

FACULDADE DE ENGENHARIA DA UNIVERSIDADE DO PORTO

Finite Element Analysis of Bicycle Frame Impact Testing: Simulation and Validation

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Mestrado em Engenharia Mecânica

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Resumo

A bicicleta desempenha um papel fundamental na mobilidade urbana sustentável, oferecendo uma alternativa ecológica ao transporte motorizado, promovendo o bem-estar físico e reduzindo o congestionamento nas cidades. Com o aumento significativo do seu uso, garantir a segurança e durabilidade dos quadros tornou-se um desafio crítico para a engenharia. Para responder a esta necessidade, os fabricantes têm vindo a adotar metodologias digitais de ensaio para complementar a validação experimental.

Este trabalho incide na correlação entre os resultados obtidos por simulações de impacto com o método dos Elementos Finitos (FEA) e os resultados experimentais realizados em condições reais na empresa Gazelle. O objetivo é validar a precisão e a fiabilidade das simulações na previsão do desempenho do quadro sob condições extremas. Utilizando o software Dassault 3DEXPERIENCE Simulia, o ensaio de queda do quadro é replicado digitalmente para avaliar a integridade estrutural, os critérios de falha e as características de deformação. O estudo segue as normas ISO 4210 e NEN-EN 15194, incorporando ainda os requisitos internos da Gazelle, que impõem critérios mais conservadores.

São analisados aspetos como o impacto das propriedades do material, dos processos de soldadura e das simplificações do modelo na fidelidade das simulações. Através da comparação sistemática entre os resultados numéricos e experimentais, este estudo visa refinar as metodologias digitais, contribuindo para a digitalização do laboratório de testes da Gazelle. Os resultados obtidos apoiam o desenvolvimento de uma abordagem *first-time-right*, reduzindo a dependência de protótipos iterativos, melhorando a eficiência do processo de engenharia e acelerando o ciclo de desenvolvimento de novos componentes.

Abstract

Bicycles play a fundamental role in sustainable urban mobility, offering an environmentally friendly alternative to motorized transport while promoting physical well-being and reducing urban congestion. As cycling continues to gain popularity, ensuring the safety and durability of bicycle frames remains a critical engineering challenge. To meet this demand, manufacturers are increasingly adopting digital testing methodologies to complement physical validation.

This study focuses on the correlation between Finite Element Analysis (FEA) impact testing and real-life experimental results at Gazelle. The objective is to validate simulation accuracy and reliability for predicting frame performance under extreme conditions. Using Dassault 3DEXPERIENCE Simulia, the *frame falling test* is digitally replicated to assess frame integrity, failure criteria, and deformation characteristics. The study adheres to ISO 4210 and NEN-EN 15194 standards while incorporating Gazelle's internal safety requirements, which impose more conservative criteria.

Key aspects include investigating the effects of material properties, welding processes, and model simplifications on simulation fidelity. By systematically comparing simulation outputs with physical test data, this research aims to refine digital methodologies, ultimately contributing to the digitization of Gazelle's test lab. The findings support the development of a first-time-right design approach, reducing reliance on iterative prototyping, improving engineering efficiency, and accelerating the development cycle of new components.

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Another chapter of this journey comes to an end. I'm grateful for the opportunities, the people, the conversations, and everything I've learned along the way. To everyone who crossed paths with me or contributed in any form, thank you. I've grown because of it, both professionally and personally.

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Thank you to Jordi Lowijs for the guidance, for keeping me sharp in the way I approached the work, and for the support and friendship throughout my time in the company. To Guido van Rijckevorsel, thank you for the opportunity to be part of the team, for your ideas, your willingness to collaborate, and your genuine interest in everything going on around you. That kind of engagement is rare and valuable. To Maarten Pelgrim, all the best in continuing to lead such a talented group.

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To my family, thank you for the constant support, for always being there, and for giving me the space to follow the path I believed in. To my friends, thank you for shaping who I am, and for always being ready to listen when I needed to talk.

To all the colleagues, forum users, and software support teams who helped along the way, thank you for your time and your input. Even without direct contact, your support made a difference.

In the end, we're all trying to find a sense of purpose in doing what we care about. I believe the world would be better with more bicycles on the streets, and I'm proud to have contributed in some way to that. I'm thankful for this experience and looking forward to what comes next.

Pedro Fernandes Martins de Sousa

“The bicycle is a means to experience pain, passion, and pure happiness, all within the same moment.”

Marco Pantani

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List of Acronyms

BEV	<i>Battery Electric Vehicle</i>
CAD	<i>Computer-Aided Design</i>
CFRP	<i>Carbon Fiber Reinforced Polymers</i>
CoM	<i>Center of Mass</i>
DIC	<i>Digital Image Correlation</i>
EPAC	<i>Electrically Power Assisted Cycles</i>
FEA	<i>Finite Element Analysis</i>
FEM	<i>Finite Element Model</i>
FPS	<i>Frames per Second</i>
GHG	<i>Greenhouse Gas</i>
HAZ	<i>Heat-affected Zone</i>
ICEV	<i>Internal Combustion Engine Vehicle</i>
ISO	<i>International Organization for Standardization</i>
JC	<i>Johnson-Cook</i>
NDI	<i>Non-Destructive Inspection</i>
PEEQ	<i>Equivalent Plastic Strain</i>
RMSE	<i>Root Mean Square Error</i>
UTS	<i>Ultimate Tensile Strength</i>
YS	<i>Yield Strength</i>

Chapter 1

Introduction

This thesis investigates the development, validation, and refinement of a Finite Element Analysis (FEA) methodology for simulating bicycle frame impact tests, with particular emphasis on the *frame falling test*. By correlating numerical simulations with experimental data, the objective is to establish a foundational framework that can be adopted to enhance Gazelle's design process. This integration aims to improve engineering efficiency, elevate product safety, and reduce dependency on physical testing.

The chapter begins by introducing Gazelle, the host organization, and its role in testing and validating bicycle frame performance. This is followed by a discussion on structural integrity, international safety standards, and the motivation for simulation-based impact assessment. The main objectives of the thesis are then outlined, along with the methodology adopted throughout the project. The chapter concludes with an overview of the dissertation structure.

1.1 Host Organization

Gazelle is a renowned bicycle manufacturer headquartered in the Netherlands, internationally recognized for its innovative and sustainable cycling solutions. With a legacy spanning over 125 years, Gazelle designs and produces bicycles that prioritize comfort, safety, and performance, especially for urban and recreational contexts.

The company has established itself as a leader in incorporating modern engineering practices into traditional bicycle manufacturing. Gazelle complies with international standards such as ISO 4210 and NEN-EN 15194, while also implementing more stringent internal testing protocols to ensure enhanced durability and rider safety.

This thesis was developed in collaboration with the internal test laboratory at Gazelle, which plays a pivotal role in validating the structural integrity of bicycle components. The project supports the strategic initiative of the company to digitize testing procedures by developing a validated FEA methodology for impact analysis. Enhancing predictive accuracy through numerical simulations contributes to the vision of Gazelle to achieve a *first-time-right* approach in the development of new frame designs.

1.2 Structural Integrity and Safety of Bicycle Frames

Ensuring the structural integrity of a bicycle frame is essential for rider safety, especially in impact scenarios such as crashes or abrupt load application. A well-engineered frame must endure a combination of axial, bending, and torsional forces without compromising its structural stability.

To ensure reliability and safety, the bicycle industry follows internationally recognized testing protocols. Two key standards are ISO 4210 and NEN-EN 15194, which define performance and durability requirements for conventional and electrically assisted bicycles, respectively.

ISO 4210 specifies mechanical performance criteria for traditional bicycles, covering fatigue, impact, and static load tests for frames, forks, and other structural components. These requirements ensure that the frame can withstand both typical and extreme usage conditions [1].

NEN-EN 15194, in turn, targets Electrically Power Assisted Cycles (EPACs) and includes additional provisions related to electrical components, battery safety, and motor-assisted performance. Structurally, it builds upon ISO 4210, incorporating similar tests adapted to the higher loads and unique operating conditions associated with e-bikes [2].

By adhering to these standards, manufacturers can ensure their products meet stringent safety, reliability, and durability benchmarks.

Experimental research has shown that frame geometry and material choice have a significant impact on structural performance. FEA has emerged as a powerful tool in this domain, enabling detailed evaluation of load paths, stress concentrations, and potential failure locations [3]. Even slight changes in tube geometry, weld location, or material properties can noticeably affect structural response, highlighting the importance of precision in the design phase.

A major factor influencing durability is the welding process. Heat-affected zones (HAZ) introduced during welding often contain residual stresses that may reduce fatigue life and impact resistance. For high-strength aluminum alloys frequently used in frame construction, such localized effects are critical to understand and mitigate [4]. Advanced computational methods, including harmonic fatigue analysis, are increasingly employed to evaluate these phenomena prior to prototyping [5].

In response to these challenges, this thesis aims to establish a validated correlation between numerical frame falling impact simulations and real-world experimental testing. In doing so, it contributes to more predictive and efficient frame design methodologies, aligned with the vision of Gazelle for digitized and first-time-right product development.

1.3 Objective of the Thesis

The objective of this thesis is to develop and validate a FEA methodology for simulating frame falling impact tests, with a focus on reproducing the global deformation observed in physical experiments. Rather than investigating failure mechanisms in detail, the primary aim is to ensure that the simulation accurately reflects the overall structural response under impact loading.

Validation is performed by comparing the deformation patterns from the simulation with ultra-slow-motion experimental footage, using visual tracking of reference points on the frame. This correlation serves to assess the reliability of the material model, boundary conditions, and general setup employed in the simulation.

Establishing confidence in the global response is a key prerequisite for future work focused on local failure criteria. As such, the validated methodology developed in this thesis is intended as a foundational step toward the integration of impact simulation into the digital test lab at Gazelle, supporting more predictive and efficient product development processes.

The research adopts a structured and iterative approach aligned with scientific and engineering best practices, focusing on the validation of global deformation response under impact:

1. **Literature Review and Contextual Research** – The study begins with a comprehensive review of existing work on impact testing and simulation of bicycle frames. This includes both experimental methods and numerical strategies using FEA. Relevant regulatory standards such as ISO 4210-6:2023 and EN 15194:2017+A1:2023 are examined to ensure alignment with industry procedures. Key academic contributions, including Covill et al. [6] and Alves [7], inform the simulation setup and validation methodology.
2. **Numerical Simulation Setup** – A detailed FEA model of the selected Gazelle frame is constructed in 3DEXPERIENCE, replicating the frame falling impact scenario. The simulation incorporates realistic geometry, boundary conditions, impact configuration, and material properties of aluminium 6061-T6 (Al 6061-T6). Particular attention is given to mesh quality and contact definitions to ensure computational robustness.
3. **Experimental Testing** – Real-life frame falling tests are performed on the bicycle frame, following internal testing protocols that reproduce the standardized impact scenario. Ultra-slow-motion video is employed to capture deformation during impact, enabling accurate tracking of displacement at predefined reference points.
4. **Correlation and Model Refinement** – The deformation pattern obtained from the simulation is compared to the experimental results, focusing on overall displacement and structural response rather than localized failure. The goal is to achieve reliable global correlation, laying the groundwork for future studies involving the integration of damage criteria on the model and more detailed material characterization.

This comparison provides the basis for evaluating the practical applicability of the simulation framework within the design and engineering workflow at Gazelle.

1.4 Methodology Followed in the Project

A flowchart summarizing the overall methodology and the sequence of steps followed in this project is presented in Figure 1.1, providing a clear overview of the systematic approach adopted.

The methodology begins with the creation of the CAD geometry, followed by geometry verification, weight scaling (comparing the mass of physical components with simulation outputs), material assignment, mesh generation, and definition of the mechanical scenario. After setting up the simulation in 3DEXPERIENCE, initial trial runs are used to refine mesh quality and connection definitions based on observed results. Once a stable setup is achieved, the frame falling impact test is simulated. In parallel, experimental testing is conducted on the physical frame. Finally, simulation results are compared to experimental observations to assess the correlation of global deformation, validating the effectiveness of the chosen modeling approach.

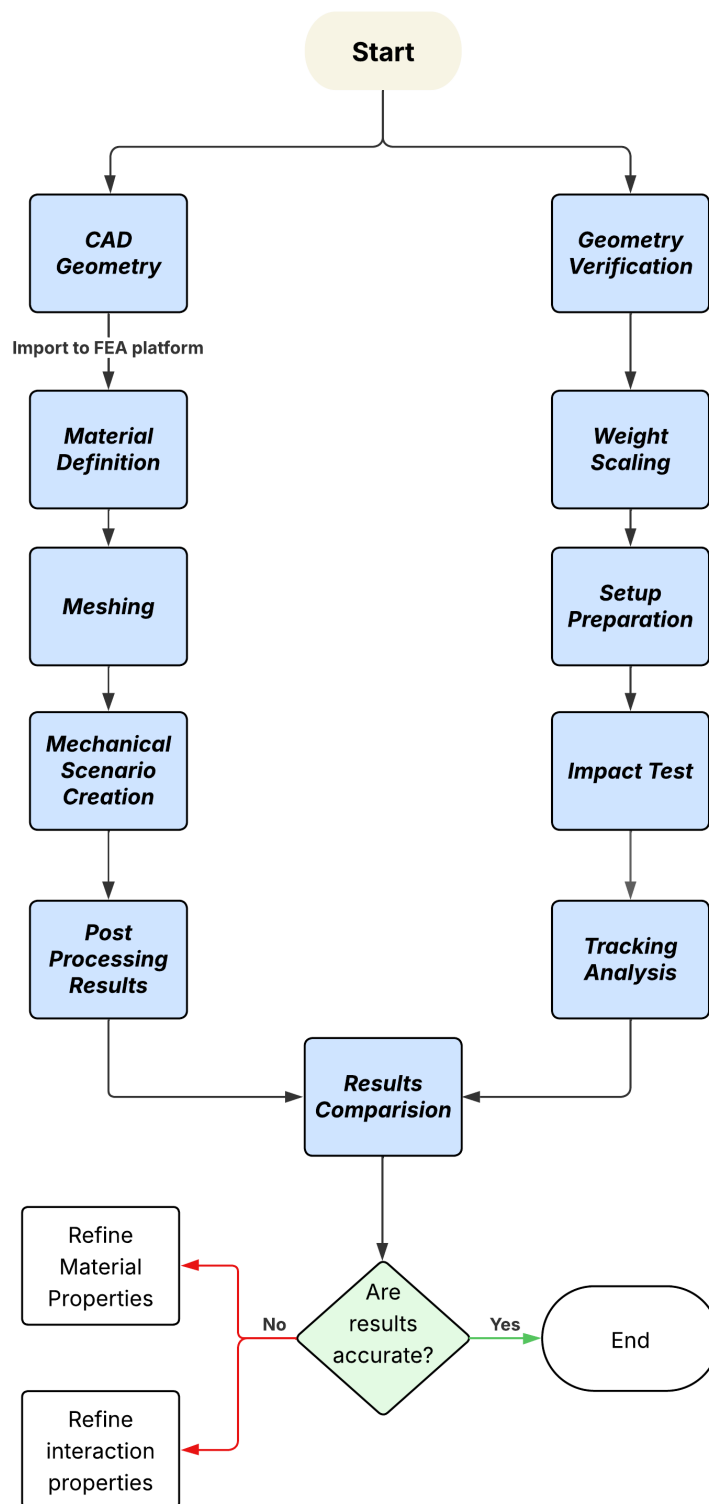


Figure 1.1: Flowchart summarizing the project methodology. Simulation refinement was carried out prior to comparison with experimental results. The physical testing and simulation were conducted in parallel, converging at the comparison stage.

1.5 Dissertation Structure

In addition to the introduction, this dissertation contains five more chapters.

In Chapter 2, the role of bicycles in the context of urban mobility is discussed, highlighting their relevance as a sustainable mode of transportation and the challenges associated with modern urban environments.

Chapter 3 presents the state of the art in bicycle frame analysis using FEA, including a review of relevant studies, material selection criteria, mechanical properties of common frame materials, and the plasticity model employed in numerical simulation.

In Chapter 4, the adopted simulation methodology is described, detailing the FEA environment, model setup, and key input parameters used for replicating the frame falling impact scenario.

Chapter 5 focuses on the experimental procedure, describing the physical testing process and presenting a comparative analysis between the observed results and the simulation outcomes.

Finally, Chapter 6 summarizes the main findings of the study, evaluates the results in light of the initial objectives, and outlines recommendations for future work.

Chapter 2

Urban Mobility and the Role of Bicycles

Urban mobility is a fundamental aspect of contemporary urban life, significantly influencing the functioning of cities and the daily experiences of millions of people. As urban populations continue to expand, the demand for transportation solutions that are not only efficient and accessible but also sustainable becomes increasingly critical. This growing imperative has driven a transformation in urban mobility systems, with bicycles emerging as a central component in providing environmentally friendly and cost-effective transportation options.

The increasing adoption of cycling goes beyond being a mere trend; it addresses pressing urban challenges such as traffic congestion, air pollution, and the promotion of healthier lifestyles. In response to the pressing need to reduce urban carbon emissions and mitigate global warming, bicycles offer an effective and sustainable mode of transportation that contributes directly to these objectives. Moreover, cycling fosters physical activity, contributing to the mitigation of health issues related to sedentary behavior, including obesity, cardiovascular diseases, and diabetes [8].

Beyond environmental and health benefits, cycling also enhances overall urban livability by creating quieter, safer, and more pedestrian-friendly environments. By reducing reliance on motorized vehicles, bicycles help alleviate traffic congestion and decrease noise pollution, thereby improving the quality of life in urban areas.

Recognizing these benefits, urban policymakers are increasingly integrating cycling into comprehensive mobility strategies. Such policies emphasize the integration of cycling with other transport modes, the development of dedicated cycling infrastructure, and ensuring equitable access to bicycles for all residents. As a result, bicycles are becoming a pivotal element in the evolution of more sustainable and resilient cities [8].

This chapter aims to provide a broad overview of the role of bicycles in urban mobility. It will examine the rising importance of cycling in urban contexts, explore its social, environmental, and economic advantages, and discuss the challenges involved in promoting bicycles as a sustainable mode of transport. By the end of this chapter, readers will gain a comprehensive understanding of how bicycles contribute to urban mobility systems and the ongoing efforts to foster healthier, more sustainable, and efficient cities.

2.1 The Role of Bicycles in Modern Mobility

Bicycles have become an integral part of urban transportation, offering a sustainable, cost-effective, and environmentally friendly alternative to motorized vehicles. As cities experience population growth and congestion, the demand for sustainable mobility solutions has intensified. Bicycles, due to their efficiency and minimal environmental impact, present a practical solution to reducing traffic congestion, lowering carbon emissions, and improving the quality of urban life.

The integration of cycling into transportation systems is particularly notable in some European cities. In countries such as the Netherlands, Denmark and Germany, cycling is not only a recreational activity, but also a common mode of transport. In these nations, cycling rates often surpass car usage for short-distance trips. For example, in the Netherlands, approximately 28% of all trips are made by bicycle, compared to just 2% in the United States, where the use of cars predominates [9]. This growing trend towards cycling is a clear indication of a societal shift towards more sustainable mobility practices and healthier lifestyles.

The next pie chart, Figure 2.1, shows the modal split of all journeys made in the Netherlands in 2022.

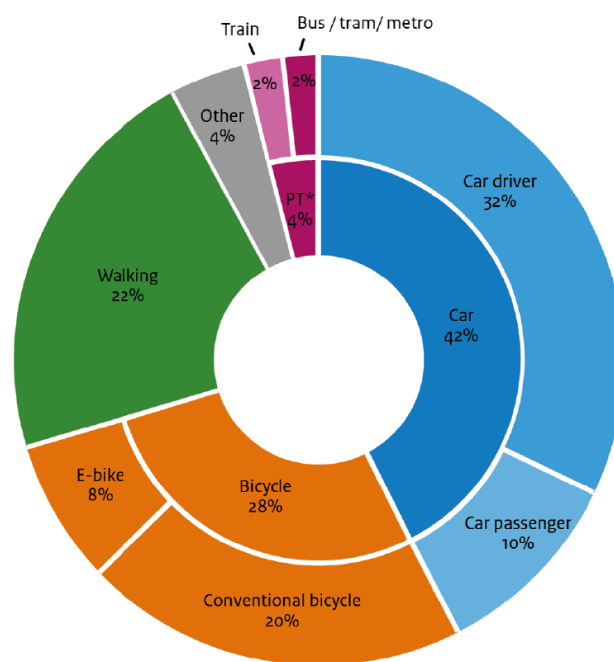


Figure 2.1: Total amount of journeys per mode of transport (walking, bicycle, car, bus/tram) in The Netherlands in 2022 [9].

Notably, cycling plays a major role in daily mobility:

- 28% of all journeys are made by bicycle, of which 20% are by conventional bicycles and 8% by e-bikes;
- This puts cycling ahead of walking (22%) and all forms of public transport combined (4%);

- Car usage remains the largest, with 42% of journeys involving a car (as driver or passenger).

This data reflects the strong cycling culture in the Netherlands, where the bicycle is a dominant mode for short to medium-range trips.

Figure 2.2 presents the proportion of total trips made by bicycle in various countries during the 2000s. The data shows a considerable variation in cycling mode share across countries. The Netherlands recorded the highest share, with 26% of trips made by bicycle, followed by Denmark (18%) and Germany (10%).

In contrast, countries such as the United States, Australia, Canada, and the United Kingdom reported significantly lower mode shares, ranging from 0.5% to 2%. Intermediate values are observed in countries like France (3%), Austria and Norway (4%), and Belgium (8%).

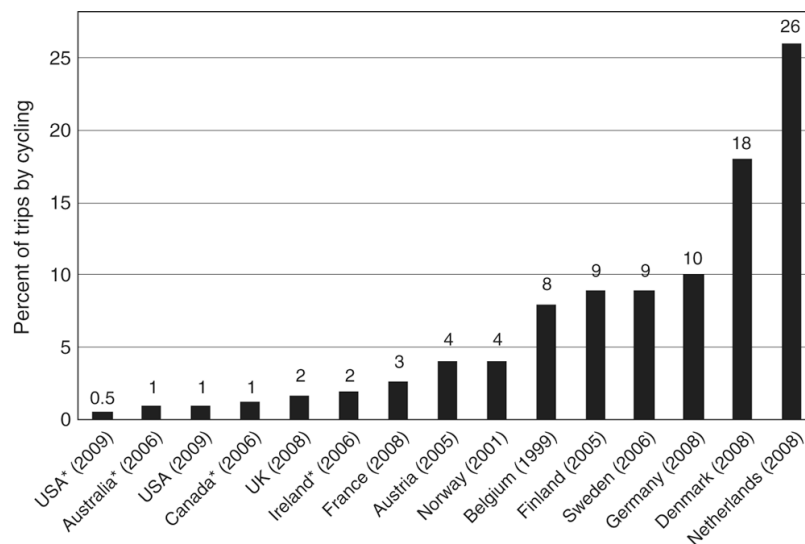


Figure 2.2: Cycling share of daily trips in Europe, North America, and Australia, 1999–2008 [8].

The variation in these figures reflects differences in national transport systems, infrastructure provision, land use planning, and policy emphasis on active mobility. Countries with higher shares generally have dedicated cycling infrastructure, integrated urban networks, and long-standing governmental support for cycling as a mode of transport.

The integration of cycling into daily mobility patterns is influenced not only by road infrastructure, but also by the presence of end-of-trip facilities that support practical and consistent use of bicycles. Figure 2.3 presents the availability of cycling-related facilities offered by employers in the Netherlands from 2020 to 2023. These include changing rooms, showers, lockers for clothing, e-bike charging points, bicycle pumps, and tyre repair facilities.

In 2023, 75% of employers provided showers, and 73% offered changing rooms, reflecting a consistent upward trend since 2020. Lockers for clothing were available at 70% of workplaces, while 60% had e-bike charging stations. Other amenities such as bicycle pumps (55%) and tyre repair facilities (approximately 40%) were also commonly reported. The share of employers with no

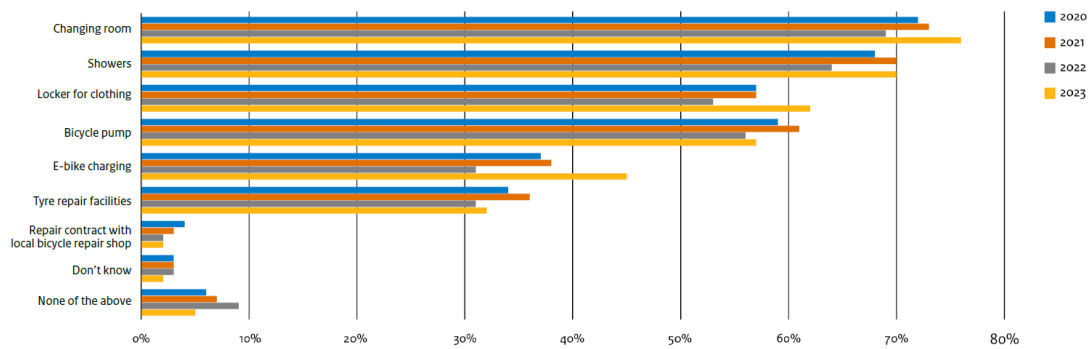


Figure 2.3: Facilities available for workers who travel to work by bicycle by employers in the Netherlands [9].

cycling-related facilities declined to under 5% by 2023, showing a continued institutional support for cycling as a commuting mode.

