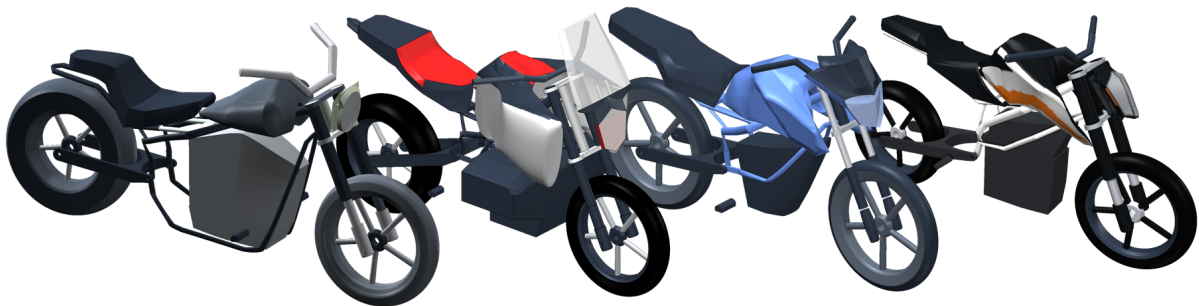




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Development of a generic morphable motorcycle model for crash analysis

Wilmer Helland
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DEPARTMENT OF MECHANICS AND MARITIME SCIENCES

CHALMERS UNIVERSITY OF TECHNOLOGY
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MASTER'S THESIS 2026

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Department of Mechanics and Maritime Sciences
CHALMERS UNIVERSITY OF TECHNOLOGY
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Cover: All morphed motorcycle models next to each other.

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Abstract

There is currently a lack of openly available finite element motorcycle (MC) models suitable for crash analysis. This thesis has aimed to address this problem by creating a generic, morphable MC that can be adapted to geometrically represent a range of commonly encountered motorcycles on the road and reproduce their characteristic behaviour during crash simulations. Additionally, the goal is to make the model open source, allowing users to access, morph, modify, and share it freely.

As part of this project, a base model was morphed into four different MC presets: the Harley-Davidson Sport Glide, BMW GS1300, Honda CB125F, and KTM Duke 390. These models were calibrated against physical test data obtained from frontal impacts with a rigid wall (RW), this was done by tuning the structural stiffness of the components and by tuning the spring-damper characteristics of the front suspension. In the physical tests, the RW was angled at 10° for the Harley-Davidson Sport Glide, BMW GS1300, and Honda CB125F, and at 0° for the KTM Duke 390. All MCs travelled at 40 km/h upon impact.

After the calibration was completed, the KTM Duke 390 model was validated against a physical test of a KTM Duke 390 with a post-mortem human subject (PMHS) impacting the side of a Honda Accord. This was replicated in the simulation using a human body model (HBM), a Honda Accord car model (National Highway Traffic Safety Administration, n.d.), and the morphed KTM Duke 390 model. In the physical test, the car was angled at 30° , and the motorcycle had a velocity of 50 km/h upon impact (Carrol, Bolte, 2024).

The calibration results show that, in many cases, the overall crash response of the bike is captured. Moreover, the validation shows good agreement with the physical test data, where the resultant behaviour of the MC matches the physical test.

The results from the calibration and validation show that a simplified model, starting from a single generic model, can be morphed into specific MC models that replicate important behaviours crucial for crash outcomes and enable MC crash analysis using human surrogates.

Keywords: Morphable, Motorcycle, LS-DYNA, FEM, PTW, Generic, FE-model, Frontal impact, Crash analysis, Explicit FE.

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List of Acronyms

Below is the list of acronyms that have been used throughout this thesis listed in alphabetical order:

FEM	Finite Element Method
FE	Finite Element
MC	Motorcycle
PTW	Powered Two Wheeler
COG	Center of Gravity
CFC	Channel Frequency Class
RW	Rigid Wall
HBM	Human Body Model
PMHS	Post Mortem Human Subject

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1

Introduction

1.1 Background

Motorcycles (MCs) have historically received less priority than cars in the development of safety measures (International Transport Forum, 2015). Additionally, the biomechanical analysis of motorcycle crashes can be considered complex because the rider and the motorcycle typically separate during an accident and subsequently follow different trajectories (Lloyd, 2016). This differs from passenger cars, where occupants remain within an enclosed vehicle structure and interact with passive safety systems such as seatbelts. Crash scenarios in which riders are separated from motorcycles along variable trajectories (Giovannini et al., 2024) further highlight the need for extensive simulation and testing to support the development of effective safety measures. This is further motivated by the high fatality rate of MC riders in relation to the fatalities of car riders. According to the U.S. Department of NHTSA (National Highway Traffic Safety Administration), the fatality rate of motorcyclists was 31.39 per 100 million vehicle miles which is nearly 28 times higher than that of car occupants based on traffic crash data from 2023. Furthermore, in the same year, 15% of total traffic fatalities in the U.S. were represented by motorcyclists (National Highway Traffic Safety Administration, 2025). The disproportionately high fatality rate for MC riders compared to car occupants highlights the need for improved tools to understand motorcycle crash dynamics and evaluate safety measures, where virtual MC models offer one such tool.

