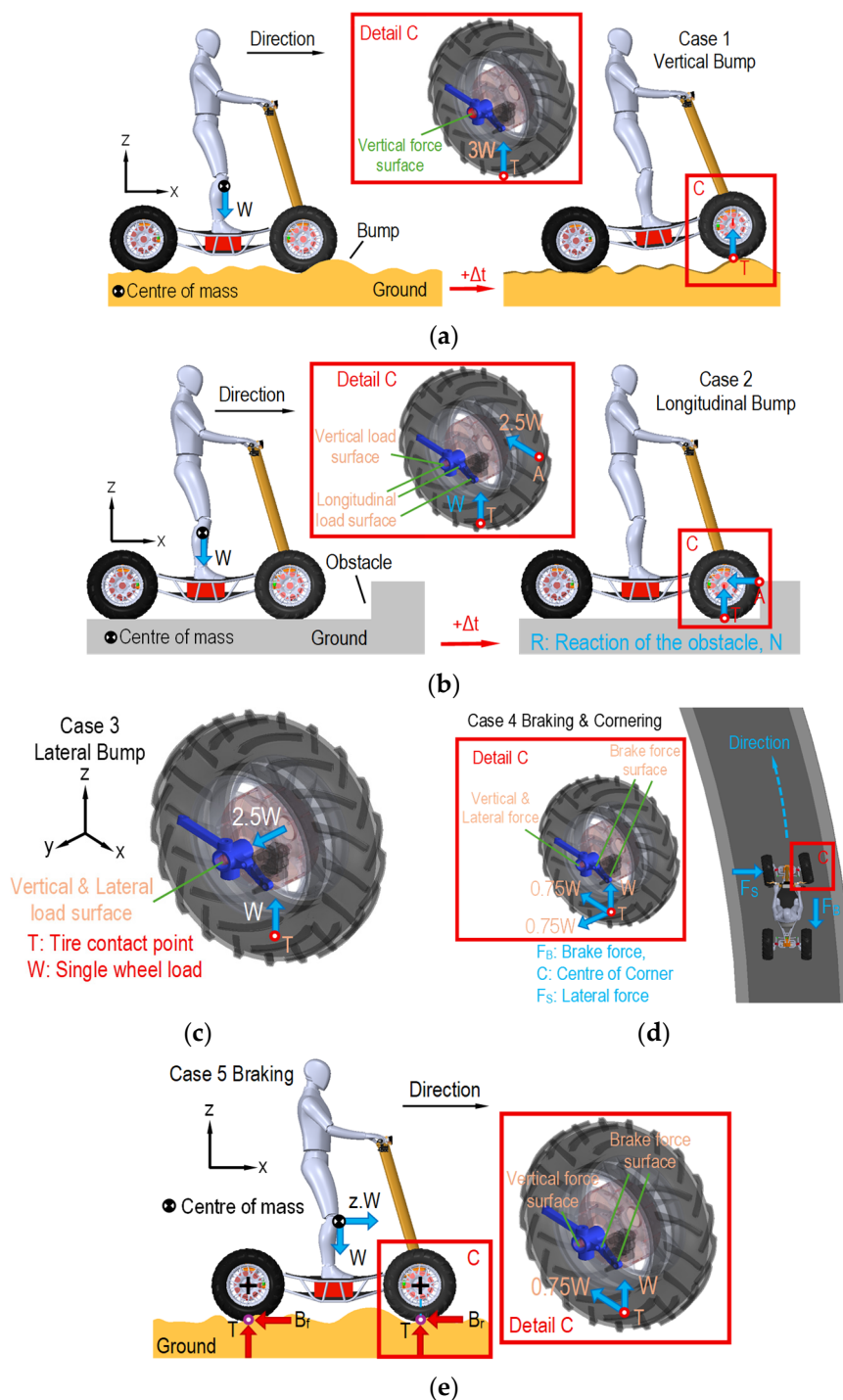


Figure 5. A schematic representation of the forces acting at the tire contact patch: (a) Vertical bump (Case 1); (b) Longitudinal bump (Case 2); (c) Lateral bump (Case 3); (d) Braking and Cornering (Case 4); (e) Braking (Case 5).

To ensure that ball joints and spherical joints can achieve their maximum range of movement under actual operating conditions, all joints have been positioned relative to their center reference points. In other words, the joints have been positioned in such a way as to avoid any angular preload or loss of angular movement capability. The maximum angular displacement limits of the joints during operation are shown in Figure 6. Consequently, the design stages for the stub axle were determined by taking into account the hard-point geometry of the suspension system, the kinematic requirements of the joints, and the assembly constraints. Thereafter, the design domain was defined based on the

region between the relevant joints and connection points, and a simplified geometry was generated for use in the topology optimization process. Points A and B represent the brake caliper mounting locations, while points C and D denote the upper and lower wishbone attachment points, respectively. Point G indicates the tie-rod attachment location, and point F represents the electric motor mounting point.

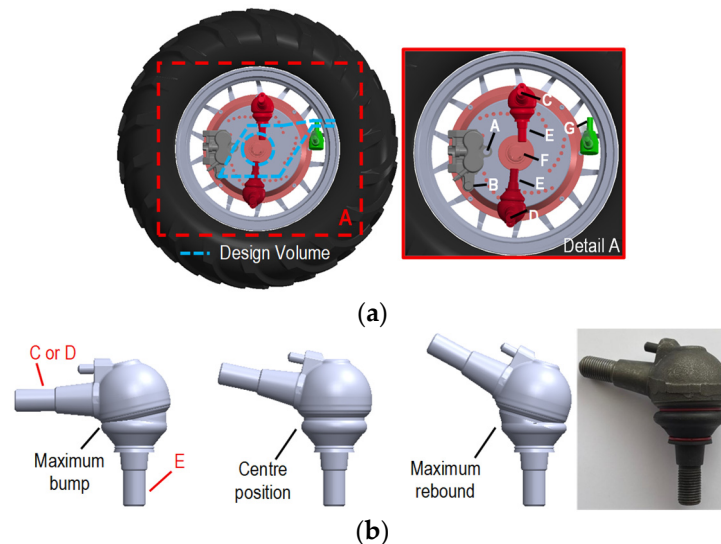


Figure 6. (a) The design envelope specified for the stub axle; (b) the maximum rotation angles of the ball joints.