The Netherlands demonstrates how widespread access to workplace facilities can reinforce a cycling culture that is already embedded in the mobility habits of the population. This high level of provision aligns with the broader transportation framework of the country, where over a quarter of all trips are made by bicycle. In contrast, countries with significantly lower cycling mode shares often lack both comprehensive cycling infrastructure and related investments, such as those illustrated in Figure 2.3. The absence of such measures is frequently cited as a contributing factor to the lower adoption of cycling in daily travel.

These findings illustrate how the availability of supportive infrastructure at the workplace contributes to the overall effectiveness of cycling policies and plays a role in sustaining high levels of cycling in urban and regional transport systems.

The importance of cycling infrastructure can be further illustrated by looking at cities like Copenhagen, Figure 2.4, where cycling is an essential part of daily life. In this city, nearly half of all trips are made by bicycle, thanks to investments in safe and convenient cycling networks.

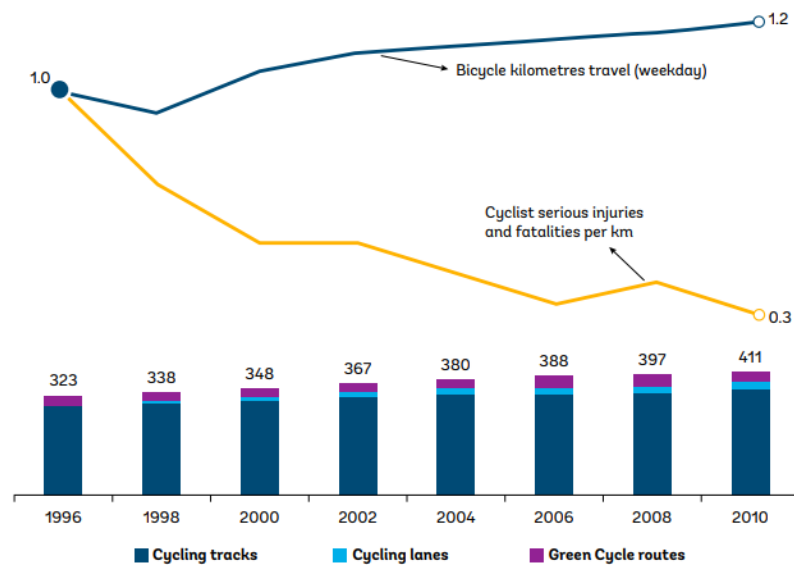


Figure 2.4: Cycling travel, per-kilometer cyclist casualties, and kilometers of cycling infrastructure in Copenhagen [10].

Figure 2.4 also illustrates trends in cycling travel, infrastructure expansion, and safety outcomes in Copenhagen between 1996 and 2010. Over this period, the number of kilometers cycled on weekdays increased by 20%, from an index value of 1.0 in 1996 to 1.2 in 2010. Concurrently, the cyclist casualty rate (serious injuries and fatalities per kilometer travelled) decreased significantly, from 1.0 to 0.3.

This improvement in safety occurred alongside a gradual expansion of cycling infrastructure. The total length of cycling tracks, lanes, and green cycle routes increased from 323 km in 1996 to 411 km in 2010. These developments reflect the continued investment made by the city of Copenhagen in protected and dedicated cycling infrastructure, which separates cyclists from motorized traffic and pedestrians [10].

As reported by *VisitCopenhagen*, the city's broader cycling strategy includes the integration of safe cycling networks, traffic-calming measures, and high-quality bike parking facilities. Copenhagen has been recognized as one of the most bicycle-friendly capitals globally, with nearly half of its residents commuting to work or school by bicycle [11].

2.2 Environmental Impact of Bicycle Manufacturing and Use

Bicycles, particularly traditional pedal-powered models, have a significantly lower environmental impact compared to motorized transport modes such as internal combustion engine vehicles (ICEVs). Battery electric vehicles (BEVs), a rapidly growing alternative to ICEVs, offer significant emission reductions during use but still involve substantial emissions during production. Studies evaluating full life cycle assessments indicate that bicycles generate fewer greenhouse gas (GHG) emissions during production, operation, and end-of-life phases than cars.

The average cumulative distance driven by a car in Europe is estimated to be approximately 15,000 kilometers per year [12]. According to the European Automobile Manufacturers Association (ACEA), the average age of passenger cars in the European Union was 12.3 years in 2023 [13]. National variations are observed: Portugal reports an average vehicle age of 13.6 years, Sweden 10.7 years, and Greece 17.3 years.

Figure 2.5 presents estimates for CO₂ emissions associated with ICEV production and use. The manufacturing phase contributes substantially to overall emissions, primarily due to the extraction and processing of raw materials such as steel and aluminum, and the energy consumption involved in component fabrication and assembly.

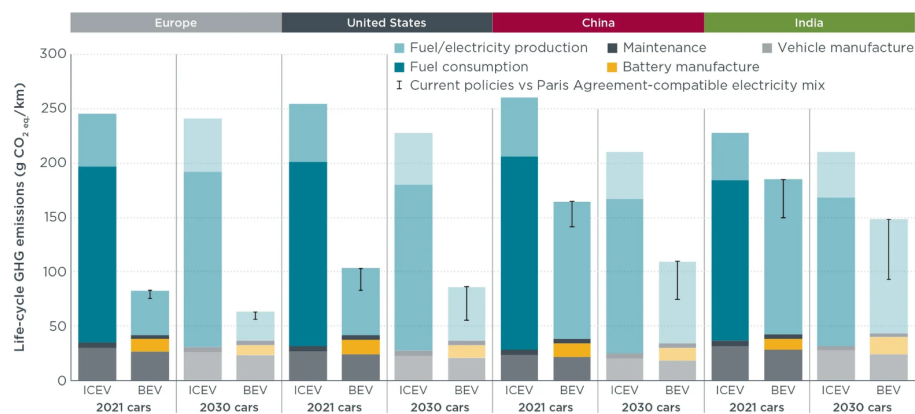


Figure 2.5: Life-cycle GHG emissions of medium-size gasoline ICEVs and BEVs in Europe, the US, China, and India for 2021 and 2030. Error bars represent differences between current electricity mix policies and those needed to meet the Paris Agreement [14].

As illustrated in Figure 2.5, BEVs offer significant life-cycle emission reductions compared to internal combustion engine vehicles (ICEVs), particularly in regions with cleaner electricity grids. However, BEVs still involve considerable greenhouse gas emissions during the manufacturing phase, especially due to electricity production. This reinforces the environmental advantage of bicycles, which not only avoid direct emissions during use but also require far fewer resources over their life cycle, making them a more sustainable mobility solution overall.

Trek Bicycle's 2021 Sustainability Report, a globally recognized bicycle manufacturer, estimates that the production of a typical aluminum bicycle frame results in approximately 174 kg of CO₂ emissions [15]. These emissions are primarily associated with the materials used in the frame and components, such as aluminum, steel, or carbon fiber as can be seen in Figure 2.6. The carbon footprint of bicycle manufacturing is therefore substantially lower than that of cars, representing a notable reduction in production-related emissions.

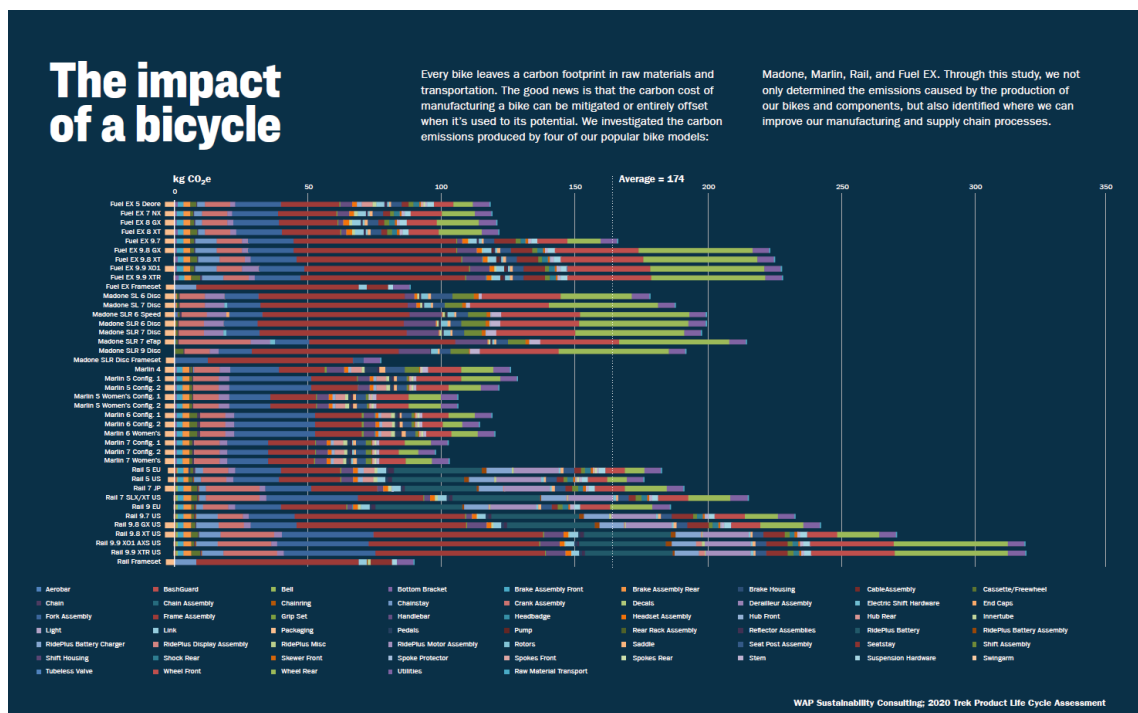


Figure 2.6: Trek Report [15].

In contrast, the manufacturing of an internal combustion engine vehicle (ICEV) typically exceeds 40 tonnes of CO₂ emissions [12, 13, 14]¹. Operational emissions from bicycles are negligible, particularly for conventional models that do not rely on electric assistance.

While bicycles are not suitable for all travel contexts, particularly long-distance or rural journeys, transport policy literature emphasizes the importance of improving public transportation to reduce reliance on private vehicles. Mode shift studies suggest that when public transport offers comparable travel time and accessibility, individuals are more likely to transition away from car usage [10].

Figure 2.7 provides a comparative overview of the carbon footprint of various urban mobility options for journeys up to 30 kilometers. Based on data published by ETH Zurich [16], the figure presents greenhouse gas (GHG) emissions in grams of CO₂-equivalent per passenger-kilometer. The emissions are broken down into key life cycle components: vehicle production (including battery manufacturing and end-of-life processing), road infrastructure, energy supply chain (fuel and electricity), food consumption (relevant for human-powered modes), and use-phase emissions such as exhaust and maintenance. This breakdown enables a comprehensive assessment of the environmental impact of different transport modes.

¹Based on average life-cycle emissions per kilometer and the average distance driven over the lifespan of a vehicle in Europe.

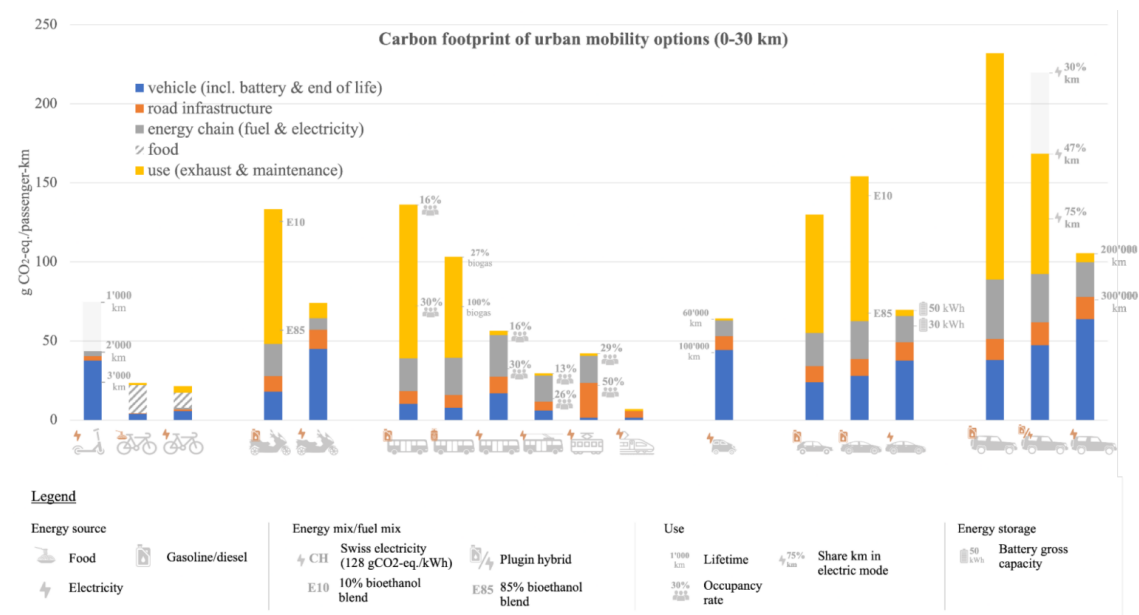


Figure 2.7: Carbon footprint per passenger-kilometer for different means of transport in the urban environment. Underlying input values reflecting average Swiss conditions are used, such as electricity mix, occupancy rate, fuel blend, annual mileage, and more [16].

The figure illustrates that micromobility options, such as bicycles and e-scooters, exhibit the lowest greenhouse gas emissions, typically ranging from 10 to 30 g CO₂-eq per passenger-kilometer. For conventional bicycles, these values account for emissions associated with the energy intake required by the rider. Public transport modes, including buses and trams, demonstrate moderate emission levels that depend on fuel type and average occupancy. Notably, electric buses powered by low-carbon electricity sources, such as the Swiss electricity mix (128 g CO₂-eq/kWh), perform significantly better than those operating on fossil fuels or bioethanol.

Private vehicles represent the category with the highest greenhouse gas emissions, particularly ICEVs with low occupancy and BEVs with short operational lifespans. Battery size and usage assumptions are especially influential in determining the overall environmental footprint of electric vehicles.

Overall, the data highlights the importance of energy source, occupancy rate, and vehicle longevity in shaping the environmental impact of urban mobility. Promoting shifts from private car use to public transport and active mobility, while extending vehicle lifespans and improving occupancy, can significantly reduce emissions.

Among low-emission alternatives, the rise of electric bicycles (e-bikes) reflects a broader move toward sustainable commuting for short and medium-distance trips. Although e-bikes involve higher emissions during the production phase, primarily due to battery manufacturing, they maintain a substantially lower environmental footprint than private cars over the course of their operational lifetime.

While the bicycle industry has already made considerable strides in reducing its carbon footprint, there remain significant opportunities for further environmental improvements during the

manufacturing process. One of the most promising areas for improvement lies in the increased use of recycled materials. For example, the incorporation of recycled aluminum in bike frames can significantly reduce the carbon footprint, as recycled aluminum saves 90-95% of the energy required to refine virgin bauxite ore ². Many bike manufacturers are already exploring partnerships with suppliers to ensure a consistent supply of low-carbon aluminum, helping to lower the environmental impact of bike production.

Another key area of improvement in the bicycle industry is the reduction of plastic waste associated with packaging. Trek has introduced significant modifications to its packaging practices to address this issue. Beginning in 2020, the packaging of its Marlin model was redesigned to reduce the number of non-recyclable plastic components from 24 to 12. During the 2021 production year, this adjustment led to the elimination of over 45 metric tonnes of plastic waste. The revised packaging approach has since been extended to other bike models, and Trek has introduced a 95% recyclable exterior bike box to further reduce the environmental impact associated with transportation [15].

Additionally, there is an increasing trend toward creating zero-waste manufacturing facilities. The goal is to significantly reduce the amount of waste sent to landfills by rethinking production processes and using materials that can be recycled or repurposed at the end of their life cycle. This move toward circularity within the production process not only reduces environmental impact but also aligns with the global shift toward sustainability in manufacturing industries.

These efforts, though still in the early stages for many manufacturers, are critical steps toward making the bicycle industry more sustainable and ensuring that bicycles continue to serve as an environmentally friendly mode of transportation. Ongoing advancements in the bicycle industry are expected to further reduce the carbon footprint associated with manufacturing and improve the overall sustainability of the product lifecycle.

2.3 Health Benefits of Cycling

Cycling offers considerable health benefits that extend beyond individual fitness. Regular physical activity such as cycling plays a key role in the prevention of noncommunicable diseases (NCDs), including cardiovascular disease, type 2 diabetes, and certain cancers. In addition, it contributes to maintaining healthy body weight and supports overall mental well-being.

The World Health Organization (WHO) identifies physical inactivity as a major global health challenge, contributing to an estimated 3.2 million deaths per year. It recommends the promotion of active modes of transport, such as walking and cycling, as effective measures to reduce sedentary behaviour and associated health risks [18].

²Virgin bauxite ore refers to naturally occurring, unprocessed bauxite that is mined directly from the earth. Bauxite is a soil or a rock formation which is composed mainly of aluminum hydroxide minerals. From an industrial perspective, bauxite is a natural material from which alumina can be extracted in a Bayer plant [17].

In the Netherlands, cycling is integrated into everyday life and has measurable effects on population health. National statistics indicate that committed cyclists tend to report better self-assessed health compared to less frequent users. The average cycling distance per person per day in the Netherlands is approximately 3 km, with leisure cycling accounting for the largest share of total distance travelled (48% for conventional bicycles and 53% for e-bikes) [9]. These patterns are closely linked to improved cardiovascular fitness and reduced morbidity.

Evidence also suggests that cycling has a positive impact on mental health. Exposure to outdoor environments, combined with the rhythmic nature of cycling, contributes to stress reduction and improved mood. According to the WHO, active transport modes are associated with increased quality of life and mental resilience [18].

Furthermore, increased cycling reduces dependency on motorized transport, which indirectly benefits public health through improved air quality. Mode shifts from car to bicycle contribute to lower concentrations of particulate matter (PM_{2.5}) in urban areas, which is relevant for reducing respiratory and cardiovascular diseases [10].

In cities where cycling infrastructure is well-developed, such as those analysed in the Institute for Transportation and Development Policy report, investments in protected bike lanes have shown significant reductions in mortality and healthcare costs. A 20-minute daily bicycle commute is associated with a 10% reduction in all-cause mortality risk, underscoring the long-term public health impact of sustained cycling promotion [10].

Taken together, these findings support the inclusion of cycling in health and transport policy as a practical and cost-effective tool to improve population health, reduce healthcare expenditure, and enhance urban liveability.

Chapter 3

State of the Art

The study and evaluation of bicycle frames have progressed considerably in recent decades, driven by growing performance demands, stricter safety regulations, and the increasing availability of advanced computational tools. While early development relied primarily on producing and testing complete physical frames, current practices incorporate digital design validation through 3D printing, simulation, and prototype modeling to assess feasibility at earlier stages.

Despite the shift toward digital workflows, physical frame production and real-world testing remain essential, particularly for verifying assembly processes and evaluating behavior under actual usage conditions. This combined approach enhances both efficiency and realism in the development cycle. This chapter reviews key developments in frame testing practices, with a focus on the role of international safety standards and the application of FEA.

3.1 Introduction

Ensuring the structural integrity of bicycle frames is essential for rider safety and long-term product performance. To evaluate this, standardized testing protocols such as those defined in ISO 4210 have been established to simulate critical loading scenarios and verify compliance with safety requirements.

In parallel with physical testing, FEA has become a valuable tool for predicting structural behavior during the design phase. It allows engineers to analyse stress distributions, deformations, and potential failure points under various conditions, including static loads and impact events. By integrating FEA early in the development process, manufacturers can reduce the number of required prototypes, accelerate iteration cycles, and ensure alignment with regulatory standards before moving to physical validation.

While static analysis is a commonly applied method within the bicycle industry, dynamic analysis is less frequently studied. However, certain studies such as [19] and [20] delve into the modal frequencies of bicycle frames and their comparison with results obtained from FEA.

The study [19] focuses on the dynamic behavior of bicycle frames under various excitation conditions. It emphasizes the importance of understanding natural frequencies and mode shapes

for improving frame design. The results validate the FEA simulations by comparing them with experimental modal testing, showing good correlation and reinforcing the applicability of numerical methods for predicting dynamic performance. The study concludes that modal analysis is essential not only for optimizing structural integrity but also for enhancing rider comfort and safety.

Similarly, [20] presents a comprehensive approach to modal dynamic analysis of bicycle frames, employing experimental methods alongside computational techniques. The authors demonstrate that the identified modal parameters, such as natural frequencies and damping ratios, can be predicted using finite element models. The conclusions highlight that a careful calibration of the model is necessary to achieve accurate alignment with physical tests.

Together, these studies illustrate the potential of dynamic analysis as a critical tool for frame design. Their methodologies, which combine laboratory testing and numerical modeling, represent a best practice approach for verifying simulation accuracy and guiding the iterative design process. Moreover, the research underscores that a deeper insight into modal properties can lead to improved structural dynamics, potentially reducing vibration and enhancing the overall cycling experience. The next Figure 3.1 shows an example of a stress plot with FEA applied on frames.

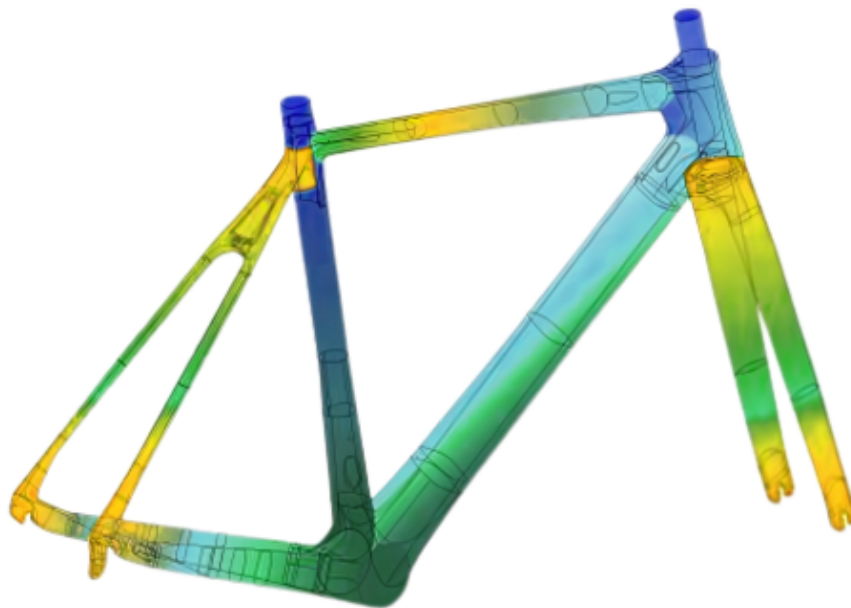


Figure 3.1: Example of FEA on cycling industry - Load Cases Simulated & Tested [21].

Dynamic impact analysis on bike frames, however, has not yet been thoroughly studied or developed using FEA. Addressing dynamic impacts involves complex scenarios such as sudden loads and high-velocity interactions, which require sophisticated modeling approaches. Accurate representation of these scenarios in FEA models is critical, as it must account for transient effects, precise boundary conditions, and detailed interaction definitions. Overcoming these challenges will enable engineers to better predict frame behavior under realistic impact conditions, enhancing safety and performance criteria for bicycle designs.

3.2 Bicycle Safety Standards and Testing Protocols

Standardized testing procedures play a fundamental role in ensuring the structural integrity and safety of bicycle components. Among these, frame testing is particularly critical, as the frame serves as the primary load-bearing structure. Regulatory bodies such as the International Organization for Standardization (ISO) and the European Committee for Standardization (CEN) have developed specific guidelines to evaluate how frames perform under realistic load conditions, impacts, and fatigue cycles. These protocols provide not only a benchmark for manufacturers but also a validated basis for engineering simulations.

Two of the most relevant standards in the context of bicycle frame testing are ISO 4210 and EN 15194. While they are closely related, they are designed to serve different segments of the bicycle industry:

- ISO 4210 is a globally recognized standard that applies to conventional bicycles, including city, trekking, mountain, racing, and young adult bicycles. It specifies mechanical safety and structural durability requirements through various static, fatigue, and impact tests. Of particular relevance to this work is the falling frame impact test, as defined in ISO 4210-6:2023(E) [1]. Due to confidentiality restrictions, detailed reference material and documentation are not included in the appendix of this thesis;
- EN 15194, developed under the framework of the CEN, targets a more specific category: *Electrically Power Assisted Cycles* (EPACs). While it incorporates many structural tests from ISO 4210, EN 15194 adds further requirements related to electrical safety, electromagnetic compatibility, battery behavior, and control system robustness [2].

In summary, ISO 4210 provides the mechanical testing foundation applicable to most types of bicycles, while EN 15194 extends this framework to address the unique needs of electrically assisted cycles. The falling frame test is adopted directly by EN 15194 for the structural assessment of EPACs, further confirming its relevance across both standardization domains.

3.2.1 ISO 4210-6:2023(E) — Falling Frame Impact Test

The ISO 4210-6:2023(E) standard outlines mechanical testing procedures to evaluate the structural strength and durability of bicycles. Among these, the falling frame impact test is particularly relevant to this project, as it simulates a forward fall to assess the behavior of the frame and front fork assembly under impact loading [1].