The development of virtual MC models for crash analysis has long been dominated by multi-rigid-body models, with the advantages being their low simulation times. Now, because of geometrical accuracy and more accurate deformation during impact, along with advances in computational power, FE-based simulation tools have become more robust and are widely used in vehicle crash and occupant safety analysis (Schulz et al., 2016). Through an extensive review of available sources, it was found that the number of currently available MC FE models is relatively small, and the number of open source models, as a result, being even smaller.

1.2 Purpose

Due to the current lack of openly available FE MC models suitable for crash analysis, a gap exists that limits research and development in MC safety. The purpose is therefore to close this gap by creating a generic, morphable MC model capable of

representing a range of common MCs sold today. Closing this gap would increase the accessibility of MC safety development, reduce the need of physical testing, and potentially accelerate MC safety improvements.

1.3 Goals

The objective of the project is to create an open-source, generic morphable FE-model of an MC that allows for geometric variability between MC models. The FE-model should be representative of real MCs in frontal crash scenarios in order to allow for injury risk assessment of an MC rider.

1.4 Limitations

The project was limited to include four different MCs. Prior to the start of the project, four MCs were selected capturing variation in geometry found among commonly sold MC types.

Physical tests were conducted before the start of the project to obtain data for the calibration of the FE models. The calibration data will therefore be limited to these load cases.

Validation is limited to the KTM Duke 390 only, as independent validation tests for the remaining three MCs could not be performed within the time constraints of this project.

Since the FE models are intended to simulate frontal crashes involving a rider, the front sections of the models will be prioritized. The sections that interact with the rider will also be prioritized, as these have been shown to influence rider kinematics during a crash (Lundin, Iraeus, Pipkorn, 2023).

The FE-model has been limited to not model material failure in order to reduce complexity.

2

Theory

The following chapter aims to provide a theoretical foundation for understanding the project. The theory introduces FEM, the solver LS-DYNA, different element types, element quality, MC models and properties specific to certain MC classes, the impact sequence for MCs, relevant material behaviour, and filtering methods for simulation results.

2.1 Finite element method

Partial differential equations (PDEs) can be used to describe the laws of physics in space and time. These PDEs can be rather complex and, therefore, cannot be solved analytically, requiring them to be discretized in order to approximate the solution. The Finite Element Method (FEM) can be used to approximately solve these equations and is an essential tool in the design process of many types of structures (COMSOL, 2016). It allows engineers to evaluate structures under different loading conditions before they are constructed (Hefmec, 2026). For example, FEM can be used to assess the effects of a short-duration event, such as an MC crash, where the structural response can exhibit complex non-linear behaviour (Ansys, n.d.).

FEM divides geometries into smaller regions called elements, which are defined by their nodes. The solution for each node can differ depending on the material properties or, for example, the size of the element. The constellation of elements and nodes over the geometry together forms what is called a mesh. The mesh can be adjusted depending on the application, and a finer mesh (relatively many elements) generally yields better approximations than a coarse mesh (relatively few elements). The mesh size mainly involves balancing computer resources with sufficient result output (COMSOL, 2016).

2.1.1 Explicit and Implicit FEM

FEM has different approaches regarding whether the solution can be considered time-independent or not. These approaches are called implicit and explicit FEM, where the calculation methods differ with respect to time. This is illustrated below in equations 2.1 and 2.2 (Ansys Learning, 2021).

$$\text{Explicit: } y(t_{n+1}) = y(t_n) + \Delta \cdot \dot{y}(t_n) \quad (2.1)$$

$$\text{Implicit: } y(t_{n+1}) = y(t_n) + \Delta \cdot \dot{y}(t_{n+1}) \quad (2.2)$$

The explicit method is a time-dependent approach and, in relation to implicit FEM, uses smaller time steps that are directly scaled by the size of the smallest element in the mesh. This is illustrated by the Courant-Friedrichs-Lewy (CFL) condition, which is defined below in Equation 2.3, where h is the characteristic length of a finite element, c is the speed of sound in the material, f is the safety factor, and Δt is the time step.

$$\Delta t \leq f \cdot \left[\frac{h}{c} \right] \quad (2.3)$$

The propagation of sound is, in turn, dependent on Young's modulus and the density of the material, which is illustrated below in Equation 2.4. Since the propagation of sound is directly dependent on the mass, which, in turn, is dependent on the density, some explicit solvers allow for mass scaling to compensate for too large a time step. Mass scaling increases an element's mass which, if not controlled, can lead to non-physical results (ANSYS, Inc., 2020).

$$c = \sqrt{\frac{E}{\rho}} \quad (2.4)$$

Explicit FEM is used when the dynamic effects of a system, such as acceleration, can be considered to have a considerable effect on the outcome. This applies to sudden dynamic events, of which motorcycle crashes are one example (SimScale, 2024).