5. Finite-Element Analysis and Topology Optimization

A schematic diagram of the components, connection points and wheel-hub-motor layout of the electric scooter's front suspension and steering system is shown in Figure 7. Finite-element analyses (FEAs) were carried out using ANSYS® Workbench 2023 R2 (Canonsburg, PA, USA) software.

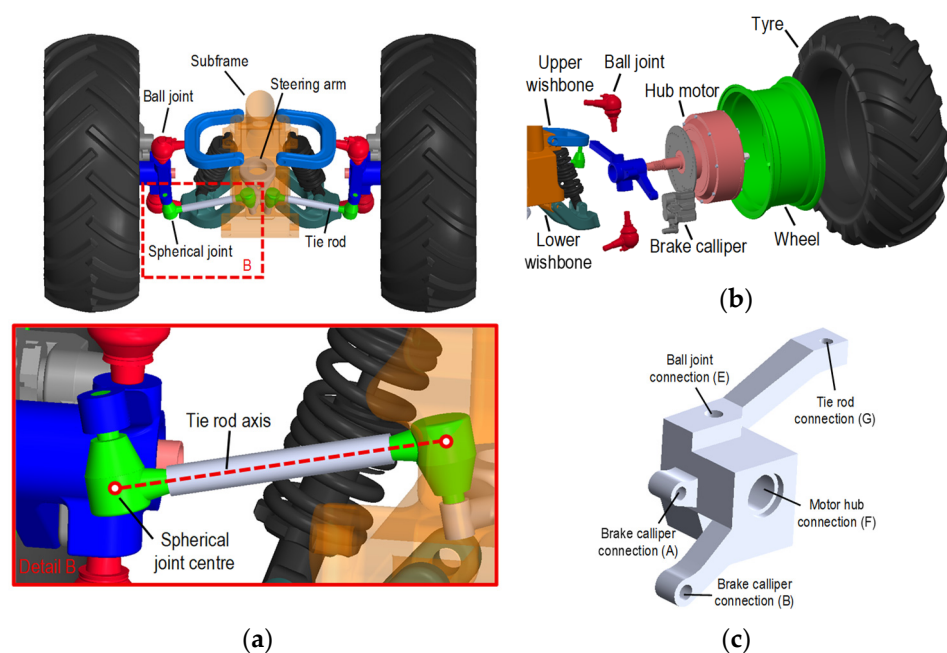


Figure 7. Schematic representation of the vehicle: (a) the suspension and steering system; (b) component layout; (c) a simplified stub axle design and assembly connection points.

The driving conditions and forces acting at the tire contact point used in the analyses were determined in accordance with the standard load cases defined in the open literature, as shown in Figure 5. Subsequently, these loads were rescaled considering the load-carrying capacity of the vehicle. During the topology optimization stage, a simplified stub axle model was built and integrated into the analysis model. In the model shown in Figure 8a, the point where the helical spring is attached to the subframe and the point where it is attached to the lower wishbone are defined. In the analysis model, the rubber protective boot of the ball joint component was not included in the simulations. The connections of the upper and lower wishbones to the subframe were defined as revolute joints, whereas their connections to the stub axle were modeled using spherical joints. All connections associated with the tie rod were also defined as spherical joints.

For the selected load cases, the calculated forces were applied to the model such that the loads were transferred from the tire contact patch to the stub axle hub. Braking and longitudinal loads were applied both to the stub axle hub and to the brake caliper mounting points on the stub axle.

Considering the identified critical loading conditions, the objective was to obtain an optimum geometry with minimum mass while preserving the structural integrity of the stub axle. Accordingly, at the beginning of the optimization process, the regions designated for material removal and the areas to be preserved and excluded from the optimization were separately defined. These regions are illustrated in Figure 8b.

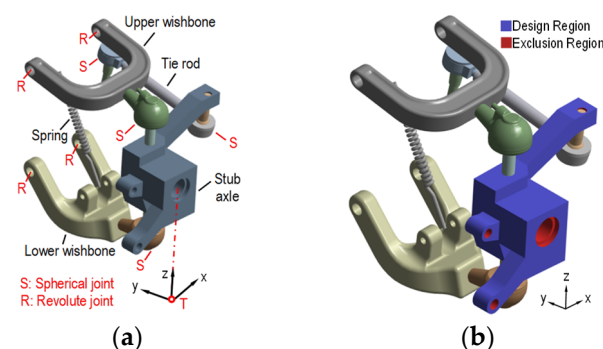


Figure 8. Definitions and boundary conditions of analysis models: (a) FEA model; (b) Topology Optimization model.

The volume constraint is one of the key parameters in topology optimization, governing the balance between structural weight and mechanical performance. Based on a similar vehicle application reported in the literature, a volume fraction of 30% was retained in the present study, corresponding to a material removal ratio of 70% within the design domain [40]. While a high penalty factor creates clearer and more intense regions, very high values can lead to convergence problems. For this reason, values in the range of 3–6 are generally recommended in the literature; in this study, a penalty factor of 3, which is the default value in ANSYS software, was used [41].

In the Case 4 and Case 5 loading scenarios, which represent braking conditions, the brake caliper has also been included in the FE model in order to provide a more accurate representation of the load transfer mechanism. The total braking torque at the wheel hub was calculated using the braking forces shown in Figure 5d,e and the moment arm from the line of action of these forces to the wheel hub center. The resulting braking torque was converted into the force that the brake caliper must apply, taking into account the geometric properties of the brake disc, and applied to the analysis model. The force diagram and finite-element model for these loading conditions are shown in Figure 9. Here, h_t represents the vertical distance between the wheel contact patch and the wheel hub center, B_t represents the braking force at the wheel contact point, M_{BR} represents the braking torque

required to stop the wheel, and F_{BR} represents the force that the caliper must apply to the wheel, calculated based on the brake disc radius (r_{BR}).

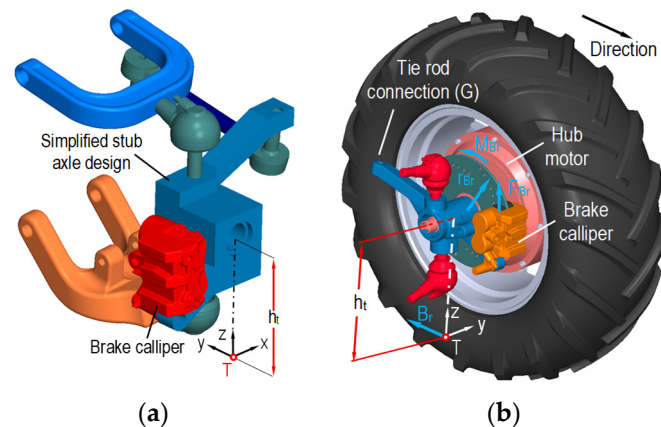


Figure 9. (a) The TO model used in braking load conditions; (b) The forces and moments arising in braking load conditions.