In this test, the frame is mounted to a rigid support at the rear axle attachment points, allowing it to pivot vertically. The front fork rests on a steel anvil, and three masses are applied to specific locations: the seat-post position (3), the steering head (4), and the bottom bracket (5). The frame is then rotated about the rear axle until it reaches the drop height h_2 , defined in Table 3.1. Upon release, the front end falls freely, striking the anvil. Figure 3.2 illustrates the test setup, with the legend for numbered components provided in the key below.

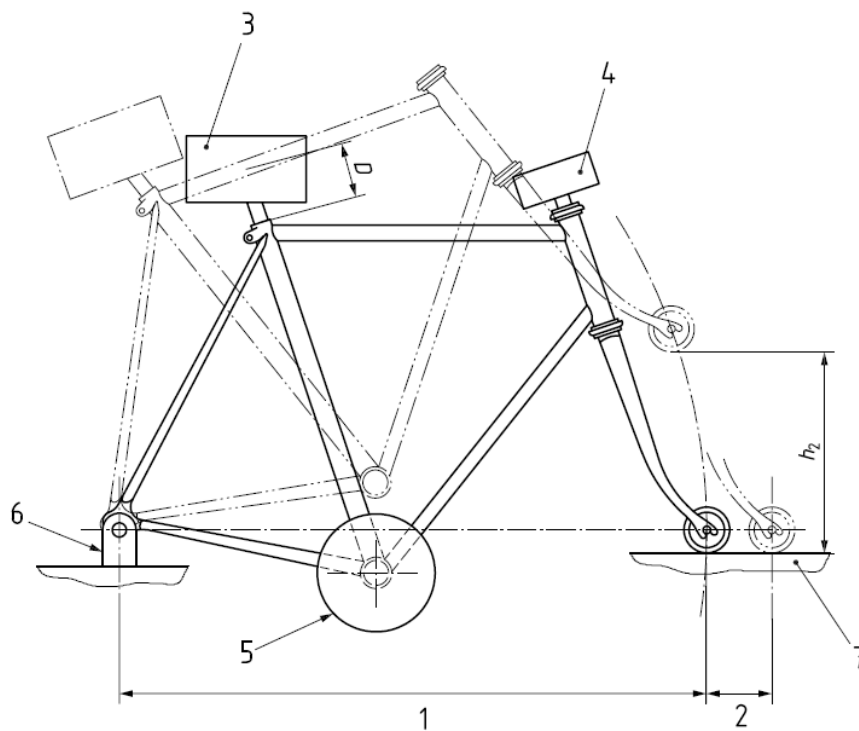


Figure 3.2: Setup of the falling frame impact test showing frame rotation and location of applied masses [1].

Key

1. Wheelbase;
 2. Permanent deformation;
 3. Mass 1 (m_1);
 4. Mass 2 (m_2);
 5. Mass 3 (m_3);
 6. Rigid mounting for rear axle;
 7. Steel anvil;
- D Distance to center of gravity (75 mm);
- h_2 Drop height.

The test is performed twice in succession. After each drop, the wheelbase is measured with the roller in place to determine the permanent deformation of the frame. According to the standard, the total deformation after both drops must not exceed 60 mm [1].

Impact parameters, including mass distribution and drop height for different bicycle categories, are summarized in Table 3.1.

Table 3.1: Drop heights and distribution of masses as per ISO 4210-6:2023(E) [1].

Bicycle Type	m_1 [kg]	m_2 [kg]	m_3 [kg]	Drop Height h_2 [mm]
City and trekking	50	10	30	200
Young adult bicycle	40	10	20	200
Mountain bicycle	30	10	50	300
Racing bicycle	30	10	50	200

The geometric configuration of the frame significantly affects its structural response to impact. As shown in Figure 3.3, high-step frames feature a nearly horizontal top tube that increases rigidity and resistance to deformation. Low-step frames, on the other hand, omit or slope the top tube to facilitate mounting and dismounting, but this design results in lower stiffness.

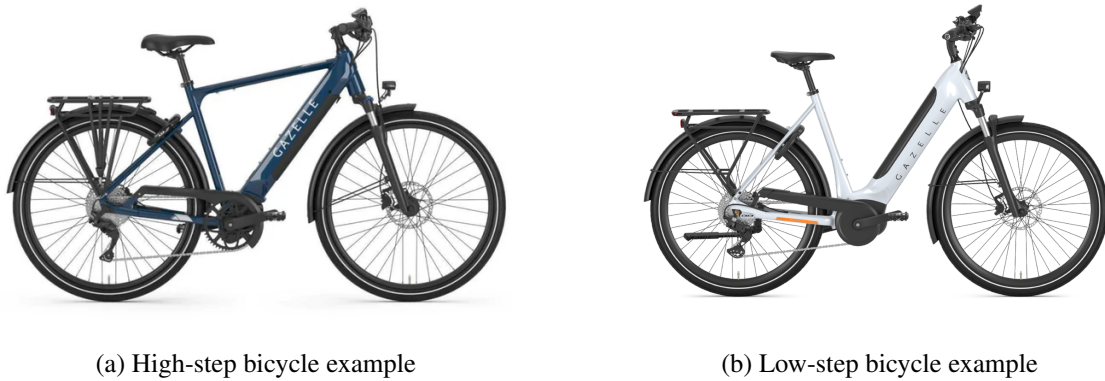


Figure 3.3: Comparison of high- and low-step bicycle frame geometries [22, 23].

To ensure rider safety, the standard mandates that, when both frame variants are available, the low-step version must be used for testing. This configuration is considered more critical due to its increased susceptibility to deformation during impact.

The falling frame impact test replicates front-end collisions or falls that may occur in urban or off-road conditions. By reproducing these loads in a controlled environment, manufacturers can verify that frames meet structural integrity requirements and remain within permissible deformation limits.

3.2.1.1 Gazelle Frame Falling Test: Internal Approach Beyond ISO Standards

Although ISO standards establish a baseline for bicycle safety and performance, real-world conditions often subject frames to more demanding loads than those defined in standardized tests. Drawing from years of in-house physical testing and field data, Gazelle has identified limitations in relying solely on ISO criteria.

To address this, Gazelle has developed internal testing procedures that extend beyond ISO minimums. These include more conservative test configurations designed to better account for rider variability, environmental factors, and long-term durability. While the exact values are proprietary, the guiding principle is to more closely replicate actual usage conditions and minimize the risk of structural failure during the product's service life.

This internal approach reflects a data-driven, safety-oriented philosophy and plays a central role in the validation and release of new frame designs.

3.3 Application of Finite Element Analysis in Bicycle Engineering

In recent decades, FEA has become an indispensable tool in mechanical and structural engineering, and its integration into the bicycle industry is steadily gaining traction. Originally developed for complex problems in aerospace and civil engineering, FEA is now widely used to simulate the mechanical behavior of consumer products, including sports equipment such as bicycles [24].

The demand for lighter yet structurally robust bicycle frames, particularly with the increasing use of advanced materials such as carbon fiber and aluminum alloys, has led manufacturers to adopt virtual testing environments. Although still maturing within the bicycle sector, FEA has already demonstrated its potential through a number of studies, some of which are presented in this section. The method provides an efficient way to simulate the performance of a frame under realistic loading conditions, enabling the evaluation of key parameters such as stiffness, stress distribution, deformation, and fatigue life, often before any physical prototypes are built.

One of the earliest documented applications of FEA in the bicycle industry was by Peterson and Londry [25], who applied the method to refine the design of the Trek 2000 aluminum frame. Their model used beam elements to represent the tubular structure of the frame (excluding the fork) and simulated various load cases at critical locations including the seat tube, head tube, brake bridge, and bottom bracket. They compared the Trek frame against a steel frame and another aluminum frame, evaluating mass, stiffness, and strain energy distribution.

The following Figure 3.4 illustrates the simplified finite element model (FEM) used in their study, with restraints placed at the rear axle and head tube:

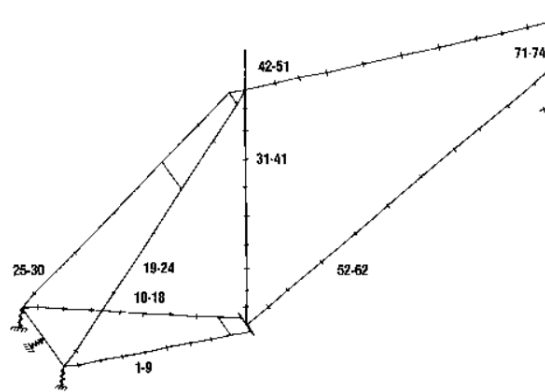


Figure 3.4: FEM subdividing the frame into approximately 100 beam elements. Boundary conditions are applied at the rear axle and head tube [25].

Their results revealed that the down tube consistently absorbed the greatest portion of strain energy, ranging from 38% to 49%, followed by the seat tube with values between 19% and 25%. This insight enabled frame designers to better allocate material for an optimal balance between weight and stiffness. Importantly, they also observed that increasing vertical compliance had minimal negative impact on climbing performance, highlighting how FEA could be used to optimize both comfort and efficiency.

Expanding on this foundation, Covill et al. [6] conducted a comprehensive parametric study to investigate the influence of geometry on the mechanical behavior of bicycle frames. Their research evaluated 82 road frame geometries using FEA with both beam and shell elements, focusing on vertical compliance and lateral stiffness. The study applied two load cases representing vertical rider weight and lateral pedaling forces, allowing consistent comparison across designs.

The simulation setup and example displacement plots are shown below in Figure 3.5.

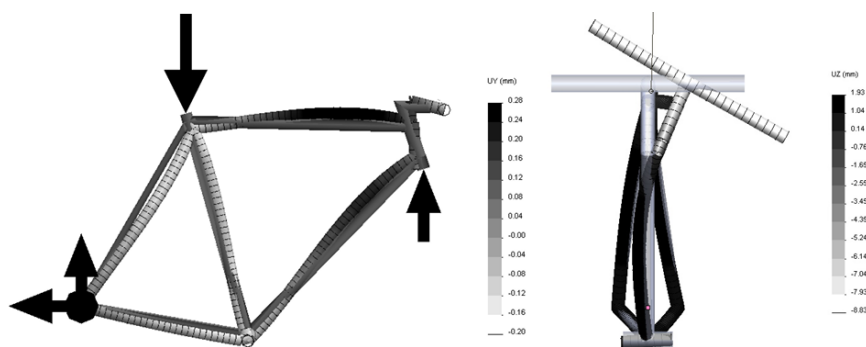


Figure 3.5: Example displacement results: vertical load at the seat tube (left) and lateral loads applied at the bottom bracket and handlebars (right). Displacements are scaled for visualization [6].

Key findings from their study included clear correlations between frame geometry and mechanical performance. Frames with shorter top tubes and steeper head tube angles exhibited higher

lateral stiffness, beneficial for sprinting and handling. Conversely, longer seat stays were associated with greater vertical compliance, improving comfort on rough terrain.

In addition to metal frames, FEA has been effectively applied to composite bicycle frames, particularly those made of carbon fiber reinforced polymers (CFRPs), to leverage their high strength-to-weight ratio and directional tailoring capabilities. Lessard et al. [26] conducted one of the earliest and most comprehensive studies using FEA to evaluate the structural behavior of composite frames. Their work combined both beam and shell elements and included experimental validation, examining vertical compliance and lateral stiffness. The study demonstrated how different layup strategies and material properties could influence frame deformation under load, offering insight into the relationship between frame geometry, laminate configuration, and ride characteristics.

Further advancing this area, Liu and Wu [4] focused specifically on the effect of fiber direction and stacking sequence in carbon/epoxy composite laminates for bicycle frames. Using shell element models subjected to frontal, torsional, and vertical loads, they analysed stress distributions and optimized fiber orientations to minimize peak stresses and enhance structural integrity. Their simulations revealed that strategic variation of the fiber angles in regions with high stress, particularly around the head tube and bottom bracket, could significantly reduce localized stress concentrations and improve overall durability. These findings illustrate the value of FEA not only for shape optimization but also for micromechanical design of composite materials, enabling frame configurations that are both lighter and more resilient.

In commercial applications, FEA is now routinely employed to validate frame designs against both international safety standards and internal requirements defined by companies. Most simulations typically address static loading scenarios, such as vertical forces resulting from rider weight or lateral loads generated during pedaling. However, some manufacturers have begun to extend analyses to include dynamic and nonlinear events, such as impacts and fatigue-related failures. These advanced simulations are inherently more complex, often requiring transient, nonlinear approaches that incorporate contact interactions and material plasticity. Best practices for modeling these conditions are detailed in the comprehensive guide on simulation methodologies by Madier [24].

3.4 Experimental Validation and Correlation with FEA

Experimental validation plays a pivotal role in the FEA process. Numerical models, while powerful, must be rigorously compared with real-world data to ensure predictive reliability and physical accuracy. In this project, validation involves replicating standardized impact test conditions and comparing key outputs, such as stress distributions, permanent deformations, and crack initiation, with results from physical testing. To support this comparison, non-destructive inspection (NDI) techniques and motion analysis tools are used to capture deformation and failure behavior in the actual frame.

Despite the increasing use of FEA in bicycle frame design, simulation results must be interpreted with caution unless supported by experimental evidence. Madier outlines three key validation steps to ensure simulation credibility: accuracy checks, mathematical validity checks, and final correlation with physical test data [24]. Only after all three steps are satisfied can simulations be trusted for decision making in safety-critical applications.

Furthermore, Alves highlights that under dynamic impact conditions, it is essential to model strain rate sensitivity and failure behavior accurately. Correlation with experimental results, such as force-displacement curves and post-impact deformation patterns, is necessary to prevent false confidence in numerically stable but physically inaccurate models [7].

By applying controlled loading scenarios to both digital and physical models, this project seeks to assess simulation accuracy and identify potential limitations in material modeling, particularly in relation to strain rate and temperature sensitivity of aluminium 6061-T6.

3.4.1 Techniques for Correlating Simulation and Test Data

Several experimental methods are used to validate FEA results in the context of bicycle frame analysis. A commonly employed approach involves bonding strain gauges to critical areas of the frame, such as the head tube, bottom bracket, or down tube, during impact or fatigue tests. The recorded strain histories are then compared to simulation outputs at corresponding node paths. For example, Arola et al. performed dynamic drop tests and compared peak stress values measured by multiple rosettes, showing that front tire impacts generated stresses exceeding 400 MPa in the top tube, which informed the calibration of FEA models [3].

For full-field deformation analysis, Digital Image Correlation (DIC) has become increasingly adopted. This optical technique tracks the displacement of speckle patterns on the frame surface, producing detailed strain maps. Alves demonstrated the effectiveness of DIC in identifying high-strain regions and verifying zones of plastic deformation, particularly in the context of crack formation and stress localization [7]. The non-contact nature of DIC enables it to capture data during dynamic impact events, where bonded sensors may detach or fail due to overload.

To detect cracks, EN 15194 recommends standardized visual and NDI techniques. These include dye penetrant inspection, as outlined in EN ISO 3452-1 through 3452-4, as well as auxiliary treatments such as white paint or reflective coatings to enhance crack visibility under cyclic loading [2]. These visual inspections are typically used post-test to verify whether damage predicted by the simulation, such as stress levels exceeding material limits, corresponds to observable crack initiation.

The study by Manolova et al. [27] further emphasizes the use of strain gauge instrumentation as a non-destructive method to evaluate frame behavior under realistic loading. In their investigation, a BMX frame was instrumented and tested under static and dynamic pedaling conditions. The placement of strain gauges was guided by prior FEA results, targeting high-stress areas including the seat tube and the right chain stay near the bottom bracket.

Their results showed a maximum stress of approximately 41 MPa, well below the yield strength (YS) of the frame material (460 MPa). Moreover, the consistency between pedal forces and measured strain supported the method's applicability in validating and calibrating FEA models.

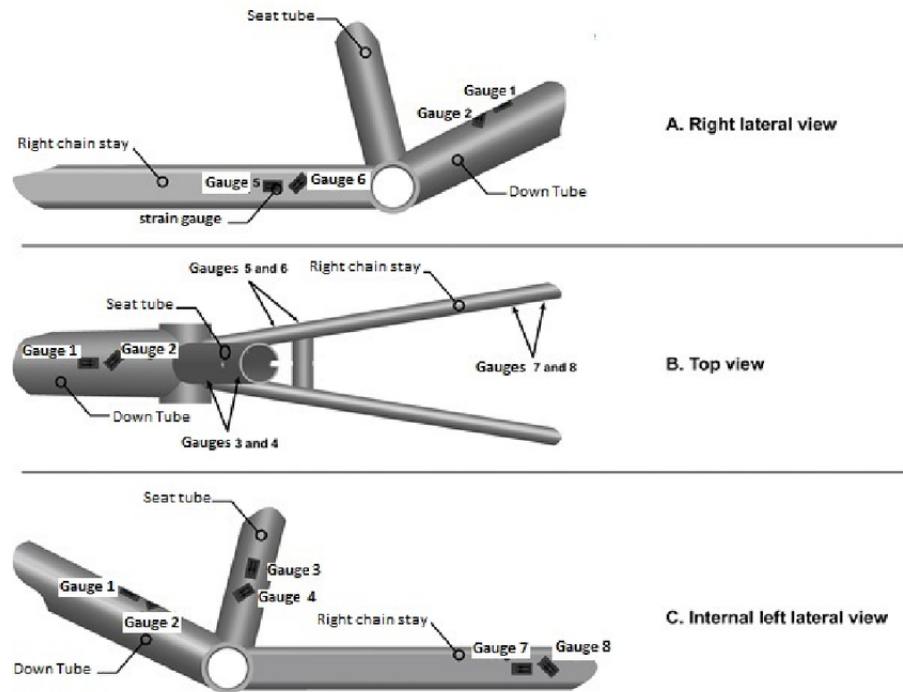


Figure 3.6: Strain gauge locations on the BMX frame [27].

To quantify the correlation between simulation and experimental data, metrics such as Root Mean Square Error (RMSE), relative percentage error, and normalized strain energy differences are commonly used. Covill et al. emphasized the importance of accurately matching both the location and magnitude of stress concentrations, especially at joint transitions and butted tube regions, which are typically more susceptible to crack initiation under dynamic loads [6].

In cases involving failure prediction, techniques such as cohesive zone models or extended finite element methods (XFEM) can be used to simulate crack initiation and propagation. These simulations can then be compared with observed fracture surfaces or crack growth markers in post-test inspections. Although limited literature exists for the bicycle industry, established practices from aerospace and automotive fields provide transferable methodologies [7].

Experimental validation remains a fundamental part of the FEA process in bicycle development. Techniques such as strain gauging, digital image correlation, and crack inspection provide essential data for assessing the accuracy of numerical predictions. Their integration supports the use of simulation in certification workflows and the implementation of first-time-right design strategies.

3.5 Materials in the Bicycle Industry

Material selection is a critical factor in bicycle frame design, directly influencing performance, durability, weight, and cost. This section provides an overview of the most commonly used materials in the bicycle industry, with a particular focus on Al 6061-T6. Its properties, advantages, and industrial relevance are discussed and compared to other popular alternatives.

3.5.1 Material Property Comparison

A comparison of commonly used frame materials, including Al 6061 (wrought, 6000 series), alpha-beta titanium alloys, high-alloy steels, and carbon fiber composites, is presented in terms of yield strength, specific stiffness, and relative cost. The data used in this analysis were extracted from the EduGranta Pack software suite, which provides comprehensive material property datasets across multiple domains.

Figures 3.7, 3.8, and 3.9 illustrate these comparisons.

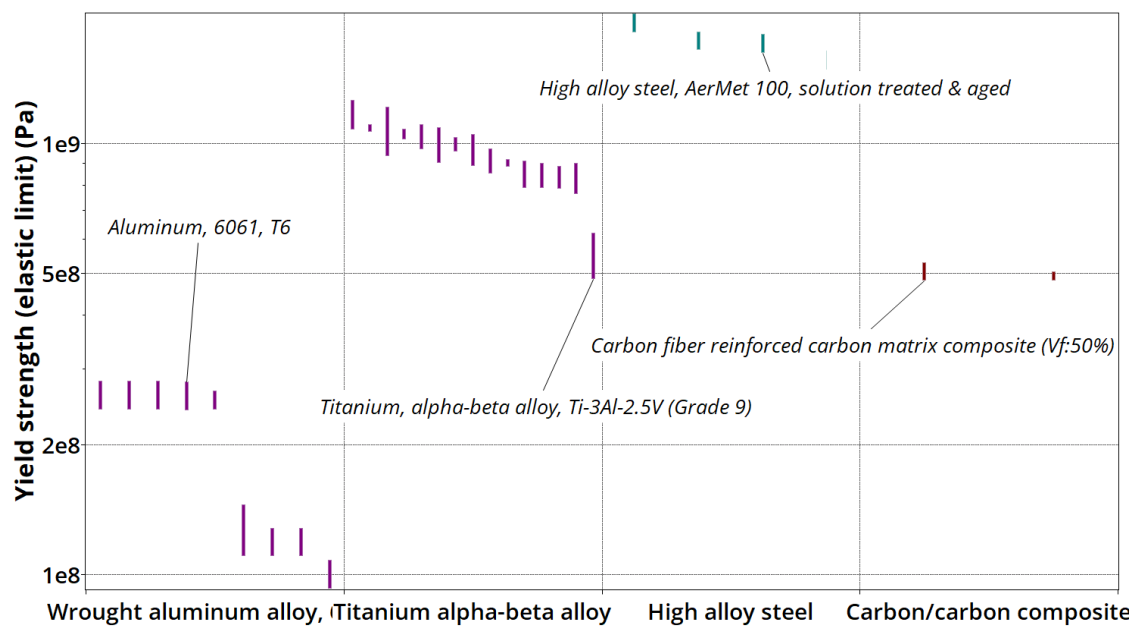


Figure 3.7: Yield strength comparison of bicycle frame materials.

High-alloy steels, such as AerMet 100, exhibit the highest YS, followed by titanium alloys and carbon fiber composites. Al 6061-T6 offers moderate yield strength, around 270 MPa, sufficient for many structural applications in bicycle frames.

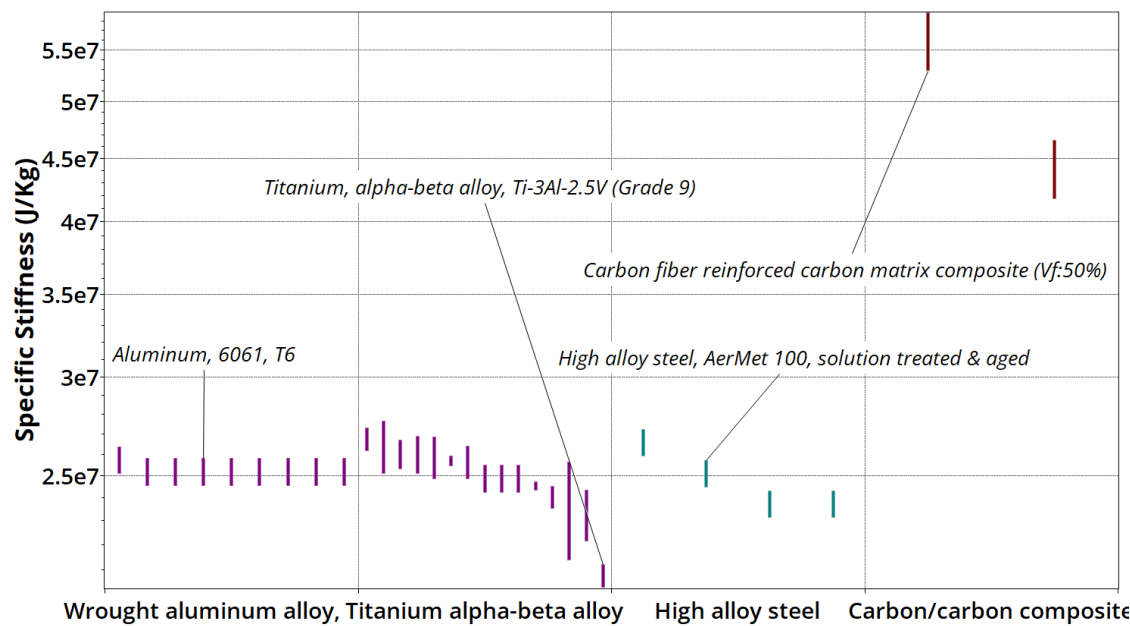


Figure 3.8: Specific stiffness (stiffness-to-density ratio) comparison.

Carbon fiber composites demonstrate the highest specific stiffness, followed by titanium alloys. Al 6061-T6 reaches approximately 2.5×10^7 J/kg, outperforming titanium alloys in this category.