2.1.2 LS-DYNA

LS-DYNA is used in this project as the explicit solver together with the preprocessor ANSA (BETA CAE Systems, 2026a), which provides an interface for model definition. LS-DYNA's extensive library of elements, contact formulations, material models, and other features can be used to define the simulation environment for models exhibiting complex behaviour. Among its many applications, LS-DYNA supports airbag simulations, impact analysis, and vehicle crash simulations (Ansys, 2026b). Due to its broad range of capabilities, LS-DYNA is widely used in the automotive industry for crash simulations and the evaluation of different vehicle designs (Ansys, 2026c).

2.1.2.1 Constraints

Constraints can be defined through the ANSA user interface to, among other things, model kinematic constraints between parts (Kolokythas and Rorris, 2011). The constraints used in LS-DYNA for this project are described below, together with their corresponding LS-DYNA definitions, which are provided in parentheses (Ansys, 2026d).

- Nodal rigid body (CONSTRAINED_NODAL_RIGID_BODY), used to connect two deformable parts with a rigid connection.

- Constrained extra nodes (`CONSTRAINED_EXTRA_NODES`), used to connect a rigid part to a deformable part with a rigid connection.
- Revolute joint (`CONSTRAINED_JOINT_REVOLUTE`), limiting displacement in all directions and allowing rotation about one axis. This is used to connect rigid parts that are to rotate around a specific axis.
- Cylindrical joint (`CONSTRAINED_JOINT_CYLINDRICAL`), where the movement of two rigid parts is limited to rotation and translation around and along one axis.
- Rigid bodies (`CONSTRAINED_RIGID_BODIES`), used to connect one rigid part to another rigid part with a rigid connection.

2.1.3 Shell elements

Shell elements can be used to model thin structures, meaning that one dimension is considerably smaller than the other two. The deformation in the shell surface and in the normal direction is decoupled, enabling a simple and efficient representation of thin structures (Ansys, 2026a).

The thickness of a shell can be adjusted in the LS-DYNA representation of the shell section. This change in thickness can, in turn, have a large influence on the stiffness of the shell, provided that the membrane strains are large enough. The thickness is represented by a specified number of integration points (NIPs), which are adjusted through the shell section definition in LS-DYNA, where the NIPs are based on the material definition and the simulation type (Stelzmann, 2010).

2.1.4 Beam elements

In comparison to shell elements, beam elements are one-dimensional, making them ideal for long, elongated structures, where the length is much larger than the other two dimensions. These elements typically assume a linear stress distribution across the cross section (Poletto, 2024).

2.1.5 Solid elements

Compared to beams and shells, solid elements are three-dimensional, which makes them particularly suitable for modelling bulk structures. The properties in all three directions allow for a more detailed resolution of the stress and strain fields within the structure itself (Poletto, 2024). For this project, solid elements have been used to model the bulk content of the MC saddle, which consists of the foam.

2.1.6 Element quality

There are many metrics by which element quality is determined, and these should ideally be met. The most common metrics are as follows: aspect ratio, Jacobian,

skewness, and warping. There are also metrics called minimum length and mass scaling. The minimum length is set explicitly by the user, while the mass scaling criterion is determined by the user's choice of the time step. Element quality impacts the simulation in three main ways: accuracy, efficiency, and stability. ANSA's default PATRAN, (Cadence, 2026), thresholds for each criterion can be seen in Table 2.1.

Table 2.1: Default ANSA criteria thresholds for shells and solids.

Criteria thresholds				
	Aspect ratio	Jacobian	Skewness	Warping
Shells	3	0.7	30	7
Solids	3	0.7	45	10

Element aspect ratio is the measure of the ratio between the element's shortest edge and longest edge. An element with an aspect ratio that is too large could lead to numerical errors and inaccuracies, particularly when calculating the element's stress and strain.

Element Jacobian is a measure that determines how accurately an element's idealized coordinates can be mapped to the actual coordinates on the mesh. This value should be as close to one as possible and should ideally not be too small or negative, as small or negative values could impact the accuracy and stability of the simulation.

Element skewness is a measure of how close the element's internal angles are to the ideal internal angles: 90 degrees for quadrilateral elements and 60 degrees for triangular elements. Skewed elements could lead to interpolation errors or uneven stress distributions across the element.