6. Results and Discussion

6.1. Topology Optimization Results

The material distributions obtained from the topology optimization (TO) performed for the five selected critical load cases are presented in Figure 10a. The optimum material layouts obtained for each loading condition were evaluated individually, and the resulting geometrical forms were comparatively examined.

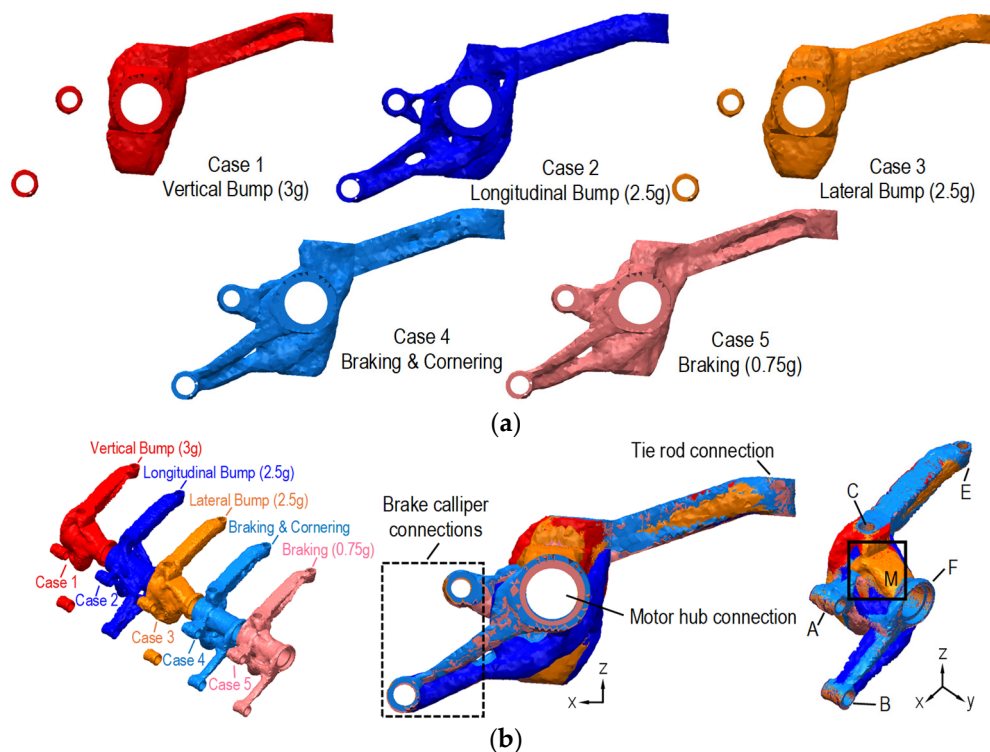


Figure 10. Topology optimization (TO) results for off-road vehicle: (a) Material distributions obtained from TO under the critical load cases; (b) Merged TO results and uniformized geometry.

As a result of this evaluation, the regions that consistently appeared under different loading conditions and were structurally critical were identified. Based on these common material regions, the TO results obtained from the different loading scenarios were

combined and geometrically refined. In this manner, integrated geometry capable of satisfying all critical loading conditions was achieved. The final topology obtained after the merging and redesigning process is presented in Figure 10b. Based on the combined geometries obtained from TO results, the final stub axle design shown in Figure 11 was built. During the final design, the brake mounting area was redesigned not only with manufacturability, but also with solidification and cooling behavior taken into account.

It is recommended to avoid sudden changes in wall thickness in cast components, as this can lead to varying cooling rates, the formation of shrinkage, and a heterogeneous microstructure [42]. For this reason, the wall thickness of the motor connection area, indicated by region F in Figure 10b, has been redesigned to have a more uniform circular cross-section compared to the solid mass defined as region M (Figure 10b). In addition, it is observed that the final design obtained through the topology optimization approach achieves a 36% weight reduction compared to the simplified reference model presented in Figure 8.

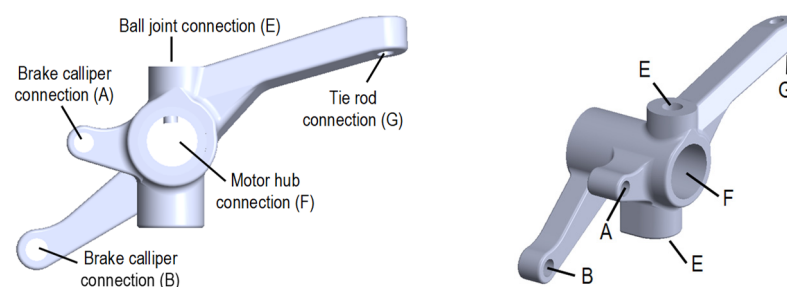


Figure 11. The final stub axle design.

As can be seen in the cross-sectional view shown in Figure 12a, region F represents the critical interface where the motor drive shaft is connected. In the geometry obtained from topological optimization, the wall thickness in region M is greater than that in region F. This situation results in the thicker section in region M solidifying later, despite region F cooling more rapidly. This difference in cooling rates has the potential to lead to the formation of a directional grain structure and grain growth in the transition zone.

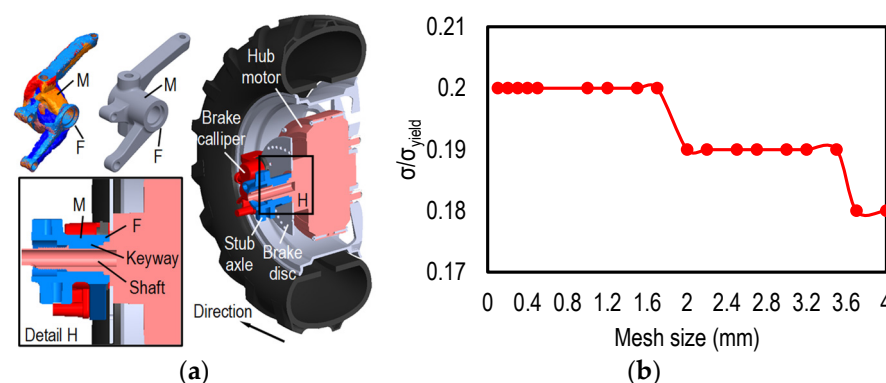


Figure 12. (a) Cross-sectional thickness distribution of the motor mounting area; (b) Normalized maximum stress values for different mesh sizes.