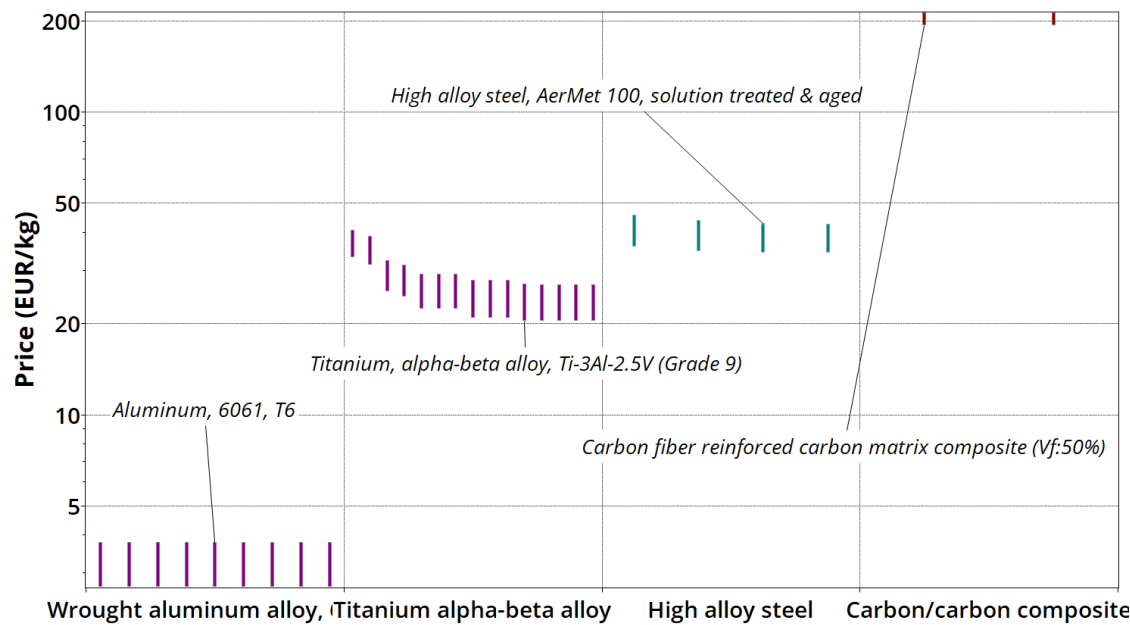


Figure 3.9: Relative material cost comparison (EUR/kg).

Carbon fiber and titanium are significantly more expensive, exceeding 150 EUR/kg and 20 EUR/kg, respectively. Al 6061-T6 is notably more affordable, with typical costs below 3 EUR/kg, making it an attractive option for performance-to-cost optimization.

3.5.2 Aluminum 6061-T6 Mechanical Properties

Although alternative materials may offer advantages in specific performance metrics, Al 6061-T6 remains a preferred material in the cycling industry due to its balance of mechanical properties, low density, manufacturability, and affordability. Its compatibility with established production techniques, including TIG welding, extrusion, and hydroforming, further supports its widespread industrial use [28].

3.5.2.1 T6 Heat Treatment Process

The T6 heat treatment enhances the mechanical strength of 6061 aluminum through precipitation hardening. This process typically involves three main stages: solution treatment, quenching, and artificial aging.

Figure 3.10 provides a schematic overview of this procedure.

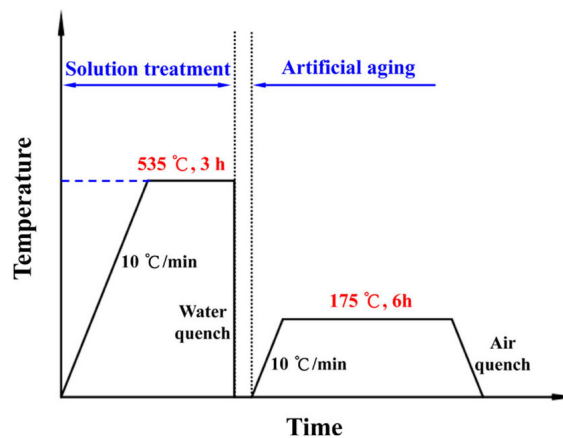


Figure 3.10: Schematic illustration of the T6 heat treatment process [29].

In this process, the alloy is first solution-treated at approximately 535 °C for 3 hours, then rapidly quenched in water to lock solute atoms in place. This is followed by artificial aging at 175 °C for 6 hours, promoting the formation of finely dispersed Mg_2Si precipitates. These precipitates increase YS and hardness, making the alloy suitable for high-performance applications [30].

The mechanical enhancements achieved through T6 treatment are reflected in the key properties of the material. These are particularly relevant for structural and impact-loading scenarios, such as those encountered in bicycle frame applications.

Table 3.2: Key Mechanical Properties of Al 6061-T6 [31].

Property	Value Range	Unit
Density	2690 – 2730	kg/m ³
Young's modulus	66.6 – 70.0	GPa
Yield strength (elastic limit)	240 – 280	MPa
Tensile strength	290 – 338	MPa
Elongation at break	10.0 – 14.4	%

The values presented in Table 3.2 refer to room temperature conditions (approximately 23 °C). Temperature-dependent variations, such as modulus and strength reductions at elevated temperatures, are not included here and are omitted from this impact-focused study for simplicity.

3.5.3 Plasticity Modeling of Aluminium 6061-T6 Alloy

Aluminum 6061-T6 has been widely studied under dynamic loading conditions due to its broad use in transportation and structural applications. Under high strain rates and potential thermal influences, such as those encountered during impact or drop scenarios, the mechanical behavior of the alloy diverges significantly from the response observed under quasi-static conditions.

Experimental studies using Split Hopkinson Pressure Bar (SHPB) testing have shown that both yield strength and ultimate tensile strength (UTS) increase with strain rate, while the work hardening rate of the material decreases as plastic deformation progresses [32]. These effects are associated with thermomechanical phenomena such as thermal softening and adiabatic shear localization, which tend to arise when strain rates exceed approximately 10^3 s^{-1} [33]. At the microstructural level, high-rate deformation may induce dislocation cell structures, grain boundary fragmentation, and, in some cases, dynamic recrystallization resulting from localized temperature rise due to plastic work.

To model these coupled effects in simulations, many studies adopt the Johnson–Cook (JC) constitutive model. This empirical model captures strain hardening, strain rate sensitivity, and thermal softening through a multiplicative formulation:

$$\sigma = (A + B\epsilon^n) \left[1 + C \ln \left(\frac{\dot{\epsilon}}{\dot{\epsilon}_0} \right) \right] \left[1 - \left(\frac{T - T_{\text{room}}}{T_{\text{melt}} - T_{\text{room}}} \right)^m \right] \quad (3.1)$$

In this formulation, ϵ is the equivalent plastic strain, which describes the accumulated irreversible deformation, and n is the strain hardening exponent that governs how the material strengthens as it deforms plastically. In literature, a range of JC parameters for Al 6061-T6 have been reported. For example, Acharya et al. [32] and Scapin et al. [33] identified yield stress values (A) between 280 and 340 MPa, hardening modulus (B) from 120 to 184 MPa, and strain rate sensitivity coefficients (C) ranging from 0.002 to 0.04. The thermal softening exponent (m) typically varies between 1.3 and 2.2, depending on the test conditions, with a melting temperature around 582 °C.

The JC model is commonly used in simulations of high-speed deformation events such as ballistic impacts, automotive crashes, or aerospace collisions, where rate- and temperature-dependent effects significantly influence material response. Its compatibility with commercial finite element software also supports its adoption in industrial and research applications.

3.6 Summary

This chapter reviewed the evolution of bicycle frame testing, emphasizing the transition from traditional physical validation to integrated simulation-based methodologies. The role of international standards, particularly ISO 4210 and EN 15194, was discussed as a foundation for impact testing procedures. FEA was highlighted as a powerful tool in the development and optimization of frame designs, supported by case studies that demonstrate its effectiveness in assessing structural behavior under static and dynamic loads.

Relevant material selection criteria were also addressed, with a focus on Al 6061-T6 due to its balance of mechanical performance, manufacturability, and cost-efficiency. Its specific mechanical properties and industrial processing steps, including T6 heat treatment, were presented in the context of its suitability for frame applications. Although this project does not explicitly model thermal or strain rate sensitivity, literature on high-speed deformation and plasticity modeling was reviewed to establish a solid theoretical basis for interpreting impact simulation results.

By connecting the principles of material behavior, standard test methods, and numerical modeling approaches, this chapter provides the necessary scientific context to support the development and validation of the frame falling impact simulation performed in this study.

Chapter 4

Methodology

The frame selected for this project was the *Gazelle Eclipse* model. This choice was motivated by the existence of previous frame falling tests conducted on this specific frame, during which engineers observed crack initiation in critical regions, such as the bottom bracket area and the vicinity of the battery housing. This prior empirical knowledge provided valuable insight into likely failure zones, allowing for a more targeted assessment of the simulation's accuracy by comparing predicted stress and plastic strain distributions with known weak points. Although this does not constitute formal experimental validation, it served as an important qualitative benchmark for verifying the simulation setup, including material properties, boundary conditions, and applied loads.

The frame geometry was originally developed in SolidWorks, allowing for a seamless import into the 3DEXPERIENCE platform. This environment enables efficient integration of SolidWorks assemblies into the Mechanical Scenario Creation module, facilitating rapid preparation of the FEM and simulation setup.

To replicate the physical test conditions and comply with relevant ISO standards, the digital model was completed with additional components such as a dummy fork, bottom bracket shell, and seatpost insert. These parts were precisely positioned using the *Mate* feature in SolidWorks to ensure geometric accuracy consistent with the real test configuration, as shown in Figure 4.1.

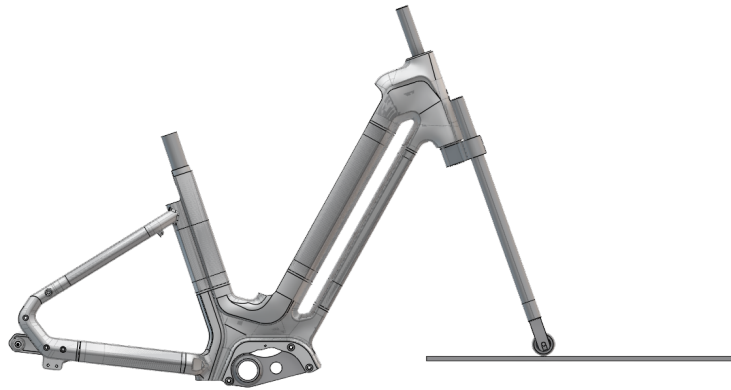


Figure 4.1: Test setup in 3DEXPERIENCE.

4.1 Bicycle Geometry

Before presenting the FEM and associated material properties in detail, it is important to outline the simplifications and assumptions made about the actual physical frame.

A typical bicycle frame consists of both extruded and cast components. This also applies to the *Gazelle Eclipse* frame. The main structural tubes include the top tube, down tube, seat tube, chain stays, and seat stays, which are extruded parts made from wrought Al 6061-T6. Cast components such as the motor mount, head tube, and dropout areas are made from aluminum alloy A356. These parts are assembled using welded joints.

The primary structural regions of the frame are labeled in Figure 4.2.

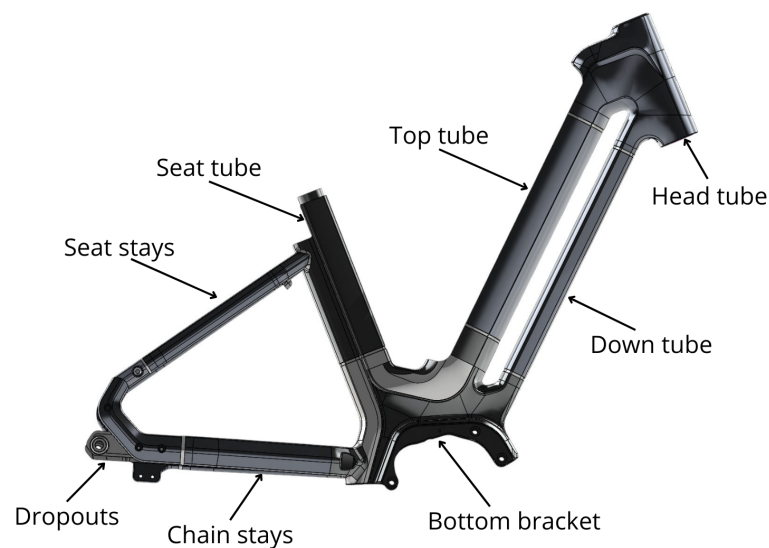


Figure 4.2: Main structural parts of a bicycle frame.

4.1.1 Simplifications

When welding similar materials, such as Al 6061-T6 extrusions, Tungsten Inert Gas (TIG) welding is typically used with filler materials like ER4043 or ER4045. These fillers, which contain increased silicon content, offer better fluidity and reduce the risk of hot cracking. However, the weld zone, particularly the heat-affected zone (HAZ), usually experiences a reduction in mechanical strength compared to the base metal. This degradation is caused by temper softening and microstructural changes introduced during the welding thermal cycle [28].

Welding dissimilar materials, such as an extruded Al 6061-T6 tube to a cast A356 part, presents additional challenges. Differences in melting points, chemical composition, and thermal expansion behavior can lead to the formation of brittle intermetallic compounds and residual stresses at the weld interface. Proper filler selection and process control are required to minimize these effects.

Microstructural variations in the weld and HAZ can significantly affect joint performance. Even frames built from identical materials and designs may exhibit different mechanical behavior depending on manufacturing tolerances or welding quality. Defects such as porosity, incomplete fusion, or microcracks can have a substantial influence on fatigue life and crack initiation behavior [28, 33].

For the purposes of this project, these local effects in the weld and HAZ are simplified and not explicitly modeled in the finite element analysis. Previous physical tests performed on the *Gazelle Eclipse* frame did not show failure in the welds between extruded and cast components. Based on these observations, the exclusion of weld defects and localized heat-affected properties is justified for the current simulation objectives.

To account for the relative mechanical weakness in the weld regions, a reduced wall thickness was applied in specific areas of the model. This approach anticipates stress and strain concentrations due to material discontinuities, as illustrated in Figures 4.3a and 4.3b. However, weld geometry and localized defects are not explicitly represented. Since prior experimental tests did not indicate failure in these zones under the tested conditions, the simplified representation of welds is considered acceptable for this simulation.

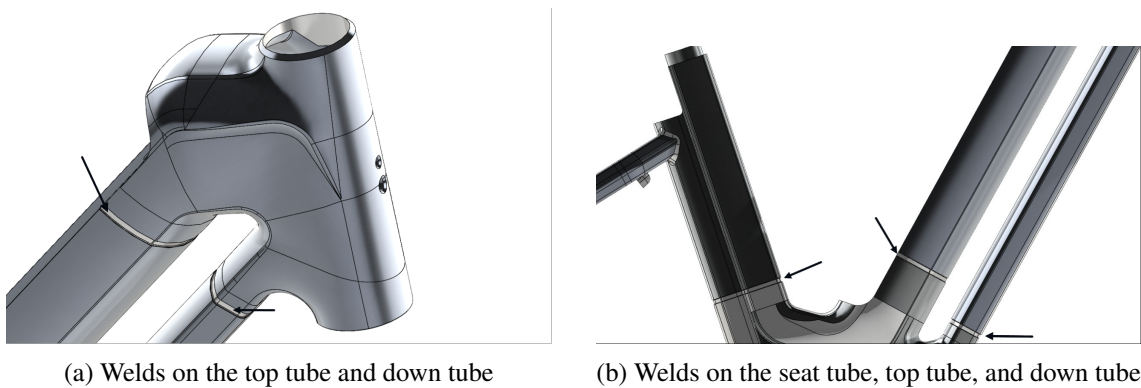


Figure 4.3: Highlighted weld regions at critical frame junctions.

4.2 Finite Element Model

This section presents the complete FEM used for the falling frame impact simulation. In addition to the frame itself, all supplementary components required to replicate the physical test setup are included. Each part is labeled according to the naming conventions used throughout this report to ensure consistency and clarity.

Figure 4.4 illustrates the assembled model with its key components and structural elements necessary for the simulation.

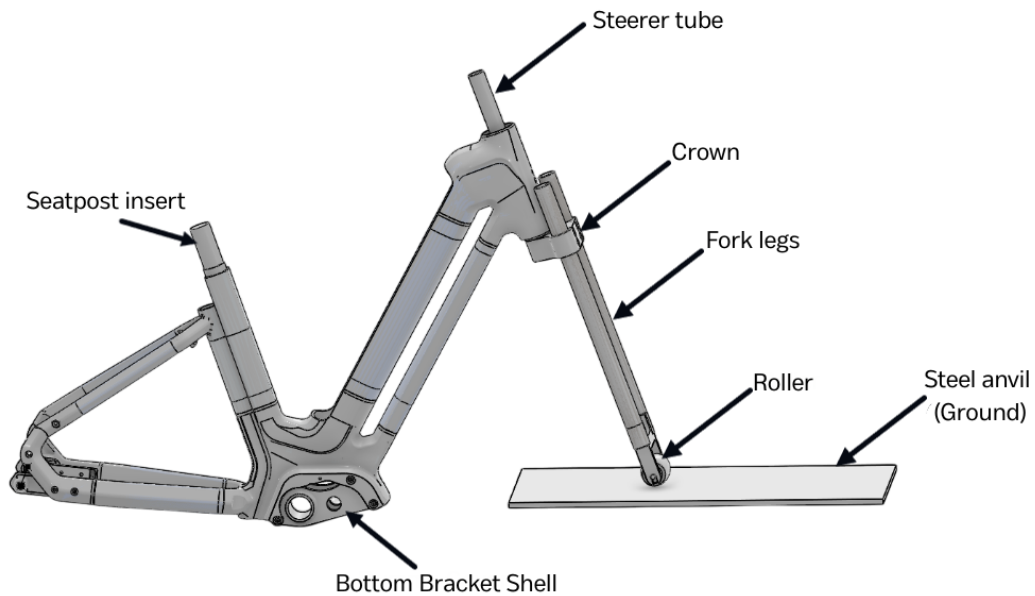


Figure 4.4: Overview of the complete FEM, including the frame and all supporting components used for the impact simulation.

4.2.1 Fork Specifications

The fork geometry adopted in this study was defined according to the technical drawing shown in Figure 4.5, which was provided by Gazelle as internal documentation [34]. The assembly includes the roller, fork legs, crown, and steerer tube, each made from different metallic materials.

The materials used are designated by their German standards: St 37-K, AlCuMgPb F38, and St 52-3. According to EN 10025-2, these grades correspond approximately to S235JR and S355JR structural steels. The fork crown is made from a 2xxx-series aluminum alloy, with copper as the principal alloying element. For the purpose of material property assignment in simulation, the alloy Al2024-T6 is used as a representative approximation due to its availability in the *EduPack* materials database [31].

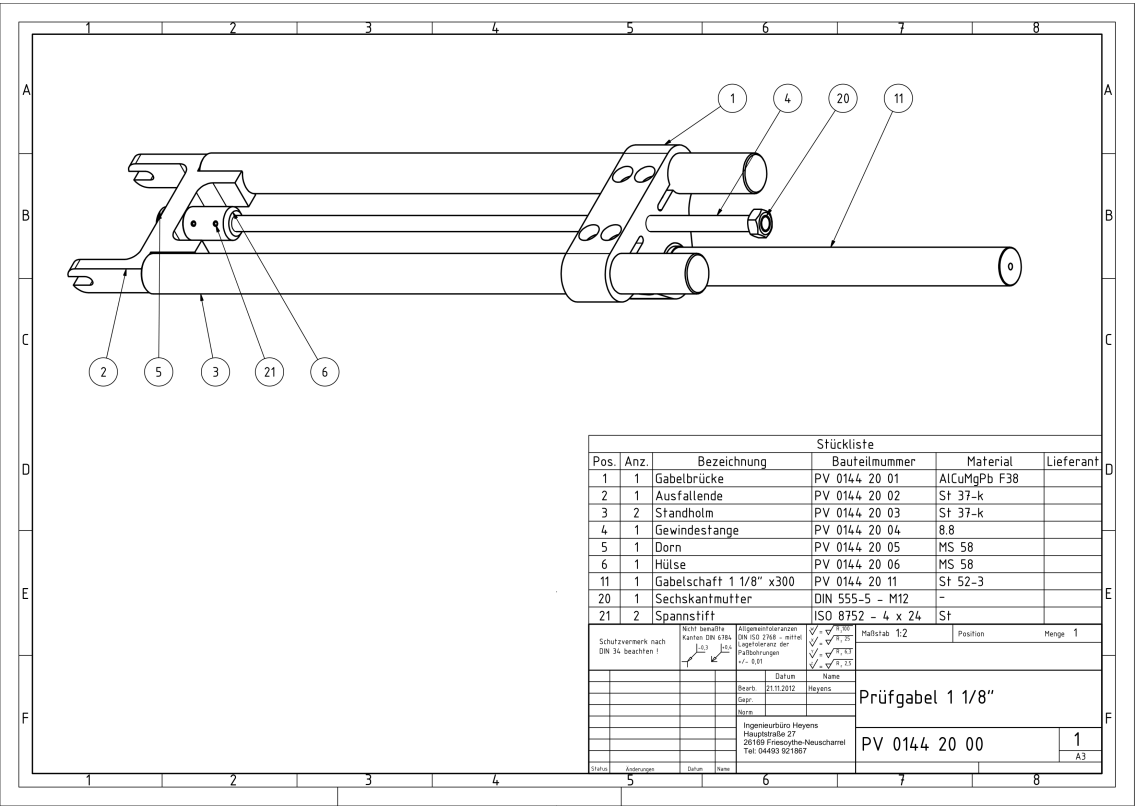


Figure 4.5: Technical drawing of the fork used in the analysis [34].

To reduce model complexity and simulation time, a simplified fork referred to as a dummy fork was implemented. This approach is aligned with the provisions of ISO 4210-6:2023, which allow the use of simplified fork geometries for standardized frame impact simulations.

The dimensional and functional requirements for the dummy fork are outlined in ISO 4210-6:2023, Annex B [1].

In this project, the same dummy fork design was employed for both the numerical simulation and the experimental validation to ensure consistency between the digital and physical setups. The physical fork used in testing is shown in Figure 4.6.



Figure 4.6: Dummy fork used during experimental validation.

Furthermore, in accordance with ISO 4210-2:2023, Section 4.9.6, the frame can be tested with a fork that has previously passed the fork impact test. This ensures that the dummy fork provides a reliable and standardized interface for evaluating frame impact resistance [35].

4.3 Material Properties and Plasticity Model

The material properties used in the finite element simulations were derived from a combination of literature sources and an internal tensile testing campaign performed on a representative Al 6061-T6 bicycle frame. The physical and mechanical properties, such as density, Young's modulus, and yield strength, are consistent with the values reported in Chapter 3. In this section, the justification for the adopted plasticity model and simulation parameters is presented, along with the results of the in-house tests that informed them.

Strain rate describes how quickly a material deforms under load and is typically measured in reciprocal seconds (s^{-1}). It represents the rate of change of strain over time and is directly influenced by the impact velocity and the geometry of the structure. In other words, it quantifies how fast the shape of an object changes during loading [7].

However, the loading conditions in the frame impact scenario simulated in this work correspond to a lower strain-rate regime, where adiabatic heating and strain-rate effects are considerably less pronounced. Therefore, the thermal softening and strain-rate sensitivity terms in the constitutive model were excluded, simplifying the material behavior [33].

4.3.1 Plasticity Model Selection

The Johnson–Cook (JC) constitutive law was selected due to its computational efficiency and widespread application in impact simulations involving metallic materials [33]. The model accounts for strain hardening, strain-rate sensitivity, and thermal softening through a multiplicative expression:

$$\sigma = [A + B(\epsilon_p)^n] \left[1 + C \ln \left(\frac{\dot{\epsilon}_p}{\dot{\epsilon}_0} \right) \right] \left[1 - \left(\frac{T - T_r}{T_m - T_r} \right)^m \right] \quad (4.1)$$

However, under the specific conditions of this project, a reduced form of the JC model was adopted.

4.3.1.1 Adapted Johnson–Cook Model for Frame Impact Simulation

Based on the relatively low impact velocity and short contact duration in the test, the temperature rise due to plastic work is negligible, and the strain rate is below the threshold at which rate effects become dominant. Therefore, both the strain-rate sensitivity and thermal softening terms were omitted, yielding the simplified expression:

$$\sigma = A + B\epsilon_p^n \quad (4.2)$$

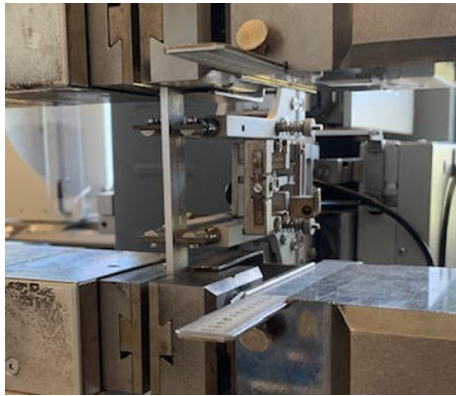
This reduced model captures the strain hardening behavior, which remains the primary non-linearity under the impact conditions considered.

4.3.2 Experimental Tensile Testing

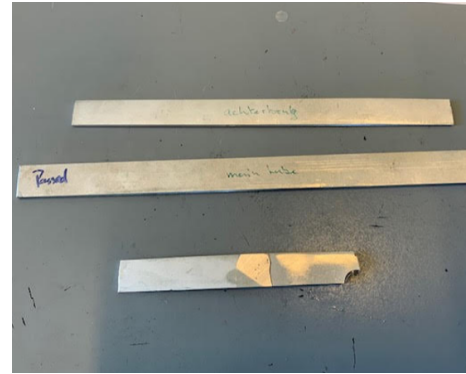
To verify the material parameters and ensure consistency with real components, a series of uniaxial tensile tests was performed on an Al 6061-T6 frame provided by Gazelle. Although the frame tested was not the one modeled in simulation, it produced on the same supplier and underwent equivalent manufacturing and heat treatment processes. This supports the assumption that its mechanical properties are representative of those used in the numerical model.