As the region is critical in terms of load transfer, the cross-sectional geometry has been made uniform in order to achieve a more homogeneous solidification behavior. In this context, part of the material in region M has been redistributed to region F. In this way, the total material distribution predicted by the topology optimization has been maintained, while also aiming to achieve a more uniform cross-section and a more balanced

cooling rate in the engine connection region, taking into account casting production constraints.

In the FEA of the final design, a convergence study was conducted for the Case 1 load condition using different mesh sizes with tetrahedral elements to verify the independence of the results from the mesh. Based on the results, a mesh size was selected that would not significantly alter the stress results while keeping the computation time within reasonable limits. In the graph shown in Figure 12b, the stress values obtained for different mesh sizes are presented in normalized form, relative to the material's yield strength. Examining the figure, it can be seen that the normalized maximum stress values do not show a significant change in the 0.4–1.6 mm range and that the solution becomes insensitive to mesh size. Although minor deviations were observed in coarse meshes, a mesh size of 1 mm was determined to be the optimal choice in terms of both numerical accuracy and computational cost. Subsequently, FEAs were repeated for the final design under all critical loading conditions. The principal stress distributions and critical regions subjected to tensile stress have been identified. These are shown in Figure 13.

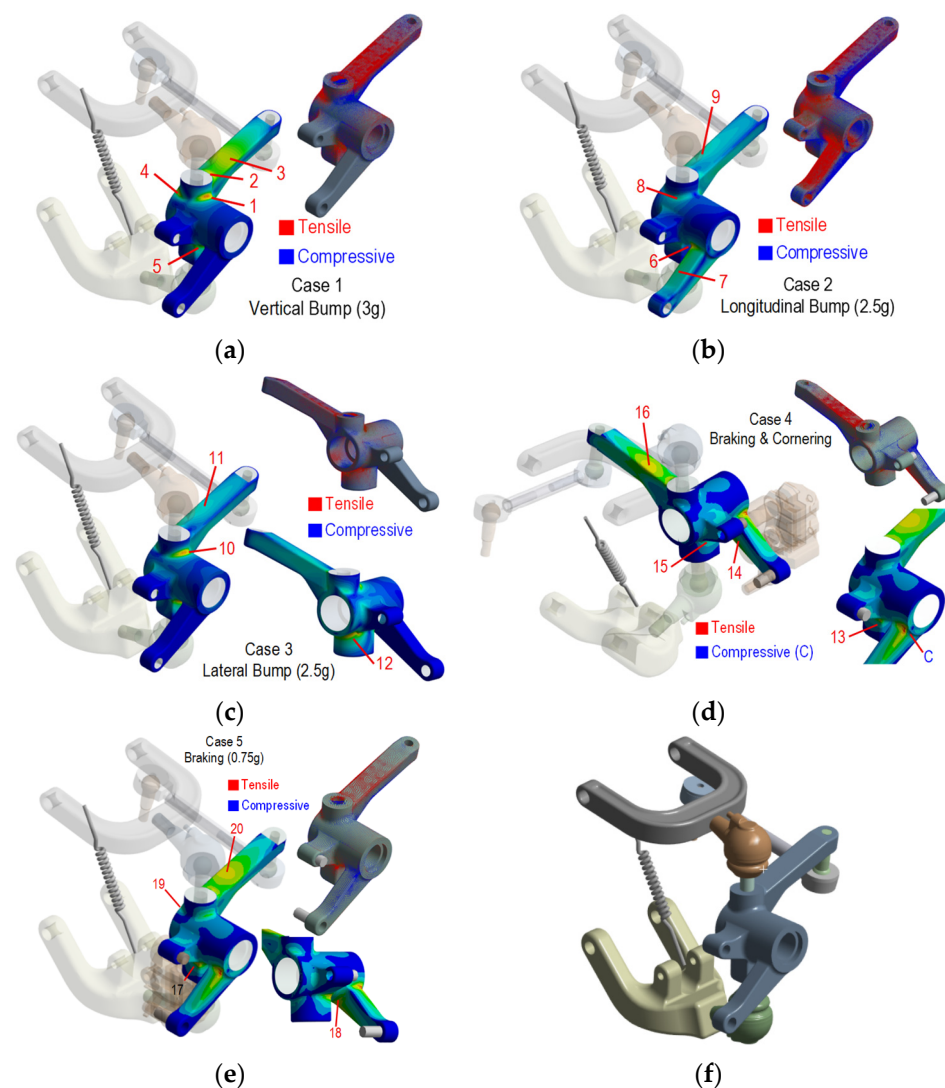


Figure 13. Principal stress distributions and critical stress regions under the selected critical load cases: (a) Case 1; (b) Case 2; (c) Case 3; (d) Case 4; (e) Case 5; (f) Overview of the analysis model.

The graph in Figure 14 shows a comparison of the maximum tensile forces for all load conditions. The stresses in the graph are shown proportional to the yield stress (σ_y). The results indicate that stress concentrations predominantly occur at geometric transition

regions and suspension connection interfaces due to abrupt load transfer and local stiffness variations. Among all loading cases, the longitudinal bump (Case 2) produced the highest normalized stress value, particularly at Point 6, identifying this region as the most critical location in terms of structural integrity. Vertical and lateral bump cases mainly induced bending-dominated stress states, whereas the combined braking-cornering case generated a more complex multi-axial stress distribution. In the braking scenario, the lower arm connection region became critical due to the moment generated by longitudinal braking forces. Overall, the stress distributions reveal that the fatigue-critical regions are concentrated around fillet transitions and load transmission paths.

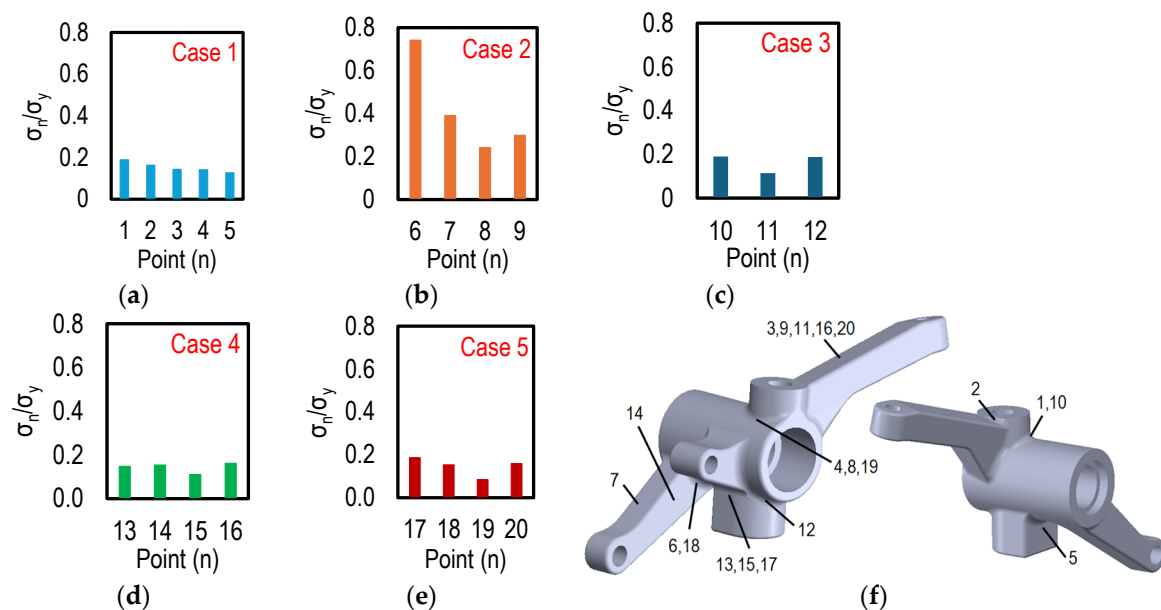


Figure 14. Stress distributions under the critical load cases: (a) Case 1; (b) Case 2; (c) Case 3; (d) Case 4; (e) Case 5; (f) Overview of the points.

Evaluation of the analysis results indicated that the stress generated under all loading conditions remained within safe limits with respect to the material yield strength. The stub axle is considered a safety-critical component, as it accommodates essential systems such as the drivetrain, braking, and suspension assemblies. In addition, it is continuously subjected to cyclic loading under irregular road conditions. Therefore, the fatigue life performance of the final design was evaluated. Within this scope, fatigue analyses were conducted for four different loading conditions using the load case pairs recommended in the literature for fatigue and durability assessments: Case 6 (Vertical Bump, 2.25 g), Case 7 (Cornering Right), Case 8 (Cornering Left), and Case 5 (Braking, 0.75 g) [7]. The principal stress distributions and the corresponding critical regions for the load cases are presented in Figure 15.

During the analyses, the principal stress distributions were examined, and the critical regions subjected to high tensile stresses were identified. The maximum tensile stress values (σ_t) shown in the figure are normalized with respect to the material yield strength (σ_y), and the regions indicated in red represent the maximum tensile stress zones. These maximum stress values (σ_t) were used as the basis for the fatigue calculations.

6.2. Fatigue Life Calculations

The fatigue life prediction of mechanical components is of critical importance, particularly for structures operating under variable loads. Several fatigue life assessment methods have been used to determine the safety and strength limits of a design.

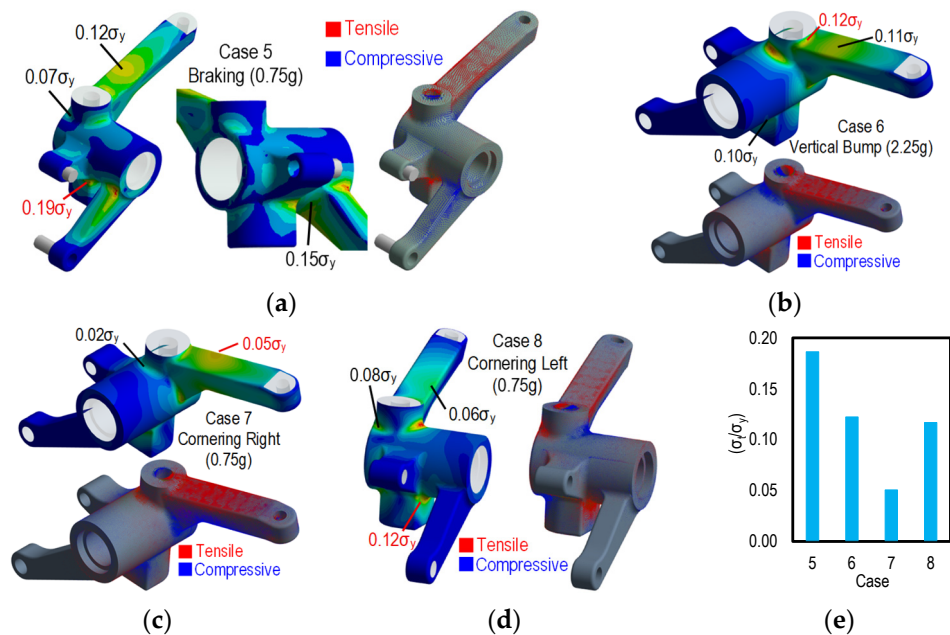


Figure 15. Stress distributions obtained from fatigue load conditions: (a) Case 5; (b) Case 6; (c) Case 7; (d) Case 8; (e) the critical regions subjected to high tensile stresses.

Among these approaches, the Goodman–Haigh method is widely employed for defining the fatigue limit of materials by considering the interaction between mean stress and stress amplitude. The Goodman–Haigh diagram has also been extensively utilized in the literature for fatigue assessment studies [43–45]. For cast iron materials, the stress-life endurance limit (S_e') is expressed in the literature as follows [46]:

$$S_e' = 0.4 \times S_{ut} \quad (3)$$

Here, S_{ut} represents the ultimate tensile strength of the material, and S_e' denotes the modified endurance limit. The S_e' is determined by considering various correction factors such as surface finish, size effect, loading type, and reliability level. The primary reason for this approach is that the endurance limit obtained under laboratory conditions cannot fully represent actual service conditions. Fatigue behavior is significantly influenced by several parameters, including material composition, manufacturing process, applied heat treatments, surface quality, friction, and corrosion. Therefore, in engineering applications, the endurance limit S_e is not used directly; instead, the S_e' is estimated using correction factors representing loading type, surface quality, size effect, and reliability level, as expressed in the following equation [46]:

$$S_e = C_L \times C_S \times C_D \times C_R \times S_e' \quad (4)$$

Here, C_L represents the loading factor, C_S the surface finish factor, C_D the size effect factor, and C_R the reliability factor.