Tensile specimens were extracted from three regions of the frame: the down tube, the chainstay, and a welded area of the chainstay. Samples were cut using a bandsaw, then ground to achieve parallelism within 0.02 mm. Midspan width was reduced manually to promote failure in the gage region. The nominal thickness was used for stress calculations, and weld beads were flattened to avoid artificial strength increase.

The tests were performed using an extensometer to ensure precise strain measurement in base material samples. The maximum tensile strength observed in the base material was approximately 275 MPa, with YS closely aligning at 260 MPa. These values are slightly lower than those reported in literature (UTS of 307 MPa, YS of 266 MPa) [36], likely due to minor heat treatment variations during post-processing.



(a) Tensile testing setup with extensometer.



(b) Al 6061-T6 test samples.

Figure 4.7: Tensile testing campaign: (a) strain measurement setup; (b) test specimens.

Figure 4.8 compares the reference stress–strain curves from ASM data with the experimental results. The yellow curve, representing the base material, shows strong agreement in YS (260 MPa), though the ultimate strength and elongation are slightly reduced.

4.3.3 Parameter Definition and Justification

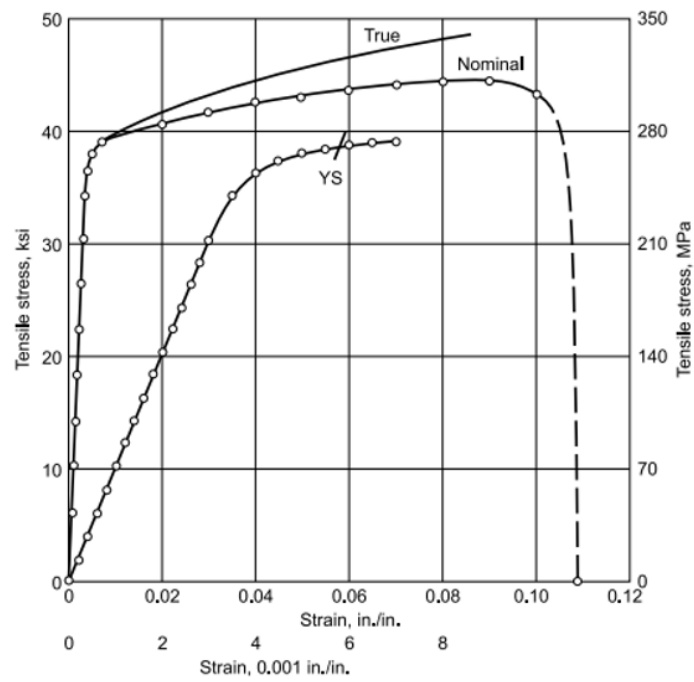
The JC parameters A , B , and n were chosen based on a combination of literature and the in-house tensile tests. The yield stress $A = 260$ MPa was directly supported by testing. The hardening coefficient $B = 113$ MPa and exponent $n = 0.42$ were selected based on data from a high strain-rate projectile impact study [38].

This extrapolation is considered valid because strain hardening reflects intrinsic material characteristics that do not vary significantly with strain rate, particularly in the low to moderate regime relevant to this project.

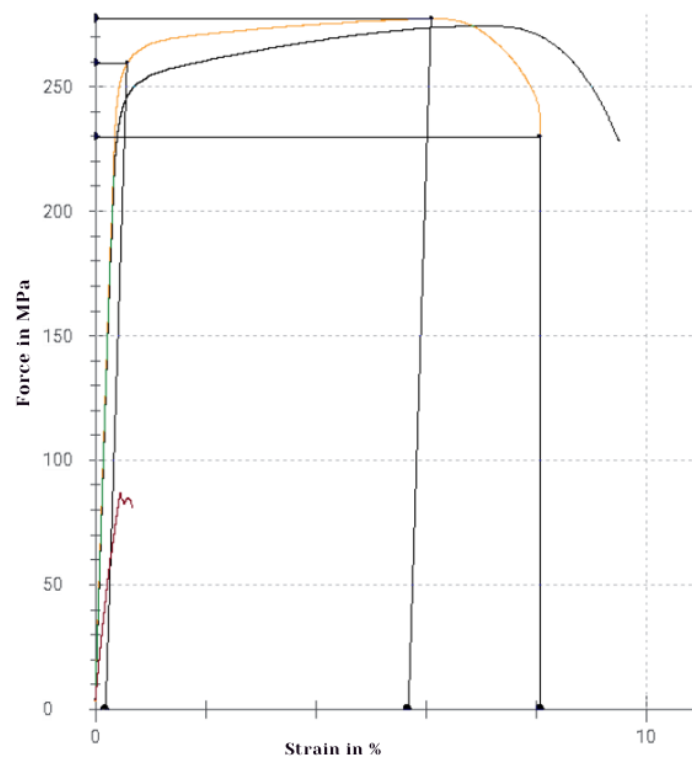
Table 4.1: Material properties of Al 6061-T6 used in simulation.

Property	Symbol	Units	Value
Density	ρ	kg/m ³	2700
Young's Modulus	E	GPa	69
Poisson's Ratio	ν	-	0.33
Yield Strength	A	MPa	260
Ultimate Tensile Strength	σ_u	MPa	310
Strain Hardening Coefficient	B	MPa	113
Strain Hardening Exponent	n	-	0.42

Although the geometry of the *Gazelle Eclipse* frame includes both extruded and cast components, all parts in the simulation were assigned the same material definition based on Al 6061-T6. This simplification was necessary due to the way the frame model was exported from SolidWorks. The casted regions, such as the motor mount, dropout areas and head tube were not represented



(a) Reference curve from ASM [37].



(b) Measured curves from frame samples.

Figure 4.8: Comparison of nominal and experimental stress–strain curves for Al 6061-T6.

as separate solid bodies in the CAD assembly, making it impossible to assign distinct material properties or sections to them in the simulation environment.

As a result, cast aluminum parts, typically made from A356 and known to have lower ductility and strength compared to Al 6061-T6, were modeled using properties of the stronger material. This means that the plastic strain values predicted in these regions are likely underestimated. While this modeling simplification limits the ability to assess local failure with full accuracy, it does not compromise the overall conclusions regarding global frame behavior and structural response.

4.4 Model Preparation

Due to the original CAD geometry being locked and uneditable within the 3DEXPERIENCE platform, an abstraction shape was created to serve as the working geometry for the simulation. This approach allows for controlled simplification and modification necessary for mesh generation and application of boundary conditions as well as other conditions, while maintaining the essential structural characteristics of the original design.

The approach used on this model was assembly of meshes, which creates a FEM representation at the global level by linking the part-level FEM representations of all individual parts in an assembly. When a part is used multiple times in an assembly, the part-level FEM representation is also instantiated along with the part. These parts contain the section properties, whereas the global FEM representation contains the connections for the assembly. This is a technique widely used on relatively large assemblies.

The parts meshed and assigned with solid section properties include the frame, seatpost insert, bottom bracket shell, steerer tube, crown, fork legs, roller, and the ground representing the steel anvil where the roller impacts.

4.4.1 Mechanical Properties

The mechanical properties described in this section apply specifically to components assembled onto the frame, including the fork, roller, and crown, presented in Section 4.2.1

Material properties were obtained through a literature review [39, 40] and represent typical behavior for structural steels and aluminum alloys. The plasticity behavior used in simulations was derived from the *Atlas of Stress–Strain Curves* [37], and implemented using the stress–strain curve shown in Figure 4.9. According to international standards, ASTM A572 steel corresponds to S355JR, while ASTM A36 is equivalent to S235JR.

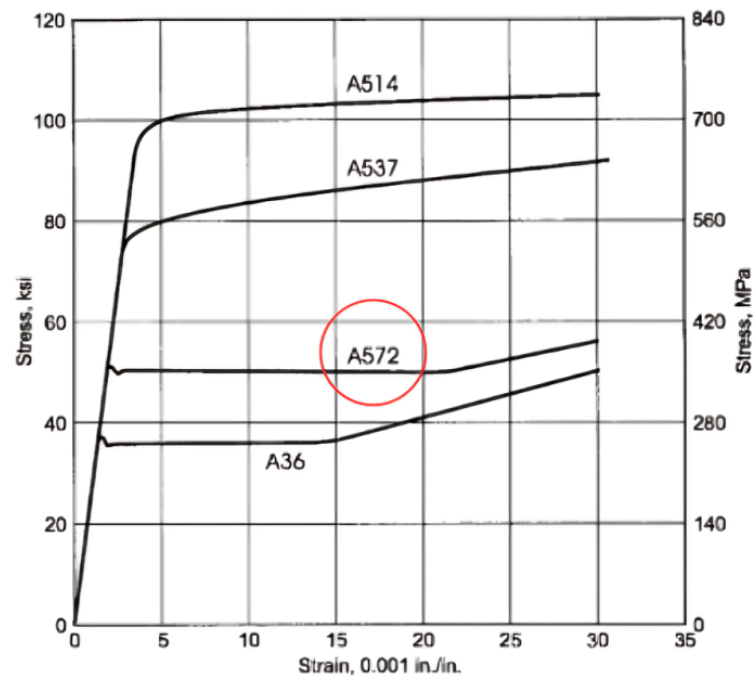


Figure 4.9: Stress–strain curve for S355JR and S235JR steels, used to define plasticity for assembled components. Adapted from the Atlas of Stress–Strain Curves [37].

The bottom bracket shell was assigned the material AISI 1035 steel, based on a scaled density compatibility check and visual assessment of the surface finish. Since this part is not the focus of the analysis and was included solely for weight representation, the exact material properties are not considered critical in this context. The same applied to the steel anvil.

4.4.2 Structural Model Creation

The structural components of the FEM were defined and assembled explicitly for simulating the falling frame impact test. This section details the assignment of solid sections and materials, the application of inertial masses, and the establishment of mechanical connections between parts.

All components within the model were assigned solid sections to accurately represent their mechanical response, particularly regarding stiffness and elastic deformation characteristics. The assigned materials for each component are summarized in Table 4.2.

Table 4.2: Assigned solid sections and materials for each model component.

Component	Section Type	Material Assigned
Ground	Solid	AISI 1035 Steel
Seatpost Insert	Solid	S355 Steel
Steerer Tube	Solid	S355 Steel
Crown	Solid	AlCuMgPb F38 (Al2024-T6)
Frame	Solid	Al 6061-T6
bottom bracket shell	Solid	AISI 1035 Steel
Fork Legs	Solid	S235JR
Roller	Solid	S235JR

All materials were modeled as isotropic, reflecting the absence of directional material properties (anisotropy) in the tested components.

4.4.3 Mass Distribution Setup

As referenced in Chapter 3, falling frame impact tests require appropriate mass placement to replicate real-world load conditions on the structure. For this study, a total mass of 95 kg was distributed across the frame using dummy weights, with 50 kg placed at the seat tube, 30 kg at the bottom bracket, and 15 kg at the head tube. This setup was selected based on the availability of weights in the physical test lab and was designed to maintain a representative moment distribution across the frame.

Although this configuration differs from official internal testing protocols, the relative balance between seat tube and bottom bracket loads was preserved to ensure structural relevance. The slight deviation at the head tube, where a 15 kg mass was used instead of a fully proportional value, was considered negligible in terms of its influence on the overall deformation response.

To ensure the validity of the analysis, this same mass configuration was implemented in the numerical model. Maintaining consistency between the experimental and simulation setups was prioritized throughout the study.

4.4.3.1 Point Inertia Modeling

To replicate the additional masses required by ISO 4210-6:2023 [1], point inertia was applied at three key locations: the seatpost insert, the bottom bracket, and the steerer tube. Point inertia was employed instead of volumetric modeling to significantly reduce mesh complexity and computational time while accurately representing inertial effects [41].

Each point inertia includes translational mass and rotational inertia (I_{11} , I_{22} , I_{33}), representing the rigid body behavior of the physical test weights. The rubber weights used in experimental testing were approximated as homogeneous solid cylinders, with their inertia tensors calculated according to the standard formulas for solid cylinders:

$$I_{xx} = I_{yy} = \frac{1}{12}m(3R^2 + h^2), \quad I_{zz} = \frac{1}{2}mR^2 \quad (4.3)$$

Dimensions for the inertia calculations were based on the actual commercial *Domyos* rubber weights:

- 20 kg: $R = 0.1975$ m, $h = 0.045$ m
- 10 kg: $R = 0.1450$ m, $h = 0.040$ m
- 5 kg: $R = 0.1250$ m, $h = 0.0325$ m.

The computed inertia values for each weight are:

- 20 kg: $I_{xx} = 0.1983$ kg·m², $I_{zz} = 0.3901$ kg·m²
- 10 kg: $I_{xx} = 0.0712$ kg·m², $I_{zz} = 0.1051$ kg·m²
- 5 kg: $I_{xx} = 0.0270$ kg·m², $I_{zz} = 0.0391$ kg·m².

Figure 4.10 illustrates the application of point inertia at the seatpost insert location, while Figure 4.11 shows the actual physical weights used in laboratory testing.

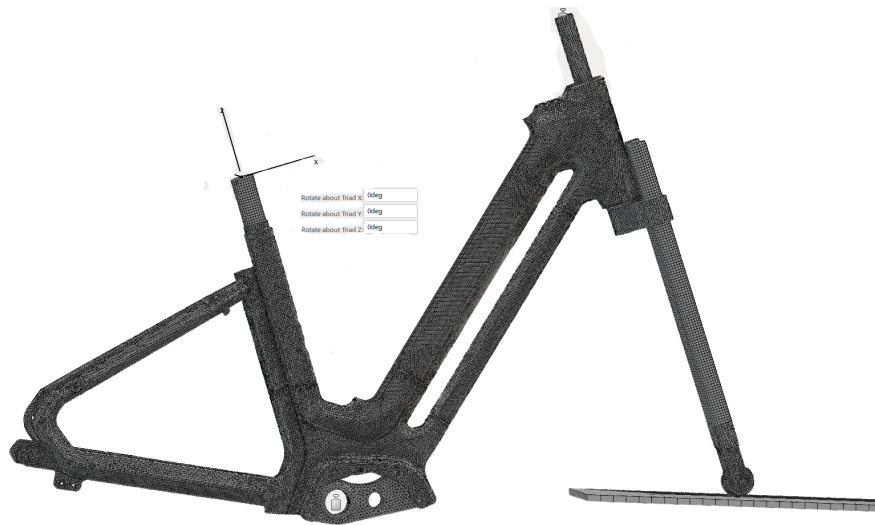


Figure 4.10: Defined inertia axes for the seatpost insert.



Figure 4.11: Physical weights utilized in experimental impact tests.

4.4.3.2 Component Connections via Tie Constraints

To maintain structural continuity and ensure accurate force transfer, tie constraints were used to rigidly connect the various components of the model. Tie constraints enforce identical displacement and rotational behavior between connected surfaces, independent of mesh compatibility. This approach ensures numerical stability and avoids complex contact formulations, thus simplifying the model setup [41].

Constraints were defined directly at the geometric level, ensuring they remain consistent through mesh regenerations. The *node-to-surface* discretization method was utilized, combined with a position tolerance to account for small geometric mismatches between mating surfaces.

A total of ten tie constraints were defined, connecting primary components and points representing mass locations. These ties are summarized in Table 4.3.

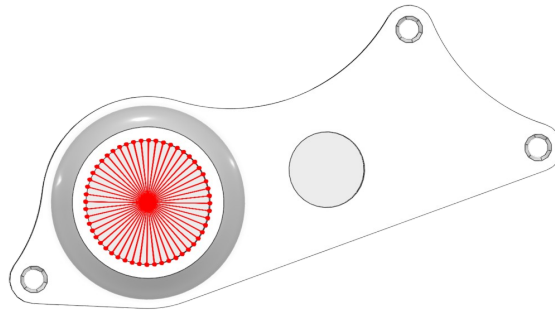
Table 4.3: Defined tie constraints between model components.

Tie Name	Main Surface	Secondary Surface
SP_Insert_Frame	Frame seatpost interface	Seatpost insert interface
Left_Bottom_Cover	Frame interface	Dummy cover left surface
Right_Bottom_Cover	Frame interface	Dummy cover right surface
HT_Frame	Frame	Head tube
LeftTube_Crown	Left fork leg interface	Crown interface
RightTube_Crown	Right fork leg interface	Crown interface
SteererTube_Crown	Steerer tube interface	Crown interface
SP_Inertia	CG1	Seatpost upper face
BB_Inertia	CG2	Bottom bracket inner face
HT_Inertia	CG3	Steerer tube upper face

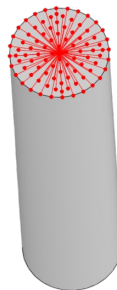
Where:

- CG1: seatpost mass center of gravity;
- CG2: bottom bracket mass center of gravity;
- CG3: head tube mass center of gravity.

Figure 4.12 presents representative tie constraints applied at the bottom bracket and seatpost insert.



(a) Bottom bracket mass application point tied to the inner face of the hole.



(b) Seatpost mass application point tied to the surface.

Figure 4.12: Tie definitions for key structural interfaces.

Figure 4.13 displays the tie connection between the steerer tube and head tube.

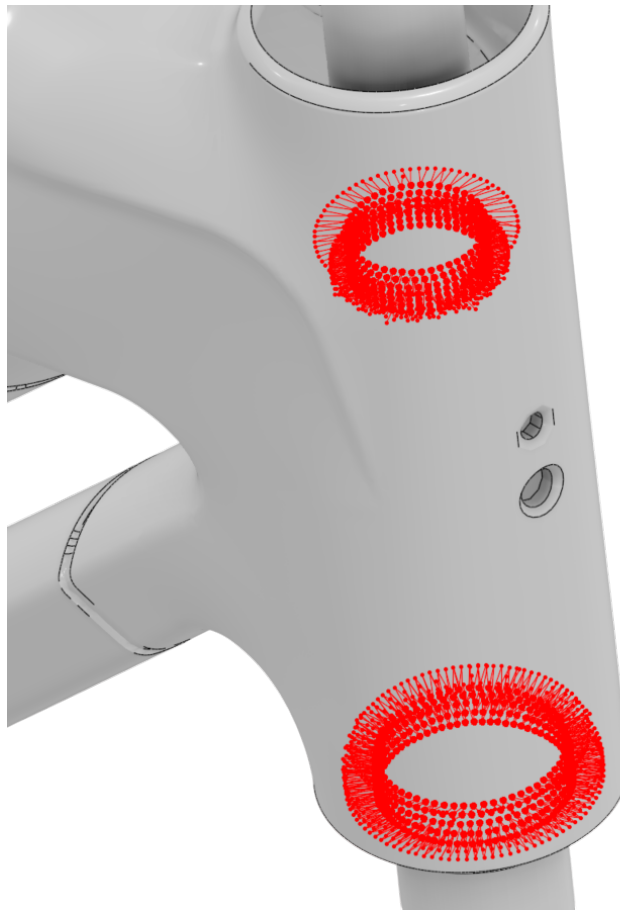


Figure 4.13: Tie constraint between steerer tube and head tube.

It should be noted that the tie constraint between the steerer tube and the head tube represents a significantly stiffer connection than in reality. In practice, this interface involves ball bearings, which allow relative rotation and do not fully restrict rotational degrees of freedom. This modeling assumption may lead to an overestimation of local stiffness in the numerical results.

The dummy cover at the bottom bracket was connected using tie constraints, omitting explicit bolt modeling due to its non-critical structural role. This simplification provides adequate accuracy and reduces computational complexity. Tie constraints involving the CG points ensured precise load transmission without artificial stress concentrations.

4.4.4 Meshing

The meshing strategy for the FEM was defined according to the geometric complexity and requirements of explicit dynamic analysis in ABAQUS/Explicit [41]. A hybrid discretization approach was adopted, combining quadratic tetrahedral elements (C3D10M) and linear hexahedral elements with reduced integration (C3D8R).

The frame, bottom bracket shell, crown, steerer tube, and roller were meshed using C3D10M elements due to their suitability for complex geometries and their robustness under large deformation scenarios. As ten-node second-order tetrahedral elements with improved interpolation capabilities, C3D10M elements are particularly effective in capturing stress gradients and localized strain distributions, making them well-suited for impact simulations involving nonlinear material behavior and contact interactions [41].

In contrast, the seatpost insert, fork legs, and ground plate, which are components with regular geometries, were discretized using C3D8R elements. These eight-node first-order hexahedral elements with reduced integration offer improved computational efficiency while maintaining sufficient accuracy in regions with linear stress distribution. Although susceptible to hourglassing, this issue is controlled within ABAQUS through the use of built-in hourglass stabilization algorithms in explicit dynamics.

To preserve structural detail in areas of interest, such as transition zones and expected stress concentration regions, geometry reduction was avoided. Instead, mesh simplification was governed by feature recognition rules, excluding elements associated with features below 2 mm in size or angular transitions under 60° . This approach ensured that relevant geometric detail was retained without excessive refinement.

Additional mesh refinement was applied locally in regions of expected high stress, particularly around the bottom bracket, where prior physical testing revealed crack initiation. Targeted rules included the use of three washer layers and ten edge divisions for holes ranging from 1–4 mm in diameter. Fillet features with radii between 1–30 mm were resolved using a single layer to maintain curvature fidelity while minimizing unnecessary mesh density.

Global mesh conformity was enforced using a node and edge capture tolerance of 0.2 mm. This ensured continuous node alignment across part boundaries, preventing interfacial discontinuities. Furthermore, geometry curves were projected onto the mesh to enhance accuracy at surface transitions. The final meshing parameters for the frame are presented in Table 4.4, which also includes the total element and node counts resulting from this discretization strategy.

Table 4.4: Final mesh parameters for the frame.

Parameter	Value
Element type	C3D10M
Element order	Quadratic
Mesh size	5 mm
Absolute sag	0.7 mm
Number of elements	336,060
Number of nodes	627,686

Table 4.5 presents the mesh quality statistics for the frame component. Element quality was assessed based on ABAQUS default criteria, including aspect ratio and element distortion metrics.

Table 4.5: Global mesh quality summary for the frame.

Total Elements Checked	336,060
Good Elements	335,292
Poor Elements	765
Bad Elements	3

The remaining model components were meshed using a consistent approach. Node and edge capture ensured mesh conformity across shared interfaces.

Table 4.6 summarizes the mesh parameters for these components, including the fork assembly, seatpost insert, bottom bracket shell, and ground.

Table 4.6: Mesh parameters and element count for fork components and ground.

Component	Element Type	Mesh Size	Absolute Sag	Number of Elements
Crown	C3D10M	5 mm	1 mm	8,311
Steerer Tube	C3D10M	4 mm	1 mm	13,903
Insert	C3D8R	4 mm	–	3,003
Fork Legs	C3D8R	5 mm	–	15,656
bottom bracket shell	C3D8R	5 mm	–	7,727
Ground	C3D8R	20 mm	–	528

The ground was discretized with structured C3D8R linear element, as deformation was not the focus. Despite functioning as an impact surface, it was modeled as deformable (not as rigid body) to improve contact response and simulation robustness.

Table 4.7 presents the global mesh quality assessment.

Table 4.7: Global mesh quality summary for the complete model.

Total Elements Checked	385,194
Good Elements	384,126
Poor Elements	1,065
Bad Elements	3

The global mesh quality assessment confirmed that over 99.7% of the elements were classified as *good*, indicating overall high mesh fidelity. Only three elements were labeled as *bad*, primarily due to aspect ratios slightly exceeding the default threshold of 10, reaching values between 10 and 11. The majority of the *poor* elements did not fail due to severe geometric distortion, but rather marginally violated the stretch criterion, with values approaching the lower limit of 0.3. To ensure solver robustness and numerical stability, all quality criteria were maintained at the default settings

recommended for ABAQUS/Explicit, avoiding over-constraint or tolerance relaxation that might compromise result and mesh accuracy.

4.5 Mechanical Scenario Creation

The Mechanical Scenario Creation stage represents the second phase in the FEM setup within the 3DEXPERIENCE platform. In this stage, the physical conditions of the simulation are defined, including the time-dependent procedure, boundary conditions, and applied loads.

The following section presents a detailed explanation of how these features were implemented in the model and establishes their correspondence with the conditions observed in the experimental impact tests.

4.5.1 Explicit Dynamic Step

The explicit dynamic step was selected to simulate the frame drop test due to the transient and contact-driven nature of the loading scenario. Although the event does not fall within the high strain-rate regime, it involves a rapid application of load over a short time interval, along with complex interactions between parts. These characteristics make the explicit formulation appropriate, as it offers stable time integration and efficient handling of highly nonlinear contact and material behavior.

In Abaqus/Explicit, the dynamic equilibrium equations are solved using a central-difference time integration scheme. This method updates nodal velocities and displacements at each increment based solely on known values from the previous step, eliminating the need to solve a global system of equations and improving computational efficiency for large models with numerous contact interactions.

The method integrates the equations of motion using:

$$\dot{u}_{(i+\frac{1}{2})}^N = \dot{u}_{(i-\frac{1}{2})}^N + \frac{\Delta t_{(i+1)} + \Delta t_{(i)}}{2} \ddot{u}_{(i)}^N, \quad (4.4)$$

$$u_{(i+1)}^N = u_{(i)}^N + \Delta t_{(i+1)} \dot{u}_{(i+\frac{1}{2})}^N, \quad (4.5)$$

where:

- u^N is the displacement (or rotational) degree of freedom at node N ;
- $\dot{u}^N$ and $\ddot{u}^N$ are the velocity and acceleration, respectively;
- the subscript i denotes the increment number in the explicit time-stepping procedure.