For cast iron materials, the loading factor C_L is recommended as 0.8 in the literature [46]. The average surface roughness value (R_a) was taken from the literature as $0.3 \mu\text{m}$ [47]. Considering this value, the surface finish factor C_S was selected as 0.96 in accordance with the literature [46]. In determining the size factor C_D , the cross-sectional areas of the critical regions subjected to tensile stress under all loading conditions were considered. Using Equations (5) and (6), calculations were performed based on the geometric parameters presented in Figure 16, and the C_D for Cases 5, 6, 7, and 8 were determined as 0.71, 0.69, 0.72, and 0.68, respectively. In addition, the reliability factor C_R was selected as 1 [46].

$$C_D = \begin{cases} 1.24d^{-0.107} & 2.79 \leq d \leq 51 \text{ mm} \\ 1.51d^{-0.157} & 51 < d \leq 254 \text{ mm} \end{cases} \quad (5)$$

$$A_{0.95\sigma} = 0.05hb$$

$$de = 0.808\sqrt{hb}$$
(6)

In the vertical fatigue test, the mean stress (σ_m) and alternating stress (σ_a) are employed. These terms, given in Equations (7) and (8), play a significant role in predicting the fatigue life of the material and determining safe design limits. Here, $\sigma_{\max}$ and $\sigma_{\min}$ represent the maximum and minimum stress values within the loading cycle, respectively. Figure 16d presents a representative loading cycle from which these values were obtained. In this study, it was assumed that the load amplitude remained constant over time (CAL) and that the maximum load condition was continuously applied throughout the entire service life.

$$\sigma_m = \frac{\sigma_{\max} + \sigma_{\min}}{2}$$
(7)

$$\sigma_a = \frac{\sigma_{\max} - \sigma_{\min}}{2}$$
(8)

In Cases 5, 6, and 8, the edge radii located within the maximum equivalent stress regions, which were subjected to tensile stress, have the potential to create a notch effect. For this reason, in addition to the correction factors specified in Equation (4), another factor must also be taken into account.

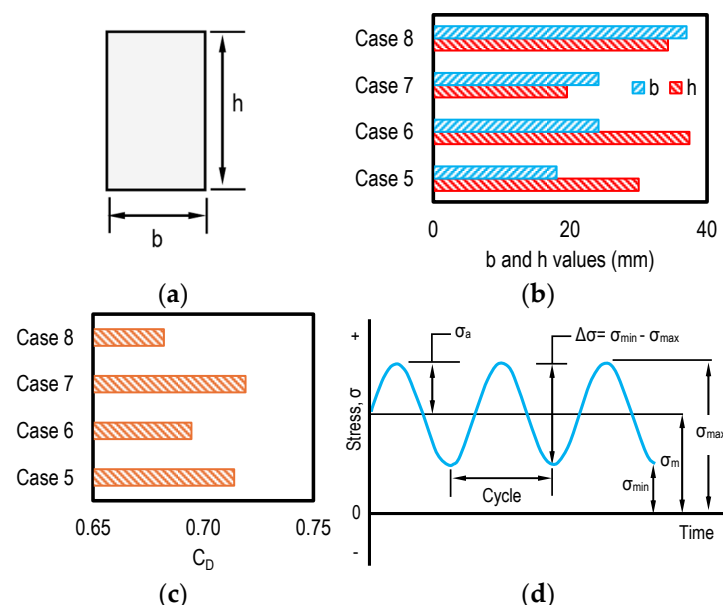


Figure 16. The results: (a) Geometric parameters; (b) Parameter values of critical cross-sectional regions; (c) the size factor (C_D); (d) representative stress cycle.

Accordingly, the k_e factor, which influences fatigue strength, is defined by taking into account the relationship between the static stress concentration factor K_t and the fatigue stress concentration factor K_f . For safety reasons, K_f can be assumed to be equal to K_t [48]. Because of the shape complexity of the axle, K_t cannot be derived from data in the standard literature. On the other hand, K_t and k_e are defined as follows:

$$k_e = \frac{1}{K_f}$$
(9)

$$K_t = \frac{\sigma_{\text{peak}}}{\sigma_{\text{nominal}}}$$
(10)

Here, σ_{peak} represents the local maximum stress at the notch root, while σ_{nominal} represents the nominal stress that would occur in the absence of the stress concentration effect [48,49]. The σ_{peak} value was used as the maximum stress value obtained from static FEA

[49]. Figure 17 shows the critical cross-sections where the maximum tensile stresses occur under the considered loading conditions. In the figure, W_b denotes the bending section moduli. For Case 5, the critical cross-section was decomposed into a circular hollow section and a rectangular section, as illustrated in Figure 17d. The corresponding section moduli were calculated, and the nominal stress values were determined using Equation (11). Since the location of the maximum stress in Case 8 coincides with that of Case 5, i.e., the same cross-sectional geometry governs both cases, the corresponding section is not presented separately in Figure 17. The nominal stress values for Case 8 were calculated following the same procedure.

$$\sigma_e = \frac{M_e}{W_b} \quad (11)$$

where M_e is the bending moment, and W_b is the bending section modulus.