A key feature enabling the computational efficiency of this method is the use of *lumped* (diagonal) mass matrices. This allows the acceleration vector to be computed explicitly as:

$$\ddot{u}^N = (M^{NJ})^{-1} (P_{(i)}^J - I_{(i)}^J), \quad (4.6)$$

where M^{NJ} is the mass matrix, P^J is the externally applied force vector, and I^J is the internal force vector. The use of a diagonal mass matrix simplifies inversion and allows the accelerations to be computed with only n operations, where n is the number of degrees of freedom. This approach requires no global stiffness matrix and no iterative convergence procedure [41].

In summary, the explicit dynamic procedure is especially suited for:

- High-speed, inertia-dominated events such as impact and crash simulations;
- Highly nonlinear contact and deformation problems;
- Cases where numerical stability demands very small time increments.

Given the transient nature of the loading, the presence of contact interactions, and the need to capture localized plastic deformation, this step is the appropriate choice for simulating the frame impact scenario considered in this study.

4.5.2 Initial conditions

In this simulation, the objective is not to model the free-fall phase but to analyse the structural behavior of the bicycle frame after it contacts the ground. To reduce computational time, the fall itself is omitted and replaced by an initial velocity condition corresponding to the estimated impact speed.

The frame drops from a height of 300 mm, rotating about the rear dropout axis, as previously illustrated in Figure 3.2. When the frame falls, potential energy is converted into rotational kinetic energy around this axis, which is aligned with the global Z-axis in the simulation. The angular velocity ω at impact is determined based on this condition.

The rotational behavior depends on the model moment of inertia about the rear axis, which governs its resistance to angular acceleration during free fall. To replicate the impact state, the ground is positioned 0.634 mm below the roller, so that contact occurs immediately and the assigned rotational velocity fully represents the pre-impact condition.

All component masses were verified against their physical counterparts to ensure consistent inertial properties, this alignment is critical for accurately capturing the frame's dynamic response at impact.

Table 4.8: Comparison between simulated and real component masses, material density, and relative error.

Component	Density [kg/m ³]	Simulation [kg]	Real Mass [kg]	Error [%]
Frame	2,700	5.413	5.400	0.24
Roller	7,850	1.300	1.200	8.33
Steerer Tube	7,850	2.120	2.300	7.83
Crown	4,250	0.564	0.600	6.00
Legs	7,850	6.458	7.000	7.77
Seatpost Insert	7,850	1.198	1.100	8.91
bottom bracket shell	7,850	1.438	1.600	10.13

In the physical test, a cylindrical support is attached to the largest hole in the dummy cover. Since this component is not structural, its influence is accounted for in the simulation by incorporating its mass into the applied load at CG2.

Additional small components such as seat clamps, spacers along the steerer tube, and fastening hardware were part of the physical setup but were not included in the FEM. Their mass contribution is negligible and does not significantly affect the simulation results regarding mass distribution or structural response.

The simulation software reports the inertial properties of the model, including total mass, center of gravity, and moments of inertia calculated both about the center of mass (CoM) and a fixed reference point. In this study, the reference point corresponds to the coordinate system origin, which is defined at the center of the largest hole in the dummy bottom bracket shell (see red circle in Figure 4.14). These inertial values are recorded in the simulation status file, along with additional information such as time increments and energy variation throughout the analysis.



Figure 4.14: Reference coordinate system used for inertia calculations.

4.5.3 Rotational Velocity Estimation at Impact

To accurately estimate the angular velocity ω of the frame at the instant of impact, an energy-based approach using conservation of mechanical energy was adopted. The frame was assumed to rotate freely about the rear dropout axis, and the assembly was treated as a rigid body in planar motion.

In this analysis, the key assumption is that the CoM of the frame follows a circular trajectory around the rear dropout axis. The moment of inertia I_z about this axis is a geometric property that remains constant for a rigid body, irrespective of spatial position. Consequently, vertical displacement of the CoM only affects the potential energy of the system, not the inertia characteristics.

In the physical test scenario, the frame was dropped from a vertical height of 300 mm above the ground. In contrast, within the simulation setup, the front roller is located only 0.634 mm above the ground plane. As a result, the vertical position of the CoM in the simulation lies significantly closer to the ground when compared to the actual physical test. To ensure energetic equivalence between simulation and physical setups, an additional vertical offset of 50 mm was artificially added to the vertical coordinate of the CoM. This offset compensates for the reduced height caused by roller positioning and enables accurate calculation of impact conditions.

Based on the above considerations, the following assumptions were made:

- The rear dropouts define the fixed axis of rotation;
- The frame is modeled as a rigid multibody system during free fall;
- Air resistance effects are neglected;
- Motion is constrained to the x - y plane (planar motion).

The ground body was not included in the rotational velocity calculation, as it remains stationary and does not contribute to the system's mass or inertia during free fall.

Energy Balance and Angular Velocity

Considering conservation of mechanical energy, the gravitational potential energy lost by the CoM of the frame during the fall is fully converted into rotational kinetic energy at the instant of impact:

$$mgh = \frac{1}{2}I_z\omega^2, \quad (4.7)$$

where:

- m is the total mass of the frame assembly [kg];
- $g = 9.81 \text{ m/s}^2$ is gravitational acceleration;
- h is the vertical distance from the CoM to the ground plane [m];
- I_z is the moment of inertia about the rear axis (global Z-axis) [$\text{kg}\cdot\text{m}^2$];
- ω is the angular velocity at the moment of impact [rad/s].

Solving for the angular velocity ω :

$$\omega = \sqrt{\frac{2mgh}{I_z}}. \quad (4.8)$$

Adjustment of the Moment of Inertia

The moment of inertia I_z in this calculation must be evaluated with respect to the rear dropout axis. Typically, the simulation software provides the inertia tensor relative to the CoM or a fixed global origin. To convert this inertia to the rear axis, the parallel axis theorem is employed:

$$I_z = I_{cg,z} + md^2, \quad (4.9)$$

where:

- $I_{cg,z}$ is the moment of inertia about the global Z-axis through the CoM [kg·m²]
- d is the perpendicular distance between the CoM and the rear axis [m].

Calculation of Drop Height

The vertical drop height h corresponds to the vertical distance between the CoM and the ground. In the simulation model, the global origin was defined at the center of the largest hole in the dummy bottom bracket shell, positioned 29.7 mm below the roller contact point. The CoM vertical coordinate relative to this origin is 357.2 mm. Considering the 50 mm offset to replicate the physical drop conditions, the total vertical distance h becomes:

$$h = 357.2 \text{ mm} - 29.7 \text{ mm} + 50.0 \text{ mm} = 377.5 \text{ mm} = 0.3775 \text{ m}. \quad (4.10)$$

Determination of Distance between Center of Mass and Rear axis

The global coordinates (in the x - y plane) for the CoM and rear axis are as follows:

- CoM: $(-5.90 \times 10^{-3}, 0.3572) \text{ m}$
- Rear axis: $(-0.4849, 0.0550) \text{ m}$.

Therefore, the distance d between these points is calculated by:

$$d = \sqrt{(x_{cg} - x_{axis})^2 + (y_{cg} - y_{axis})^2} = \sqrt{(-0.479)^2 + (0.3022)^2} = 0.5662 \text{ m}. \quad (4.11)$$

Numerical Parameters and Final Calculation

The relevant numerical parameters are summarized as follows:

- Total mass: $m = 137.94$ kg
- Vertical drop height: $h = 0.3775$ m
- Inertia about CoM: $I_{cg,z} = 17.60526$ kg · m².

Applying the parallel axis theorem, the moment of inertia I_z about the rear axis is:

$$\begin{aligned}
 I_z &= I_{cg,z} + md^2 \\
 &= 17.60526 + 137.94 \times (0.5662)^2 \\
 &= 17.60526 + 44.2203 \\
 &= 61.8256 \text{ kg} \cdot \text{m}^2.
 \end{aligned} \tag{4.12}$$

Finally, calculating the angular velocity ω :

$$\begin{aligned}
 \omega &= \sqrt{\frac{2mgh}{I_z}} \\
 &= \sqrt{\frac{2 \times 137.94 \times 9.81 \times 0.3775}{61.8256}} \\
 &= \sqrt{\frac{1021.30}{61.8256}} \\
 &= \sqrt{16.52} \\
 &= 4.07 \text{ rad/s}.
 \end{aligned} \tag{4.13}$$

Thus, the angular velocity applied in the finite element simulation as the initial rotational condition is:

$$\boxed{\omega = 4.07 \text{ rad/s}} \tag{4.14}$$

4.5.4 Boundary Conditions and Loads

The boundary conditions were defined to replicate the experimental setup as closely as possible. A hinge constraint was applied at the rear dropouts of the frame, allowing free rotation about the horizontal axis while restricting all translational degrees of freedom. This configuration enables the frame to rotate realistically during the falling motion. In addition, the ground body representing the steel anvil was fully fixed on its lower surface, preventing any displacement or rotation.

Figure 4.15 illustrates both boundary conditions implemented in the simulation environment. The clamped condition on the ground body (Figure 4.15a) and the hinge constraint (Figure 4.15b) at the rear axis are highlighted to show how the model setup reflects the actual test configuration.

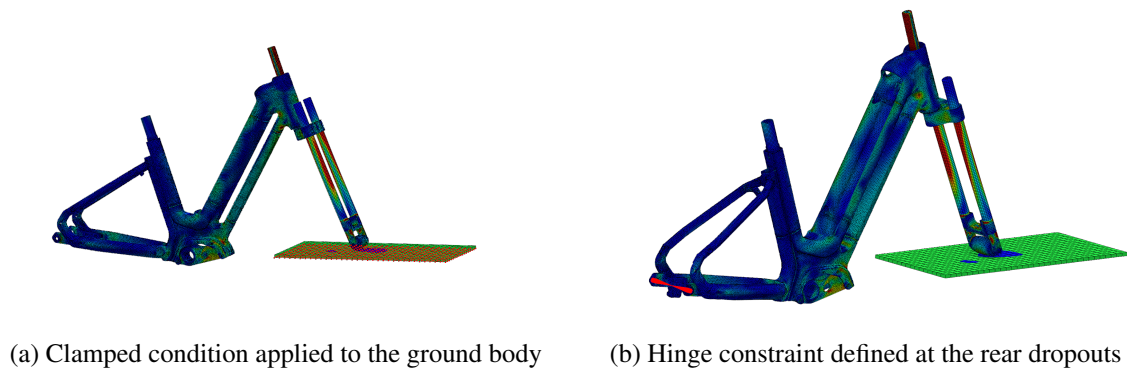


Figure 4.15: Boundary conditions applied in the simulation model.

The only external load applied in the simulation was gravitational acceleration, acting uniformly on the mass of the frame. This choice reflects the actual physical conditions of the vertical drop test, consistent with ISO 4210-6:2023, where gravity-induced free fall is the governing loading mechanism.

4.5.5 Interactions

Contact interactions between the roller and the steel anvil plate were defined using the General Contact algorithm in ABAQUS/Explicit, which is appropriate for handling severe contact conditions and large deformations [41]. The contact was modeled with a hard contact formulation in the normal direction, ensuring non-penetration under compression and permitting separation under tension. This approach closely replicates the observed physical interactions during impact tests, where no penetration between contacting surfaces occurs.

Tangential interactions were assumed frictionless to simplify the simulation and to focus purely on the structural response of the frame. This decision eliminated additional complexity, such as energy dissipation due to friction, allowing a clearer evaluation of deformation and impact behavior associated with the frame.

4.5.5.1 Damping Representation and Energy Dissipation Mechanisms

In dynamic simulations, energy dissipation occurs through multiple mechanisms. These include material plasticity, frictional contact, and numerical damping models. In the context of the frame impact test, the primary sources of energy dissipation are:

- **Plastic deformation:** the aluminum 6061-T6 alloy exhibits permanent deformation under impact, which irreversibly absorbs part of the input energy. This is accounted for by the plasticity model used in the simulation;

- **Frictional Contact** During the impact, interactions at contact interfaces such as the frame to impactor or floor may dissipate energy through Coulomb friction. This requires an explicit contact model and contributes to local energy loss. This is not explicitly modeled in our simulation, which follows a simplicity-first approach;
- **Rayleigh damping:** introduced as a numerical mechanism to emulate additional internal dissipation not explicitly captured by plasticity or contact. The damping matrix is defined as:

$$\mathbf{C} = \alpha \mathbf{M} + \beta \mathbf{K}, \quad (4.15)$$

where α and β are the mass- and stiffness-proportional damping coefficients, respectively.

Although Rayleigh damping does not correspond to a physical mechanism, it is widely used to stabilize simulations and to replicate the observed attenuation of dynamic responses. However, its implementation, particularly the stiffness-proportional term, can significantly increase computational cost due to matrix operations performed at each timestep [24, 42]. Moreover, excessive damping artificially suppresses physical responses and can lead to energy imbalance. The damping ratio curve as a function of frequency using the full model is shown in Figure 4.16.

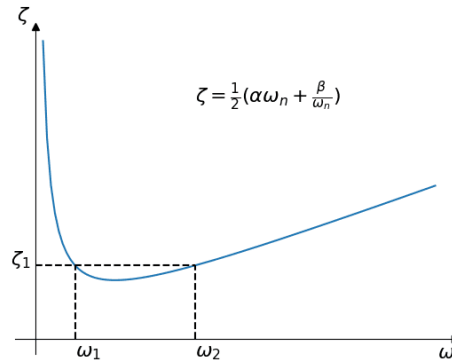


Figure 4.16: Schematic of the full damping model [43].

To evaluate the effect of Rayleigh damping in a controlled setting, a simplified model was constructed to replicate the boundary and loading conditions of the experimental setup. As shown in Figure 4.17, the model consists of a parallelepiped main body connected to a cylindrical pin and a rigid base. Second-order tetrahedral elements (C3D10M) were used for the main body to maintain consistency with the full frame model, while reduced-integration hexahedral elements (C3D8R) were used for the pin and base. Mesh characteristics are detailed in Table 4.9.

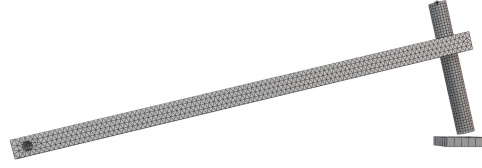


Figure 4.17: Simplified model.

Table 4.9: Mesh composition and resolution in the simplified impact model

Element Type	Element Size	Number of Elements
C3D10M	5 mm	17,311
C3D8R	3 mm	2,286
Total	–	19,597

The mesh passed all quality checks, with 100% of elements meeting the required criteria.

A preliminary damping study was conducted using this simplified model. Rayleigh damping configurations included the undamped case ($\alpha = 0$) and two low levels of mass-proportional damping: $\alpha = 0.01$ and $\alpha = 1$. A stiffness-proportional coefficient β was also evaluated but led to a significant increase in computational cost, consistent with findings in [44]. Consequently, β was set to zero, and only the mass-proportional term was retained for further studies.

These initial tests showed that even with $\alpha = 1$, viscous dissipation remained low (0.148 J compared to a total system energy of 60 J), and displacements were largely unaffected. Based on these results, the damping strategy was further investigated using the complete frame model to verify that the behavior remained consistent under the full geometric and inertial conditions.

Since Rayleigh damping is geometry dependent, due to its proportionality to the mass matrix M , which reflects the distribution of mass, the complete model was used to test larger damping coefficients: $\alpha = 5$ and $\alpha = 100$. These values were selected to explore the upper limits of damping influence while maintaining numerical stability.

For $\alpha = 5$, the viscous energy dissipated was approximately 11.71 J out of a total system energy of 411 J. At $\alpha = 100$, the dissipation increased to around 50 J, with the total energy reaching 460 J. This difference in total energy is attributed to a mass variation of approximately 5 kg, resulting from the final refinement of material densities and component weights applied in the model. Despite the higher damping in the $\alpha = 100$ case, the relative contribution of viscous dissipation remained moderate, and no evidence of numerical instability was observed.

Displacement comparisons across all configurations revealed negligible differences along the horizontal axis. In the vertical direction, some variations were observed, with maximum deviations of approximately 3 to 4 mm at peak amplitude. These results confirm that, even with elevated

damping values, the use of mass-proportional damping did not significantly alter the global dynamic response of the system.

Based on these findings, a damping coefficient of $\alpha = 5$ was selected for the final simulation. This value provided moderate viscous energy dissipation while avoiding excessive damping and preserving numerical stability throughout the analysis of the complete model.

In summary, while plasticity and contact dominate physical dissipation, Rayleigh damping is included to capture additional energy loss modes and improve solution robustness. The total energy dissipation in the simulation results from the combined effects of these mechanisms.

4.5.6 Output Requests and Solver Controls

Default output settings in ABAQUS/Explicit were used for both field and history data. The main variables monitored during the simulation were displacement (U), velocity (V), von Mises stress (S), and equivalent plastic strain ($PEEQ$).

Displacement results were used to compare the frame's movement in the simulation with the actual motion observed during the physical test. Stress outputs helped assess how loads were transmitted through the frame during impact, while plastic strain was analysed in critical regions and compared with prior knowledge on likely crack initiation zones. These outputs provided a clear view of the structural response over time without introducing additional complexity.

To capture the transient impact response with adequate temporal resolution, a total of 80 equally spaced output intervals were defined. This number was selected to align with the time intervals from the physical test tracking, allowing for direct comparison between experimental and simulated results. It provided smooth and detailed output curves while keeping computational cost and data size within reasonable limits. No additional or custom history outputs were defined, as the default settings sufficiently captured the necessary model diagnostics and global response variables.

4.6 Post Processing Results

In dynamic FEA, post-processing is essential to assess the physical realism and numerical stability of the simulation. One of the main verification steps is to check energy conservation. In an ideal explicit dynamic simulation, the total energy, which consists of kinetic energy, internal energy, artificial strain energy, viscous dissipation, and external work, should remain nearly constant. A deviation of less than 1% in total energy is typically acceptable [41].

Additionally, artificial strain energy (ALLAE), which represents non-physical numerical artifacts such as those from hourglass control, should remain significantly lower than the internal energy (ALLIE). Literature suggests that ALLAE should typically be under 5–10% of ALLIE to ensure the numerical behavior does not affect the physical response of the model [24].

Figure 4.18 shows the evolution of each energy component during the simulation. A smooth transition from kinetic to internal energy is observed, reflecting expected energy transfer during

impact. This confirms that the simulated plastic deformation and damping effects are physically consistent.

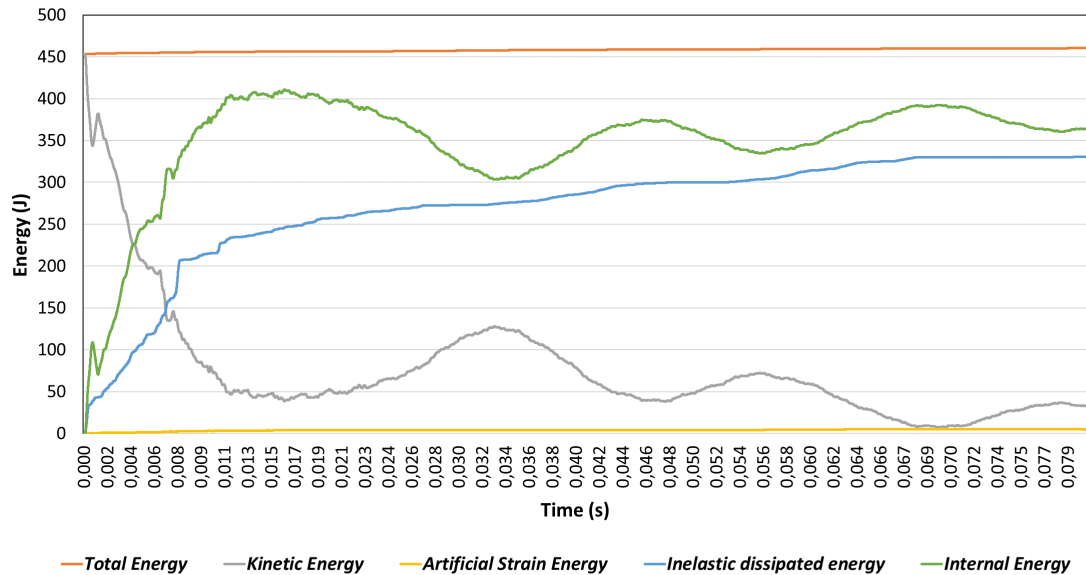


Figure 4.18: Global energy balance showing the evolution of kinetic, internal, artificial, viscous, and total energy during the impact simulation.

To further support this, Figure 4.19 shows the ratio of artificial strain energy (ALLAE) to internal energy (ALLIE) throughout the simulation. The curve remains well below the 5% threshold, staying consistently under 1.5%. This low ratio reinforces confidence that the artificial energy is negligible compared to the physical energy stored in the model, validating both the contact formulation and element quality used.

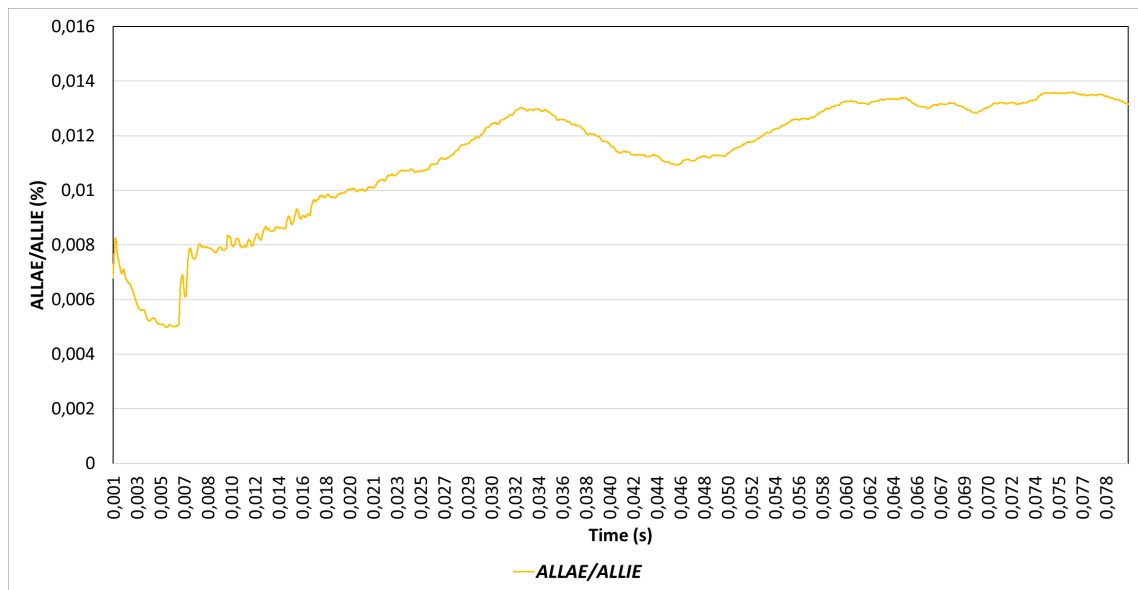


Figure 4.19: Ratio of artificial strain energy (ALLAE) to internal energy (ALLIE) throughout the simulation. The low values confirm minimal numerical artifact influence.

The total simulation time was set to 0.08 seconds, which corresponds to approximately one-fourth of the post-impact duration observed in the physical test (~ 0.35 seconds).

Preliminary runs with a 0.04-second duration showed that the plastic strain at critical areas had not yet fully developed. In contrast, the 0.08-second simulations demonstrated stabilized results, with significantly reduced kinetic energy as shown in Figure 4.18. The evolution of plastic strain and von Mises stress also indicated no relevant changes beyond this point.

This duration was therefore considered sufficient to capture the structural response of interest while keeping computational costs reasonable. Table 4.10 summarizes the key solver statistics and mesh characteristics from the final simulation run, including the number of elements and nodes, total simulation time, and computational performance details.

Table 4.10: Solver Status and Mesh Details.

Parameter	Value
Number of Elements	385,194
Number of Nodes	1,088,004
Total Computational Time	56 hours 16 minutes 46 seconds
Stable Time Increment	1.72885E-08 seconds
Total Simulation Time	0.08 seconds

The simulation was executed in a cloud environment, enabling high-performance computation without overloading local systems. Parallel processing allowed other tasks to run concurrently, improving efficiency and avoiding the need for dedicated hardware.

Following energy stabilization checks, it is essential to verify whether the simulated deformation pattern is consistent with the behavior observed in the physical impact test. This provides an initial indication of whether the model responds realistically and helps to identify any discrepancies that may arise from simplified assumptions or limitations in the setup. Such a validation step should precede detailed analysis of stress, strain, or other numerical outputs. Figure 4.20 shows the frame's deformation at selected time intervals, which will be further compared with the experimental observations in the next chapter. The model shown without a mesh corresponds to the initial time step of the simulation and is included to facilitate position comparison.

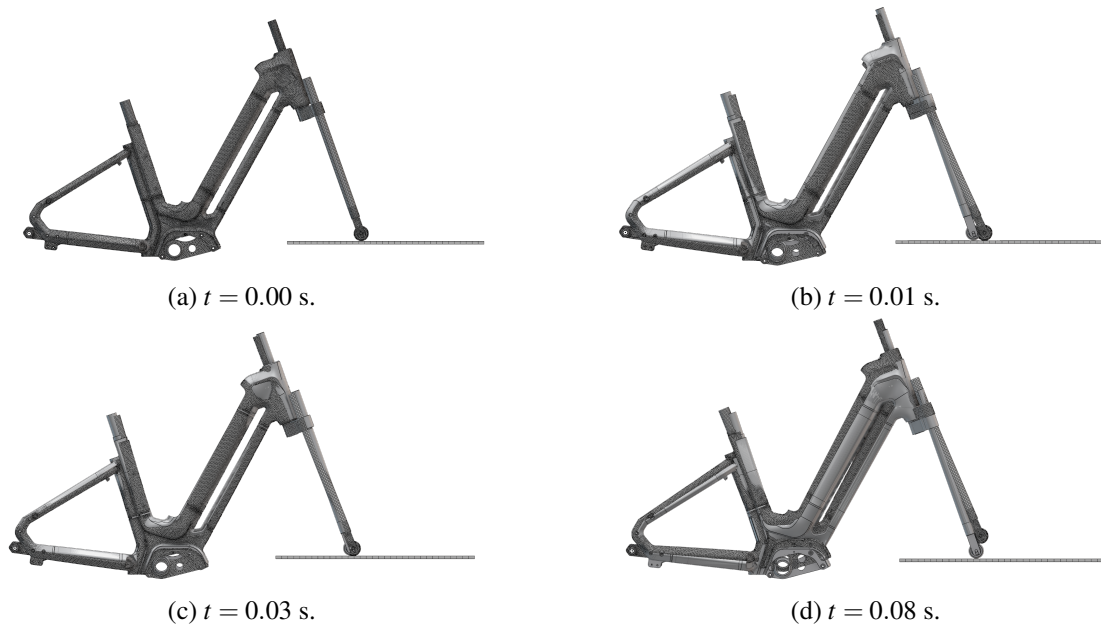


Figure 4.20: Deformation plot at different time instants.

The expected post-impact behavior, based on physical observations, involves the roller contacting the steel anvil, rolling forward in the global x -direction to its point of maximum displacement, and then slightly reversing due to elastic rebound. Small secondary bounces may also occur as part of the dynamic response, resulting from the partial conversion of stored internal energy back into kinetic energy.

This dynamic response is clearly reflected in the simulation frames. Although no numerical values are evaluated at this stage, the global movement and deformation sequence align well with physical observations. This indicates that the model setup, particularly the boundary conditions, initial velocity, and contact definitions, captures the key aspects of the real-world test scenario with good fidelity.

Stress and plastic strain contour plots were generated to evaluate whether the simulation reproduces the expected structural response under impact. These visualizations are important for identifying regions of high loading and permanent deformation, and for comparing simulation predictions with real test outcomes.

Figure 4.21 presents the von Mises stress distribution at $t = 0.043$ s, a moment where the stress state is still significant and representative of how loads are transmitted through the frame. At later times, the stress levels decrease and contribute less to further deformation, making this time step the most relevant for assessing the active stress paths.

Figure 4.22 shows the equivalent plastic strain at the end of the simulation, $t = 0.08$ s, when the deformation has stabilized. This result indicates where permanent material deformation occurred, revealing the most critical areas of the frame under the tested conditions.

Together, these results provide a clear picture of the frame's mechanical response during impact. Legends are included in each figure for interpretation of contour levels.

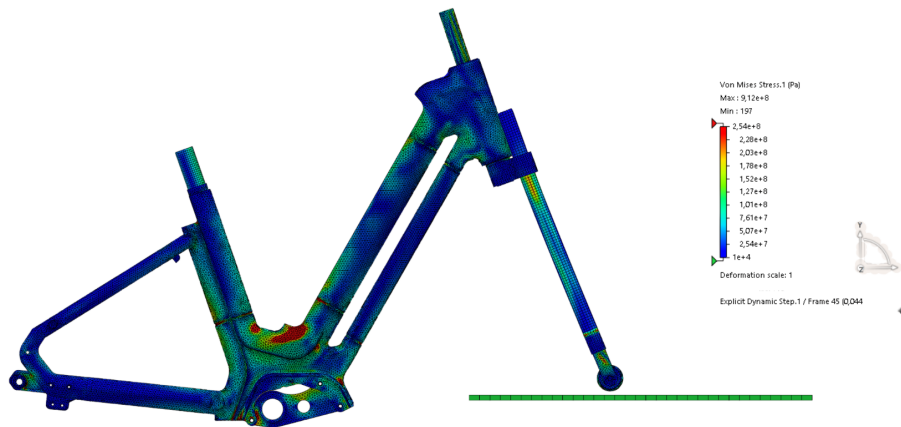


Figure 4.21: Von Mises stress distribution during impact simulation at $t = 0.043$ s.

The stress values in this plot highlight significant concentrations around the battery cover area. This is primarily due to the complex geometry in this region and the presence of an opening for the battery.

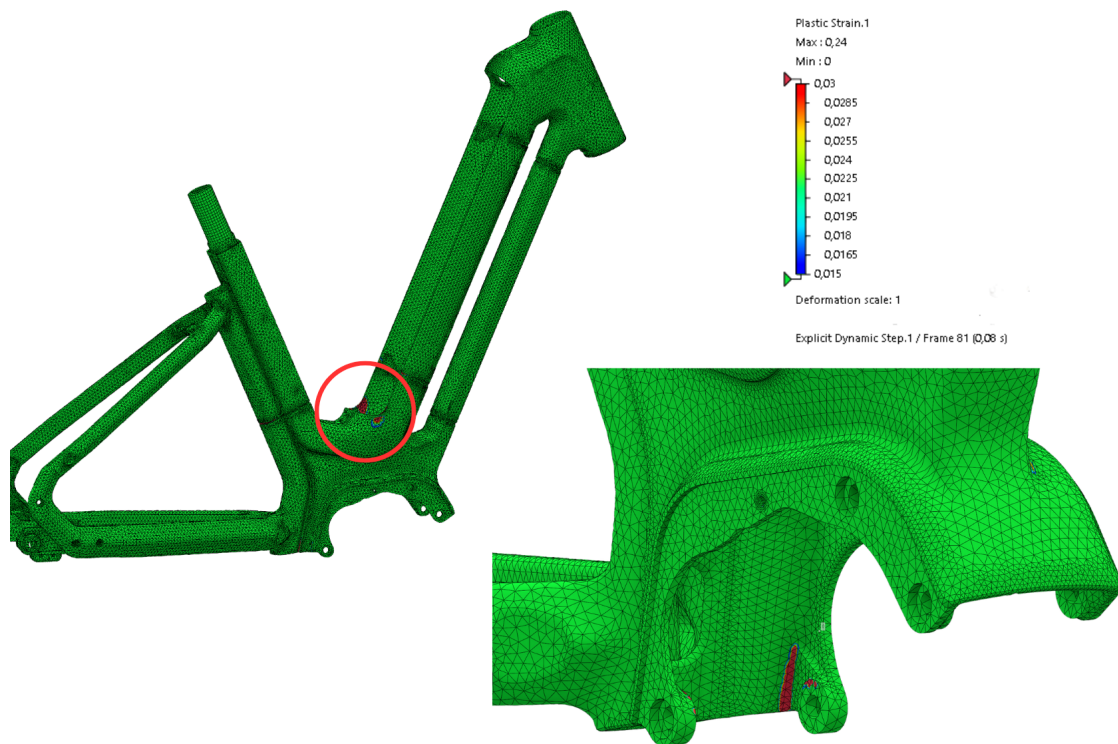
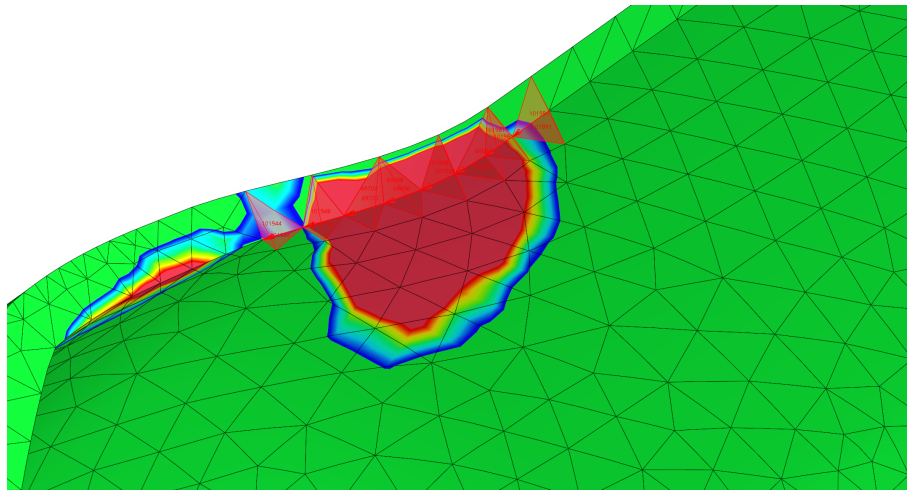


Figure 4.22: Equivalent plastic strain (PEEQ) distribution at $t = 0.08$ s.

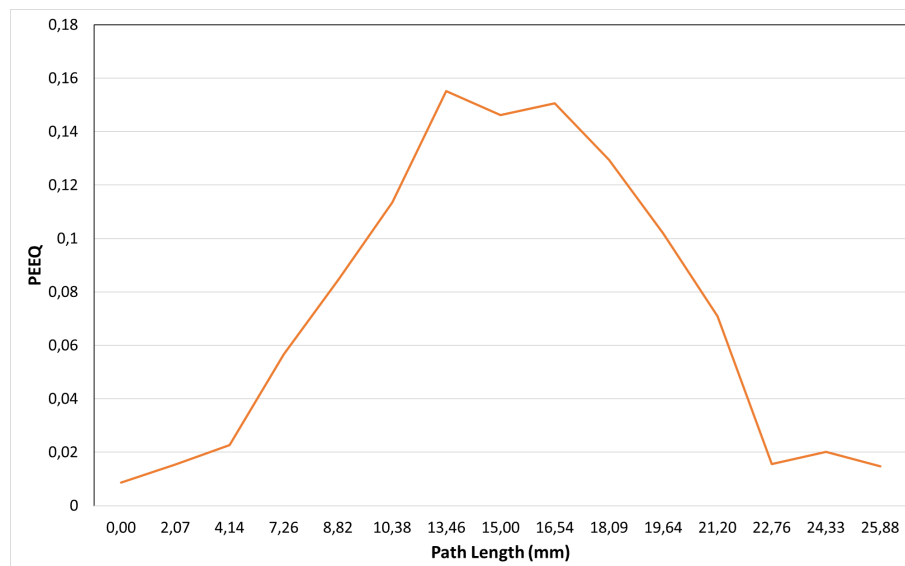
The plastic strain distribution reflects localized deformation consistent with prior in-house tests. The regions showing higher plastic strain values correspond closely to failure zones previously identified in physical experiments, as discussed in the first paragraph of this Chapter 4.

To investigate the plastic deformation behavior in critical regions of the frame, PEEQ was evaluated along predefined paths in the vicinity of the battery cover and the bottom bracket. These areas were selected based on prior experimental observations and the results from the global contour plots. The highest PEEQ values were expected to occur near the center of each path, where local deformation is typically concentrated. The paths and their corresponding strain distributions are presented below.

In the case of the battery cover, the path was defined along mesh edges, as strain accumulation was observed to be concentrated in these boundary regions. This behavior is consistent with areas of geometric discontinuity, where edge features are prone to higher deformation. As shown in Figure 4.23a, the path was created along the transition zone near the battery opening, and the corresponding PEEQ values along this path are displayed in Figure 4.23b.



(a) Path defined along the vicinity of the battery cover

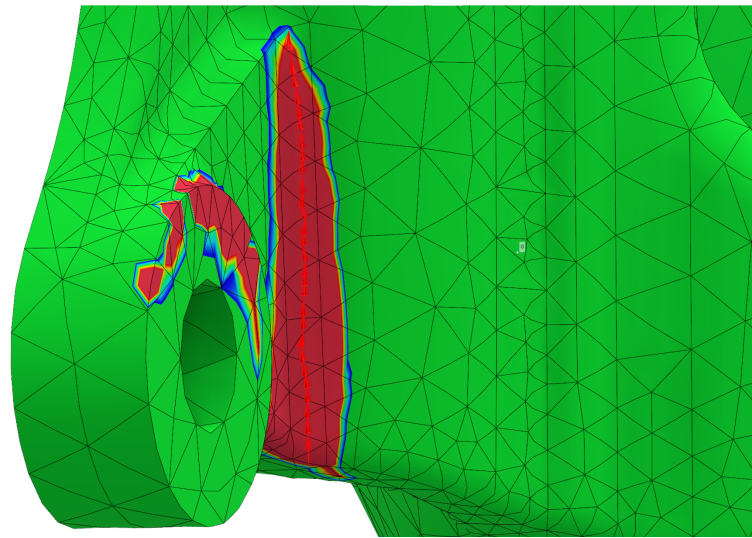


(b) PEEQ distribution along the battery cover path

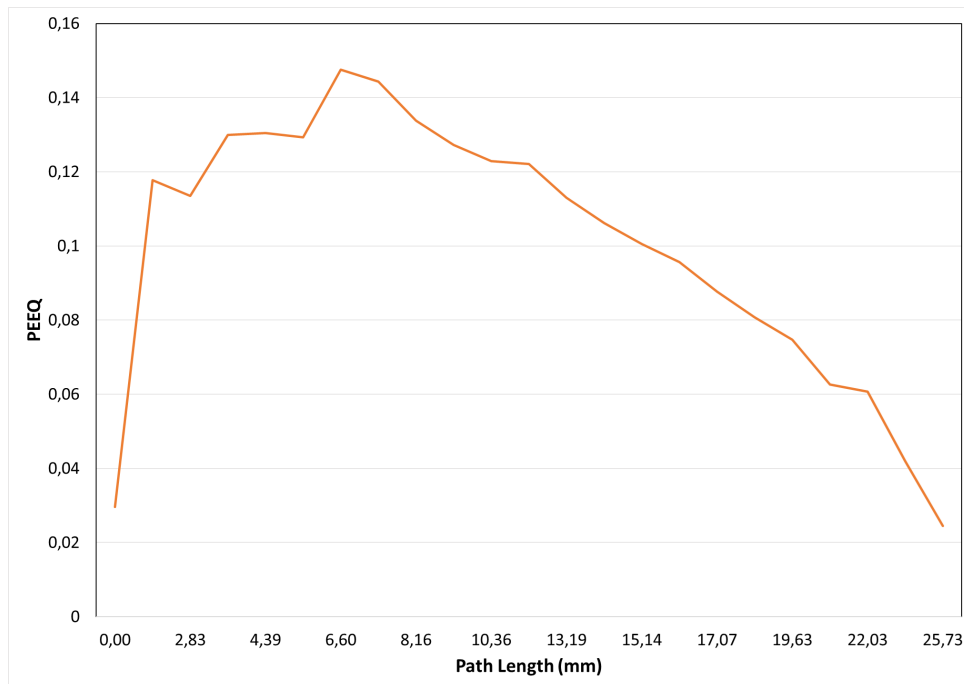
Figure 4.23: Path definition and equivalent plastic strain results in the battery cover region.

The plot confirms that the maximum plastic strain occurs near the center of the path, with a peak value of approximately 0.15. Although this value slightly exceeds the strain observed in uniaxial tensile tests, it does not necessarily indicate material failure. Under multiaxial stress conditions, localized strains may exceed those recorded in standardized tensile tests without causing rupture [7].

For the bottom bracket region, a separate path was defined along the midsection of the most critical area. Due to strain localization occurring between nodes, the path was constructed using discrete points to more accurately capture peak values. The selected path and the corresponding equivalent plastic strain results are shown in Figures 4.24a and 4.24b, respectively.



(a) Point-based path defined along the bottom bracket region



(b) PEEQ distribution along the bottom bracket path

Figure 4.24: Path definition and equivalent plastic strain results for the bottom bracket region.

The results show a clear concentration of plastic strain around the center of the bottom bracket path, confirming this area as one of the most critical in the structure. This behavior is consistent with the expected mechanical response of the frame during impact, as the bottom bracket is subjected to high stress concentrations due to its proximity with weights application and complex geometry with curves and fillets.

4.6.1 Summary

This chapter presented the finite element modeling approach and simulation results for the falling frame impact scenario. The geometry of the frame was introduced, and the material behavior was defined using a simplified plasticity model based on tensile test data and literature references. Key aspects such as boundary conditions, contact definitions, and solver settings were addressed to ensure numerical stability and physical consistency throughout the simulation.

The simulation outputs, including deformation, stress distribution, and plastic strain evolution, were analysed both globally and in detail for critical areas such as the battery cover and bottom bracket. Results confirmed expected behavior and revealed localized strain in regions typically prone to failure.

These findings indicate that the model accurately captures the essential mechanical response of the frame during impact and provides a sound foundation for further evaluation. The following chapter will focus on experimental testing and comparison with physical impact results to validate the simulation approach.

Chapter 5

Experimental Validation

This experimental test was conducted to obtain physical data for validating the numerical simulation presented in previous chapters. As discussed in Chapter 3, Section 3.4, the validation process is a fundamental step in FEA, enabling engineers to ensure the accuracy and reliability of simulated results. By correlating physical observations with the numerical response, confidence is gained in the predictive capabilities of the model.

The test conditions replicate those prescribed in ISO 4210-6:2023 for frame impact assessment [1], and were carried out using available components and equipment within the laboratory environment. The primary goal is to assess whether the simulated frame behavior under impact corresponds to the physical outcome, particularly in terms of deformation.

5.1 Test Setup and Procedure

The tested frame and mounted components are illustrated in Figure 5.1 and Figure 5.2, respectively. The steerer tube was attached to the headtube via two bearings, modeled in the simulation as rigid tie constraints. The bearings allow the steerer tube to be connected with the head tube and their main purpose is to rotate the wheel along the handlebar when riding the bike.



Figure 5.1: Frame model with dropouts mounted.



Figure 5.2: Components assembled on the frame.

At the rear end, the dropouts were fastened to the frame using two bolts, as highlighted in Figure 5.1 with a circle. While these connections are physically attached to the frame with bolts, they are modeled as continuous in the simulation since the CAD geometry includes the dropouts as part of the frame body. An additional bolt connects the seatstays to the dropout, also simplified in the simulation model through direct geometric continuity. These approaches are both valid as based on previous knowledge and experience these components never have shown problems.

The boundary conditions applied in the simulation replicate the experimental setup. A hinge constraint was used at the rear dropouts, as shown in Figure 5.3a, allowing rotation around the horizontal axle, which corresponds to the global z -axis in the simulation. All translational degrees of freedom were fully constrained. The steel anvil was fixed to the testing platform via four bolts and modeled using an *encastre* condition on its bottom face, restraining all degrees of freedom. Figure 5.3 presents the physical implementation of these constraints, showing both the hinge detail at the rear dropouts and the steel anvil plate used to secure the frame during testing.



(a) Hinge feature at rear dropouts



(b) Steel anvil plate fixture

Figure 5.3: Detail views of the rear dropout hinge (a) and the steel anvil plate fixture (b) used in the physical test setup.

Due to the absence of a crane or automated drop mechanism, a custom insert was used to simulate the 300 mm drop height (Figure 5.4). This insert includes a rope mechanism that allows manual release.



Figure 5.4: Drop insert length.

To track frame deformation, high-contrast markers were applied to specific surface locations for motion tracking highlighted in Figure 5.5.

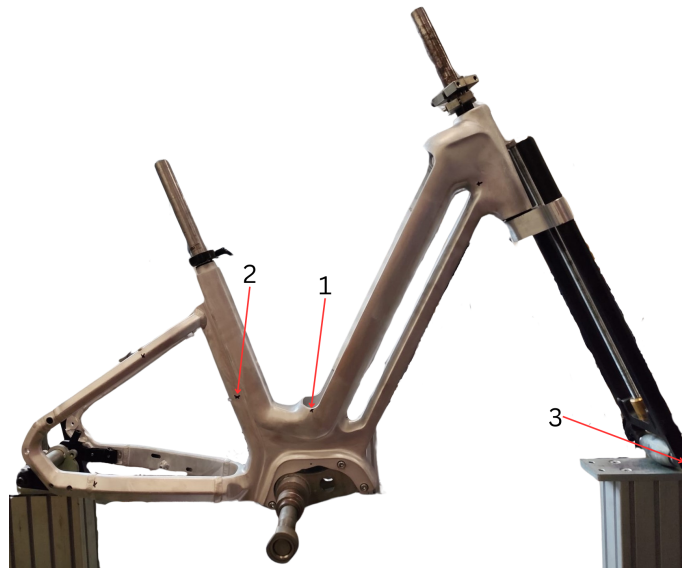


Figure 5.5: Frame with markers.

Displacement tracking was performed using the open-source software *Tracker* [45], commonly used in experimental physics for motion analysis. The main recording was captured using a consumer-grade digital video camera capable of slow-motion recording, positioned parallel to the fall plane to track in-plane deformation.

To evaluate potential out-of-plane movement, a second digital camera was positioned orthogonally to the impact direction. Although such motion was expected to remain within sub-millimeter

levels, this secondary view ensured that any out-of-plane displacement could be identified if present.

The complete camera setup is shown in Figure 5.6. The main camera alignment allowed accurate measurement of frame deformation in the plane of motion, while the side view offered additional spatial verification.

Before the final test, calibration trials were conducted to fine-tune camera positioning and exposure settings. These tests confirmed that the video resolution and frame rate were sufficient to capture the rapid deformation without motion blur or tracking loss.



(a) Main camera aligned with fall plane

(b) Side camera to detect out-of-plane motion

Figure 5.6: Camera setup used for displacement tracking.

5.2 Results

In accordance with ISO 4210-6:2023, two consecutive drop tests were performed. Visual inspection of the frame revealed no signs of fracture, crack initiation, or delamination. No paint disruption or permanent deformation was observed in the critical zones.

The displacement of the frame was evaluated in both the direction of impact and the orthogonal axis.

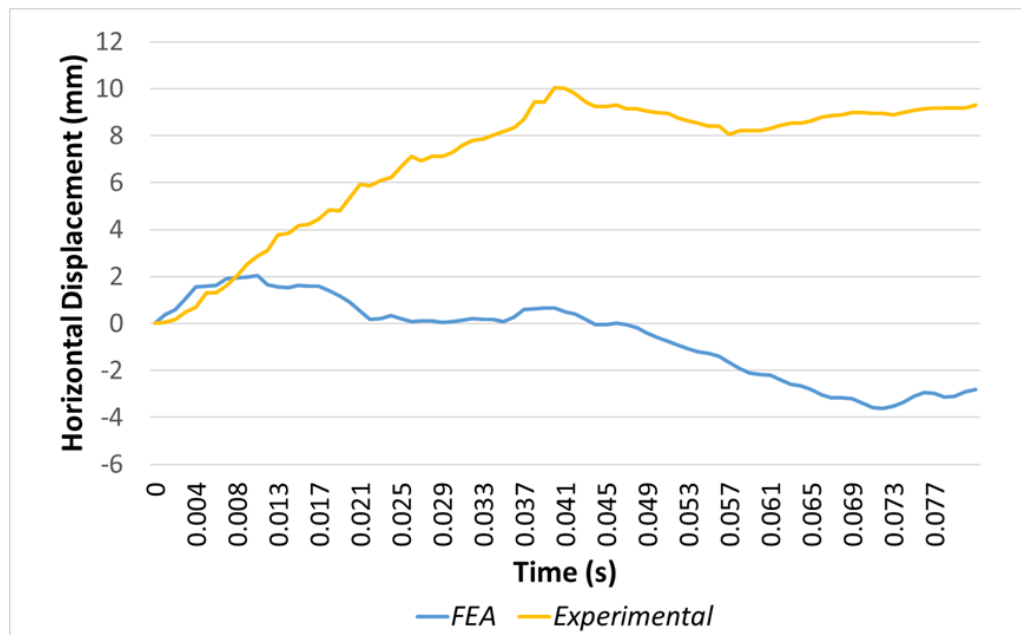
For the purpose of simulation–experiment comparison, the analysis focuses on the first 0.08 s of the initial fall, which corresponds to the total duration of the finite element simulation. This interval captures the initial contact between the roller and the steel plate, the rebound, and the upward motion of the frame, ending just before a possible secondary impact that is not modeled numerically. Pre-impact motion is excluded, as it is not explicitly simulated and does not contribute to the validation.