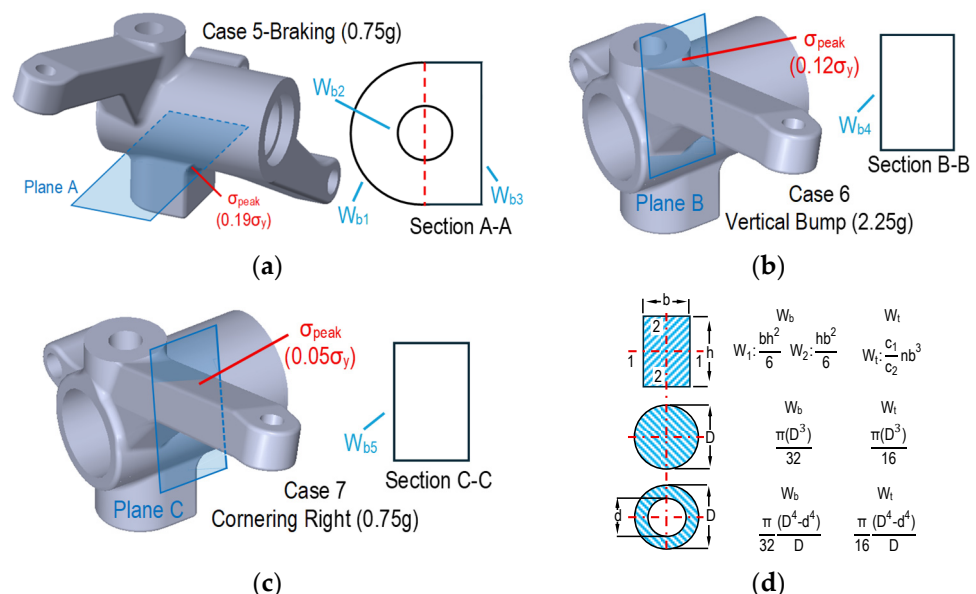


Figure 17. Critical cross-sections associated with the peak stress locations for: (a) Case 5; (b) Case 6; (c) Case 7; (d) Cross-sectional geometries and corresponding bending (W_b) section moduli.

Firstly, loading diagrams were generated for the critical regions containing notches in the fatigue calculations, and the free-body diagrams for Case 5 and Case 6 are shown in Figure 18. Here, F_A and F_B denote the reaction forces acting on the upper and lower spherical joints, respectively; the numerical values of these forces were obtained from the FEA. In the Case 5 loading condition shown in Figure 18a, the Plane A section is predominantly subjected to the F_{B5} force; taking into account the distance a , which is the distance from the joint center to the critical section, a bending moment of magnitude $M_e = F_{B5}a$ is observed in the section. Similarly, in the Case 6 loading condition shown in Figure 18b, it was determined that the dominant load type at the critical section was bending; the bending moment was calculated using the F_{A6} force and the distance b from the joint center to the critical section.

Subsequently, the section modulus (W_b) of the critical sections shown in Figure 17d was calculated, and the nominal stresses (Equation (11)) were obtained using these values. The obtained nominal stresses ($\sigma_{nominal}$) were divided by the maximum stresses at the notch root (σ_{peak}) determined from the FEA results, and the stress concentration factor (K_t) was calculated according to Equation (10) [49]. These calculations were performed separately for Case 5, Case 6, and Case 8. The calculated K_t values were used in conjunction with the strength limit (S_e) defined in Equation (4) to evaluate the fatigue strength for the notched geometry. As there was no notch in the critical stress region under the loading

condition in Case 7, fatigue calculations based on notch effects were not applied in this case, and the calculated S_e value was used directly in the assessments.

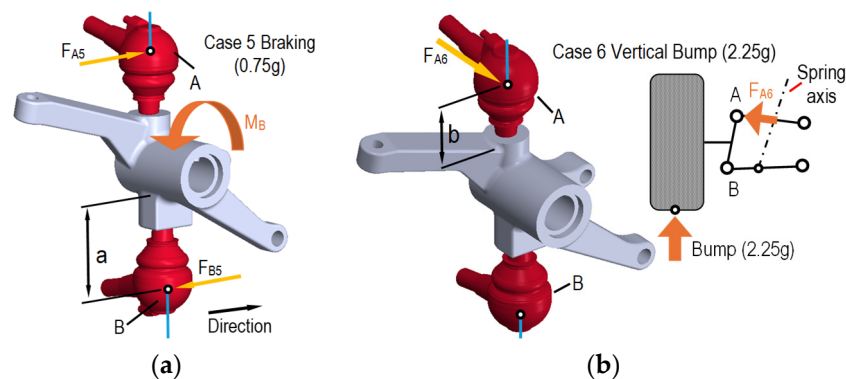


Figure 18. Force diagrams for load conditions: (a) Case 5; (b) Case 6.

In the calculations, the value of $\sigma_{\min}$ was assumed to be zero. The primary reason for this assumption is that the occurrence of a negative vertical contact force at the wheels during a severe bump event is not physically possible. In other words, when the tire loses contact with the road surface, the corresponding stress value becomes zero. Furthermore, test results in the literature indicate that the fatigue behavior of axle components generally varies between a minimum stress value close to zero and a maximum stress value, and that this assumption is appropriate for engineering applications [49,50]. Using the obtained stress values and the selected correction factors, the Goodman–Haigh diagram presented in Figure 19 was constructed.

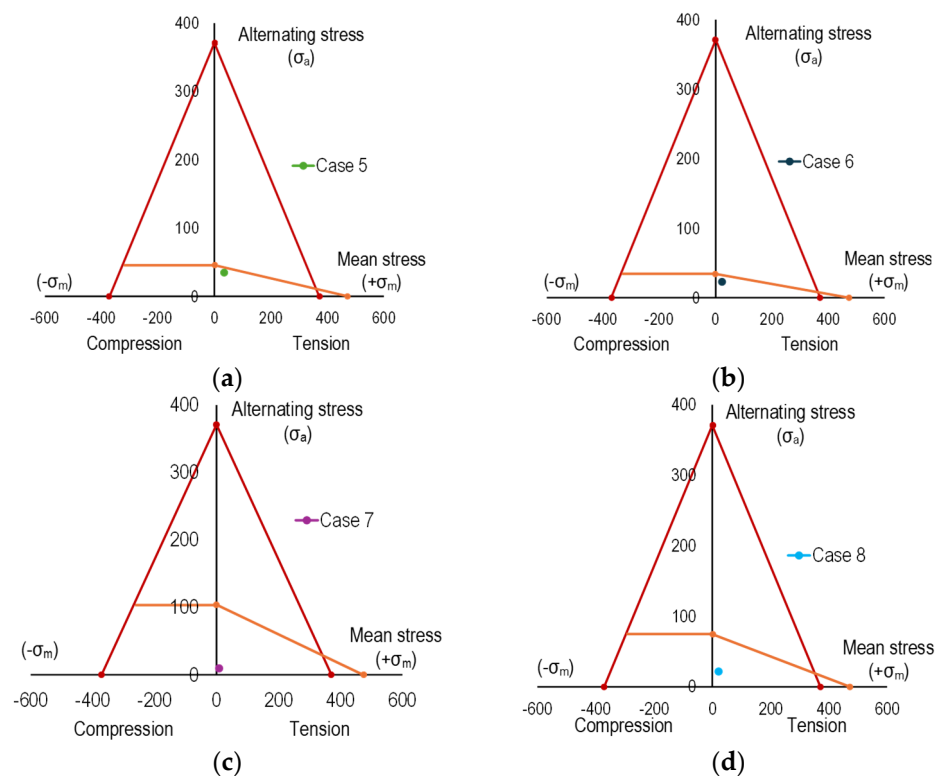


Figure 19. Goodman–Haigh diagram for fatigue assessment of the four loading cases : (a) Case 5; (b) Case 6; (c) Case 7; (d) Case 8.