The experimental test was recorded at a resolution of 1280×720 pixels (720p) and 240 FPS (frames per second) and subsequently interpolated to 960 FPS, yielding a temporal resolution of

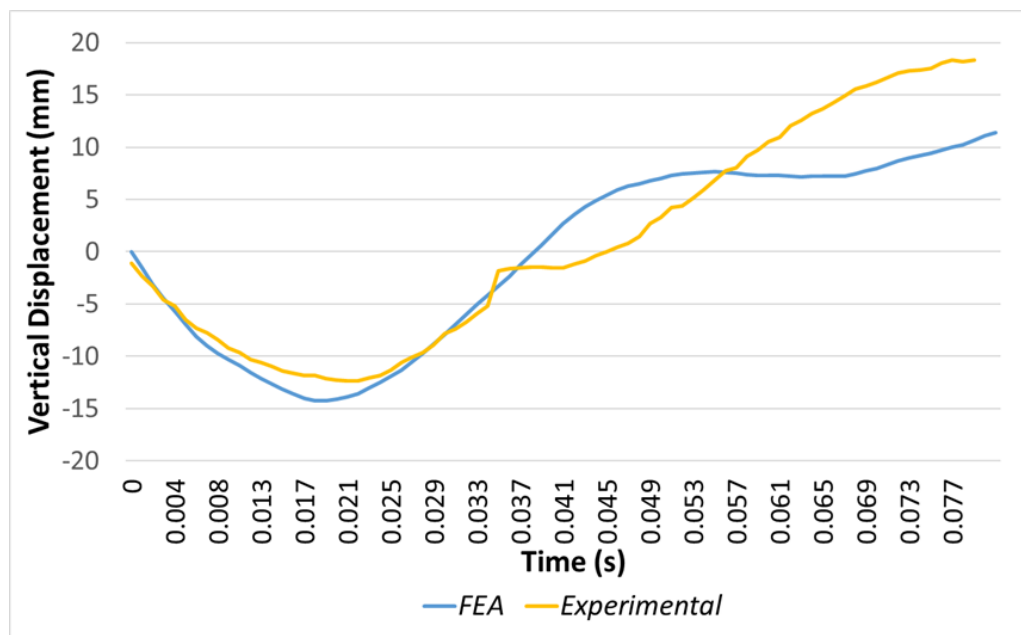
approximately 0.001 04 s. For consistency, both simulation and experimental data were evaluated at a common time step of 0.001 s, allowing accurate alignment throughout the comparison.

5.2.1 Area 1: Battery Cover Vicinity

Figure 5.7 presents the displacement results in the horizontal (x) and vertical (y) directions at the battery cover vicinity. The plots compare the experimental data with the FEA output, allowing the evaluation of the model's ability to replicate both deformation trends and peak displacement values in this region.



(a) Horizontal displacement at the battery cover (FEA vs experimental)



(b) Vertical displacement at the battery cover (FEA vs experimental)

Figure 5.7: Comparison of horizontal and vertical displacement between FEA and experimental tracking at the battery cover region.

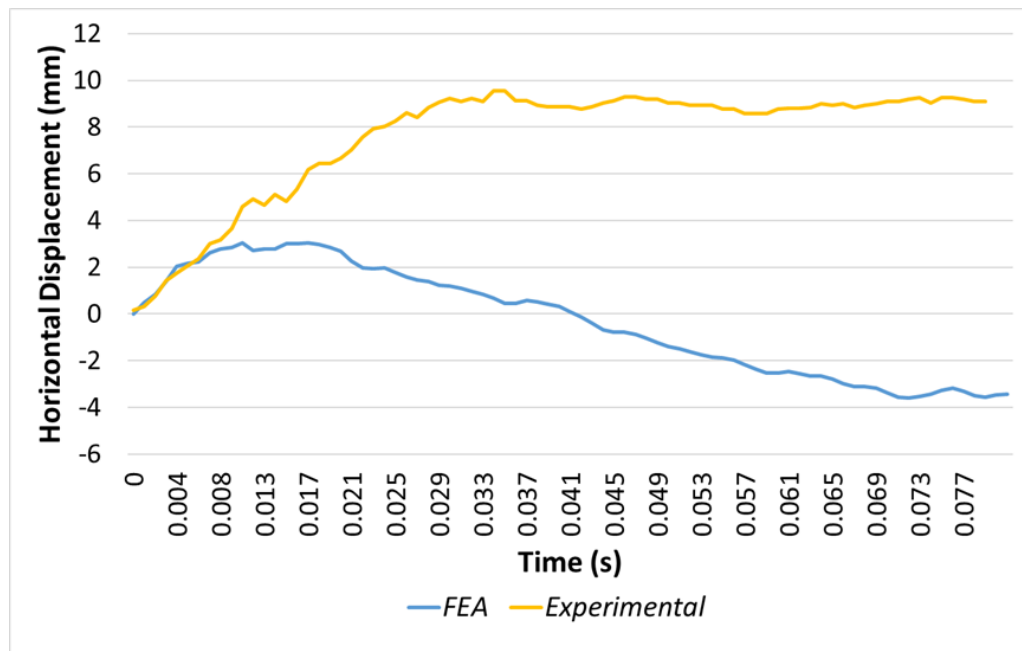
The horizontal displacement curve (Figure 5.7a) shows that the numerical model captures the initial deformation trend but exhibits a more pronounced backward motion after the peak, resulting in negative displacement values not observed in the experimental data. In the simulation, the frame rotates more freely after impact and moves past its initial horizontal position due to idealized constraints and the absence of friction. In contrast, the physical test shows greater horizontal

displacement at the battery cover, likely influenced by dynamic shifting of the suspended weights and a small forward shift of the test platform observed in high-speed footage. This platform movement, not present in the simulation, contributes to increased measured displacement and reduces the extent of backward motion recorded experimentally.

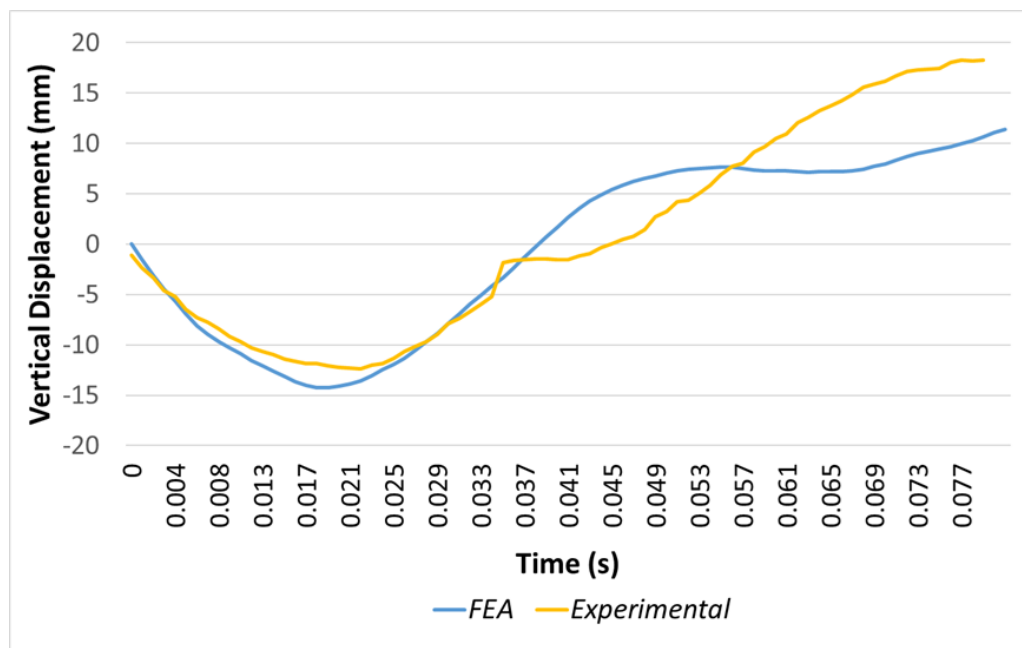
In the vertical direction (Figure 5.7b), both experimental and numerical results follow a similar trend, capturing the initial drop, rebound, and upward motion. However, the experimental curve slightly overshoots the simulated one after the rebound. This difference is likely due to the ongoing oscillation of the suspended weights after impact, which is not represented in the simulation where masses are idealized as static point loads.

5.2.2 Area 2: Seat Tube

Figure 5.8 presents the displacement results in the horizontal (x) and vertical (y) directions at the seat tube area. The top plot shows the horizontal displacement over time, while the bottom plot shows the vertical displacement.



(a) Horizontal displacement at the seat tube region (FEA vs experimental)



(b) Vertical displacement at the seat tube region (FEA vs experimental)

Figure 5.8: Comparison of horizontal and vertical displacement between FEA and experimental tracking at the seat tube region.

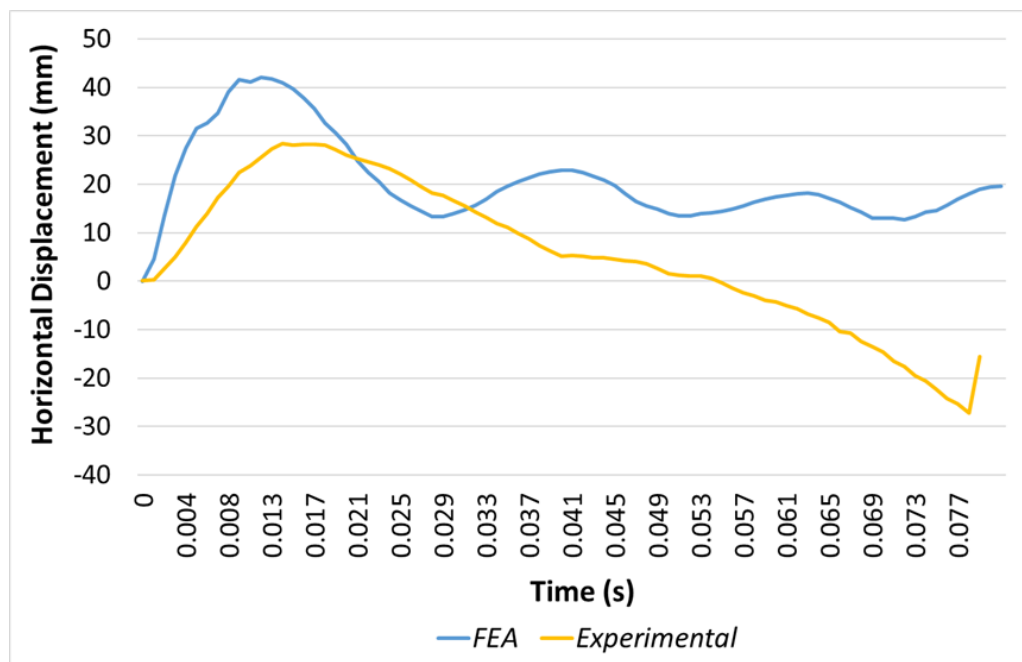
The horizontal displacement at the seat tube (Figure 5.8a) shows a similar trend to that observed at the battery cover (Figure 5.7a), with the experimental curve reaching significantly higher values than the simulation. This consistent behavior across two different points on the frame supports the previously discussed explanation related to the forward movement of the test platform.

The agreement in this trend reinforces the likelihood that the mismatch arises from physical factors not captured in the simulation rather than local modeling inaccuracies.

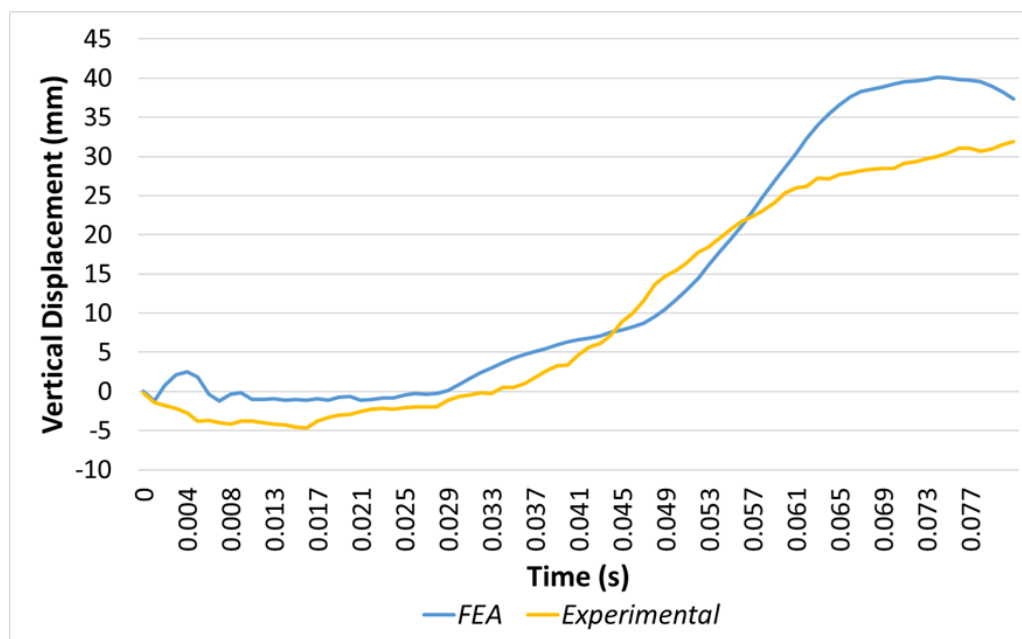
In the vertical direction (Figure 5.8b), the simulation captures the overall deformation trend well. However, the experimental displacement increases more sharply after the rebound. Slight inaccuracies in marker tracking or optical distortion in the high-speed footage could contribute to this difference, introducing small amplification effects in the measured vertical response.

5.2.3 Area 3: Roller

The same approach is applied to the roller, as shown in Figure 5.9.



(a) Horizontal displacement at the roller region (FEA vs experimental).



(b) Vertical displacement at the roller region (FEA vs experimental).

Figure 5.9: Comparison of horizontal and vertical displacement between FEA and experimental tracking at the roller region.

At the roller region (Figure 5.9a), the horizontal displacement shows a clear mismatch between simulation and experiment. The simulation produces a higher initial peak followed by noticeable oscillations, while the experimental data shows more limited forward motion and a continuous backward trend. Part of this discrepancy may result from the manual release method,

where pulling the rope induces a slight counter-rotation in the roller, opposing its forward motion. The absence of friction in the simulation contact definitions further exaggerates displacement, allowing the roller to move more freely upon impact. Additionally, high-speed footage reveals that in the physical test, the fork and head tube rise together as a rigid body, without visible oscillation. In contrast, the simulation displays a wave-like pattern, suggesting unintended relative movement at the steerer–headtube interface, which may also contribute to the overestimated horizontal displacement.

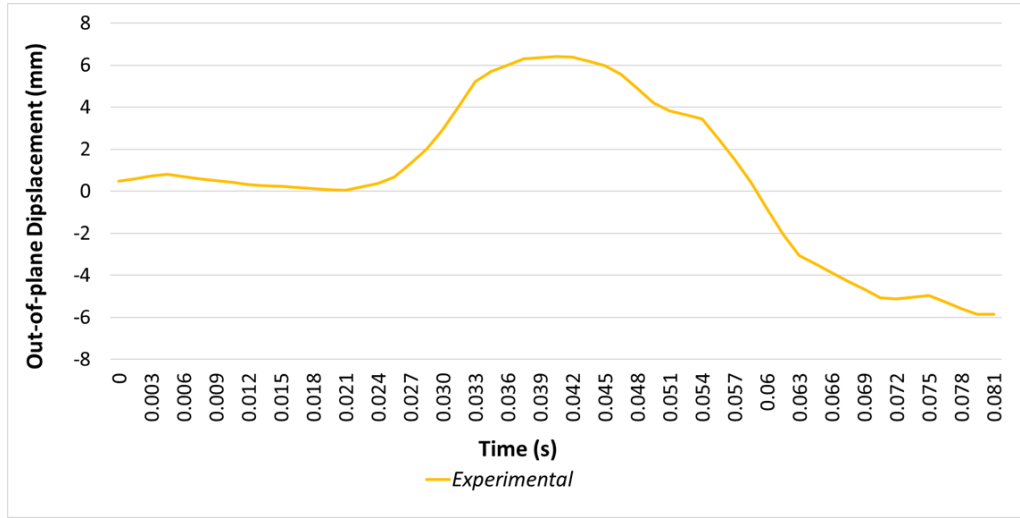
In the vertical direction (Figure 5.9b), both experimental and simulated curves show similar trends in upward displacement after impact. However, the experimental data also presents an initial downward displacement, which is absent in the simulation. This is likely due to the deformation of the test rig: although the steel anvil is rigidly fixed to the platform, the high impact force causes a small deflection that is captured by the tracking system. In the FEA model, the anvil is defined as fully fixed, which explains the absence of this initial downward response.

A slight mismatch is also observed in the timing of direction changes in the displacement curves, suggesting that either the simulation time alignment or experimental reference frame may require minor adjustments for improved synchronization.

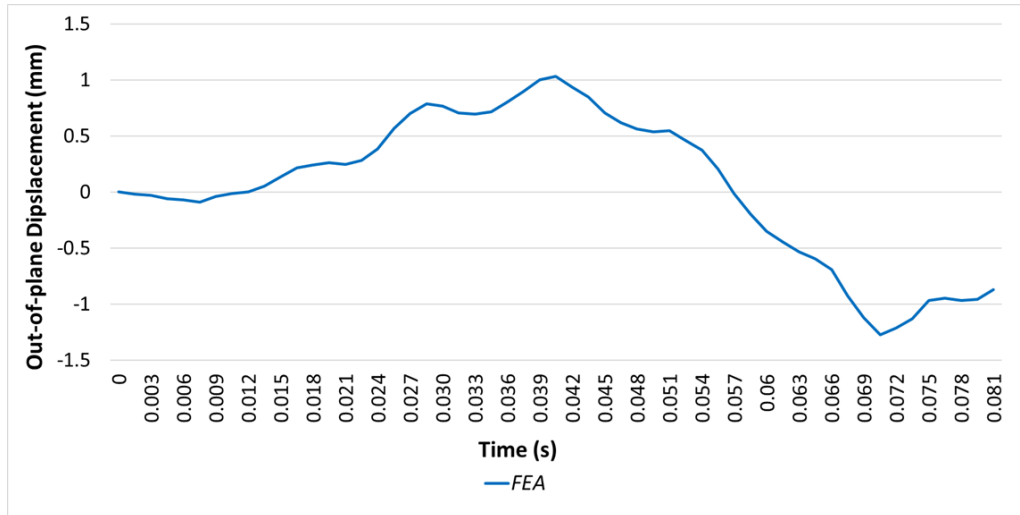
5.3 Out-of-Plane Motion Analysis

The out-of-plane motion was recorded using a smartphone camera at 240 FPS. While this provided reasonable tracking, it lacked the precision required for capturing displacements under 1 mm. As demonstrated in the graphs, the displacement remained minimal and fluctuated near zero, indicating that the frame exhibited stability with negligible out-of-plane motion. This aligns with the expected results from the FEA model.

Figures 5.10a and 5.10b display the out-of-plane displacement data. Figure 5.10a presents the experimental data, while Figure 5.10b shows the corresponding FEA results. Both confirm the minimal out-of-plane motion and the overall stability of the frame.



(a) Out-of-plane displacement (Experimental Data)



(b) Out-of-plane displacement (FEA Data)

Figure 5.10: Comparison of out-of-plane displacement between Experimental and FEA data

Due to limitations in resolution and frame rate, minor discrepancies between the experimental data and FEA results were observed, particularly in the scaling of displacement. To enhance the comparison, an interpolation curve was applied to align the experimental time data with the FEA time steps, improving the matching accuracy.

In summary, despite these limitations, the results show good agreement between the experimental and FEA data.

5.3.1 Quantitative Comparison

To complement the qualitative comparison between simulation and experimental displacements, the Root Mean Square Error (RMSE) was computed for each region and displacement direction. RMSE is a commonly used metric for quantifying the average magnitude of the error between

predicted and observed values. It is expressed in the same units as the data, in this case millimeters, and provides a direct measure of how closely the simulation matches the physical test.

Mathematically, RMSE is defined as:

$$\text{RMSE} = \sqrt{\frac{1}{n} \sum_{i=1}^n (x_i^{\text{sim}} - x_i^{\text{exp}})^2} \quad (5.1)$$

where x_i^{sim} and x_i^{exp} are the simulated and experimental displacement values at time step i , and n is the total number of time steps.

The calculation was performed over the entire simulation duration of 0.08 s, with both datasets sampled at uniform intervals of 0.001 s. Table 5.1 summarizes the RMSE values obtained for the three regions under analysis.

Table 5.1: RMSE values for displacement comparison (FEA vs experimental).

Region	RMSE _x (mm)	RMSE _y (mm)
Battery Cover	8.623	4.968
Seat Tube	8.97	4.72
Roller	19.104	4.9778

As shown in Table 5.1, the vertical displacement (Y-direction) displays consistently lower RMSE values across all regions, indicating reasonable agreement in capturing the upward motion and overall deformation trend. In contrast, the horizontal displacement (X-direction) shows higher discrepancies, particularly at the roller region, which reflects the previously discussed limitations such as: test setup movement, friction modeling and contact interactions.

It is important to note that the absolute displacement values involved are on the order of millimeters, meaning that even modest RMSE values represent a significant proportion of the measured response. Despite the deviations, the simulation successfully reproduces the overall shape and timing of the vertical displacement curves, demonstrating its capability to capture the dominant physical behavior of the frame under impact loading.

Following the test procedure, a visual inspection of the frame was conducted. No signs of crack initiation, delamination, or fracture were observed in any of the tested areas. Although no visible damage was detected, the simulation predicted localized plastic strains approaching 14%, which is above the upper elongation limit of Al 6061-T6 under standard tensile conditions. However, these reference values are derived from uniaxial, quasi-static tests and do not fully reflect the dynamic, multiaxial loading experienced during impact. Under such conditions, higher local plastic deformation can occur without leading to immediate macroscopic failure. The presence of micro-scale damage cannot be excluded and would require microscopic or non-destructive inspection techniques to confirm.

Chapter 6

Conclusion

This work concludes with a comprehensive evaluation of the project outcomes, combining experimental insights with numerical modeling to evaluate the behavior of a bicycle frame under dynamic impact loading. After addressing technical challenges, validating simulation results, and pinpointing critical areas for improvement, this chapter summarizes the primary findings and outlines directions for future research. The objective is not only to reflect upon the achieved outcomes but also to emphasize how this research provides a robust foundation for further developments in structural simulations within an industrial context.

6.1 Summary of Findings

This study investigated the dynamic behavior of a bicycle frame subjected to impact loading, following the ISO 4210-6:2023 guidelines. An in-house impact test was conducted and reproduced using explicit dynamic FEA. The simulation was validated by comparing horizontal and vertical displacement results at crucial frame regions: the battery cover vicinity, the seat tube, and the roller region.

The numerical model demonstrated good agreement with experimental results in the vertical direction, accurately capturing overall deformation patterns and rebound behavior. However, notable discrepancies arose in horizontal displacement, especially in areas sensitive to global rotation and test setup inconsistencies. These observations were quantitatively confirmed using RMSE, with significantly lower errors in the vertical direction and higher deviations horizontally.

Although visual inspections post-testing revealed no visible fractures, the simulation identified localized plastic strain values near critical levels. While no direct experimental strain measurements were performed, these simulation results highlight areas susceptible to local deformation, suggesting opportunities for design improvements such as smoother geometry or localized reinforcement. Any proposed modifications must align with manufacturing feasibility.

The analysis currently considers only the first of two consecutive impacts in the test procedure. However, the simulation framework established in this project supports restart analyses, enabling

the second impact to be modeled based on the final conditions of the first event. The validation of this initial impact event thus provides a reliable basis for further studies.

6.2 Future Work

Several avenues for future investigation are proposed to enhance the fidelity and predictive capability of the simulation:

- **Secondary Impact Modeling** - Extend simulations to include the second impact event, utilizing restart conditions from the initial validated simulation to assess cumulative deformation effects.
- **Influence of Damping and Contact Modeling** - Conduct further studies to quantify the precise influence of damping parameters on displacement outcomes, aiming to establish optimized damping settings for future analyses. Additionally, evaluate the role of frictional contact between the roller and the impact surface to understand its effect on energy dissipation and deformation behavior.
- **Experimental Improvements** - Reproduce impact tests using a rigidly fixed platform to eliminate movement and associated noise in displacement tracking. Implementing full-frame high-speed tracking will improve displacement data resolution and accuracy.
- **Connection Behavior Investigation** - Further investigate and refine modeling of the steerer tube and head tube interface to eliminate unintended oscillations observed in the numerical results, enhancing simulated frame responses.
- **Plastic Strain Validation** - Employ experimental strain measurement methods, such as Digital Image Correlation (DIC), to validate simulated plastic strains at critical locations.
- **Computational Efficiency** - Improve computational efficiency through selective mesh coarsening in low-strain regions and geometric simplifications, reducing simulation runtimes and increasing practicality for industrial use. Additionally, utilizing hardware with greater GPU processing capabilities can significantly accelerate explicit dynamic simulations.
- **Failure Benchmarking** - Conduct physical tests with incrementally increased loading conditions to approach actual failure thresholds. Correlating observed damage with critical plastic strain predictions from simulations will provide essential data for reliable failure prediction.
- **Validation through Simple Models** - Adopt initial validation strategies using simplified models and corresponding physical experiments for critical features such as material properties and connection modeling. This incremental validation will build confidence in simulation methodologies before tackling more complex real-world scenarios.

Once the accuracy of displacement response, damping behavior, friction modeling, and local strain predictions has been established through the measures outlined above, future work should incorporate advanced damage and failure criteria into the simulation. This includes implementing material degradation models or failure indicators to predict crack initiation and structural failure. These enhancements are essential for extending the model's applicability to more severe loading conditions, repeated impact scenarios, and durability assessments across the frame's service life.

Overall, this project successfully combined experimental testing with validated numerical simulations to assess the structural behavior of a bicycle frame under impact loading. The workflow established through this study provides a practical and scalable foundation for integrating simulation into the product development process. As Gazelle continues to digitize its engineering workflows, the methods and insights developed here can support more efficient design iterations, deeper structural evaluations, and greater confidence in early-stage virtual testing.

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