The graph shows that all the stress values obtained at critical areas of the final design remain within the infinite life area; thus, the design is considered to be safe. The

prototype of the stub axle, which was determined to be within the infinite life zone, has been manufactured.

The stub axle prototype is shown in Figure 20. The final mounting holes of the stub axle were machined using subtractive manufacturing processes, followed by a surface painting operation. In the manufacturing process, the as-cast raw part was used initially (Figure 20a), and subsequently, machining operations were performed in accordance with the final geometry (Figure 20b).

This study aims to establish a structural design roadmap for the stub axle using topology optimization and to obtain a fail-safe design. The results provide a holistic perspective in which the indirect effects on manufacturing efficiency and sustainability can also be evaluated. Cast iron production is widely recognized as an energy-intensive process, particularly due to the significant contribution of melting and metal processing stages to the total energy consumption [51]. It has been reported in the literature that the energy consumption in cast iron production varies between 500 and 1300 kWh per ton [52]. In this context, if the lightweight design approach obtained through topology optimization is implemented in mass production scenarios, the potential reduction in material usage may have a direct impact on manufacturing energy consumption. Similarly, raw material cost constitutes a significant portion of cast iron component manufacturing, making mass optimization a critical design parameter from a production cost perspective. Considering an average pig iron price of approximately 400 USD/ton (≈ 0.40 USD/kg) [53], each kilogram of potential material reduction at the design level can lead to considerable economic savings in mass production scenarios. Furthermore, since four stub axle components are required in a typical four-wheeled vehicle architecture, the cumulative effect becomes more pronounced at the vehicle level in terms of both economic and environmental impact. This clearly indicates that material selection, design methodology, and manufacturing processes should be evaluated not only in terms of structural performance but also from the perspective of energy efficiency and cost optimization.

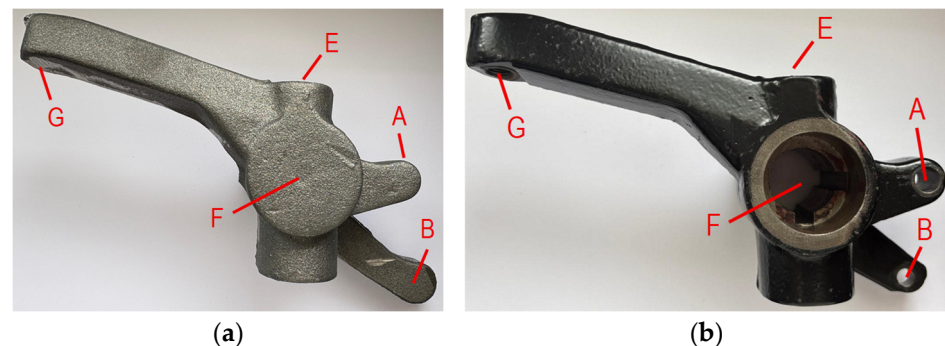


Figure 20. The prototype of the stub axle: (a) The as-cast part; (b) The final geometry following machining.

In light of these considerations, it can be concluded that the topology-optimized stub axle design exhibits significant potential not only in terms of structural and fatigue performance but also in enhancing energy efficiency and cost-effectiveness in scalable manufacturing applications.

Figure 21 presents the 4 × 4 scooter designed and manufactured through the collaboration between the Dokuz Eylul University and Tibet Makina A.Ş. and investigated within the scope of this study.



Figure 21. Prototype 4 × 4 electric scooter with independent suspension designed and manufactured within the scope of the study (Courtesy of Tibet Makine Sanayi ve Ticaret A.Ş.).

7. Conclusions

This study presented a structural design roadmap for a stub axle of an electric scooter using topology optimization, with the objective of achieving a fail-safe design. In the study, multiple load scenarios from the literature were utilized. To the best of the authors' knowledge, no study in open literature has been found that combines topology optimization with fatigue life assessment for the stub axle component used in electric scooters.

- In the initial phase of the study, the design space was defined by specifying the ball joint configuration and mounting requirements. Subsequently, a simplified analysis model was built, and topology optimization was carried out under critical load scenarios selected from the standard load conditions reported in the literature. The final geometry obtained as a result of topological optimization was examined using finite-element analysis to investigate stress concentrations arising under critical load scenarios.
- The analysis results showed that, under all selected loading conditions, the maximum stress values remained within the yield strength limits of the material.
- As a result of the FEA, the maximum principal stress distributions of the stub axle were evaluated, and potential failure regions subjected to tensile stresses were identified.
- In the fatigue life assessment carried out using the Goodman–Haigh approach, it was determined that the stress values in the regions subjected to tensile stress fall within the infinite life region.
- A 36% mass reduction was achieved through topology optimization compared to the simplified analysis model.

The evaluation of energy and cost aspects carried out as part of the study also indicates that weight reduction offers significant potential benefits in terms of production costs and energy consumption.

- Based on the energy consumption values reported in the literature for cast iron production, it has been observed that the mass reduction achieved could result in energy savings of approximately 0.49–1.28 kWh per part, and that a one-kg reduction in mass achieved within the scope of this study could correspond to a cost saving of approximately 0.40 USD per component.

The results demonstrate that the integrated use of topology optimization and fatigue assessment in the design of critical components—such as stub axles used in light electric vehicle applications—constitutes a safe and efficient engineering approach. This study also offers potential benefits in terms of energy efficiency and cost optimization in scalable production applications.

This study presents a design roadmap for the fail-safe development of a vehicle component with predefined hard points. This study does not include experimental validation. In future studies, it is planned to carry out experimental tests representing the loading conditions examined and to validate the numerical results using stress and dynamic response data obtained from strain gauges and accelerometers. For future work, different material alternatives can be investigated to further enhance lightweighting performance. In addition, the resulting mass reduction can be evaluated in terms of its associated energy savings and material cost benefits, particularly in mass production scenarios.

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