Element warping is caused by the nodes of the element not lying on the same plane and is thus a measure of how close the nodes are to this plane. Warped elements might have inaccurate stresses because of inaccurate interpolation (Yurii Shumak, 2025).

The minimum length of an element is set in order to avoid generating elements that are too small. Elements that are too small can cause instability and long run-times. This metric works together with the mass-scaling metric, since elements that are small, have low density, or have a high Young's modulus might become mass-scaled if the time step is too small to account for these factors. This can also be seen from Equation 2.4.

Examples of elements with the above-mentioned issues can be seen, colour-coded, in Figures 2.1–2.7. It should also be noted that an element might fail multiple metrics but only be assigned one colour.



Figure 2.1:
Ideal element.



Figure 2.2:
Element with aspect ratio issue.



Figure 2.3:
Element with Jacobian issue.



Figure 2.4:
Element with skewness issue.

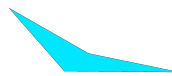


Figure 2.5:
Element with warping issue.



Figure 2.6:
Element with minimum length issue.



Figure 2.7:
Element with mass-scaling issue.

2.2 Motorcycle models

There are several types of motorcycles, each with distinct characteristics in terms of geometry, materials, and mass. Examples include cruiser, adventure, naked, and commuter motorcycles. In this section, the project-specific MC types will be introduced, as well as fundamental theory about fork construction, overall behaviour, and design aspects crucial to the behaviour of an MC.

2.2.1 Front fork setup

The choice of front fork setup plays a crucial role in the dynamics of a motorcycle, including chassis dynamics, handling, steering, and overall stability. Additionally, the forks contribute to the overall stability of the motorcycle in different settings by absorbing shocks, allowing the motorcycle to remain connected to the ground, and improving rider comfort (Öhlins, n.d.). Furthermore, the front fork acts as the load path between the wheel and the frame, transferring load from the wheel to the frame (Tan et al., 2016). These examples illustrate some of the functions of the front fork.

In commercial motorcycles, there are mainly two types of fork designs: inverted/USD (upside-down forks) and telescopic forks. Telescopic forks are designed with inner and outer tubes, where the spring and damper are situated inside. The

inner tube is located on the upper part of the fork and is attached to the triple clamps, with the thicker tubes at the bottom. An inverted fork works in the opposite way, with its thicker tube located at the top and the thinner tube at the bottom. The difference in performance comes from the location of the thicker tubes, where placement at the top for inverted forks allows for a more stable base and, therefore, more precise steering. An example of an inverted fork can be seen in Figure 2.8, and an example of a telescopic fork can be seen in Figure 2.9.

The downside of using inverted forks is often their higher cost compared to conventional forks (telescopic), as well as the fact that they are often harder to maintain over time. The telescopic fork setup is more suited for flat terrain, such as paved roads with occasional bumps, whereas inverted forks are more performance-oriented and better suited for aggressive riding, rough terrain, or pushing the bike hard (TFX Suspension, n.d.).



Figure 2.8: Inverted forks Harley-Davidson.



Figure 2.9: Telescopic forks Honda.

2.2.2 Design of the Center Of Gravity (COG)

The dynamic behaviour of a motorcycle is largely dependent on the location of its COG. This location, in turn, depends on the mass distribution among the motorcycle's different components. A major contributor to the total mass is the engine, which tends to be the heaviest component of the motorcycle. This means that the location of the COG largely depends on the mass and position of the engine. It is also important to keep in mind that the rider plays a crucial role in these scenarios, as a shift in the rider's position can result in a substantial change in the location of the combined COG, depending on the direction and magnitude of the movement.

Much of the dynamic behaviour of a motorcycle occurs when its velocity changes, such as during acceleration or braking. During braking or crashes, the height of the COG relative to the ground plays a crucial role in rider kinematics. A higher COG distributes a higher load on the front wheel, which improves retardation but also increases the risk of a flip-over. This is also largely dependent on the longitudinal position of the COG, where, for instance, a COG located closer to the front further increases the risk of a flip-over (M C (MCMX), 2026).

2.2.3 The moments of inertia

The dynamic behaviour, although largely dependent on the location of the COG, as previously stated, also depends on the moments of inertia. Some of the most important moments of inertia in the case of a motorcycle are the roll, pitch, and yaw moments of the main frame (M C (MCMX), 2026).

2.2.4 Cruiser

Cruiser bikes are designed to provide a comfortable ride, with a relatively low saddle height. In addition to comfort, this type of bike is designed with a long wheelbase, meaning a large distance between the center points of the front and rear wheel axes, and relatively low ground clearance compared to other bikes. This also results in a relatively low COG. The relatively low COG improves stability while riding. Similarly, cruiser bikes are usually relatively heavy and have wider wheels, which maintain better stability between the ground and the wheels (Viking Bags, 2025). The cruiser bike, a Harley-Davidson Sport Glide, used for this project can be seen below in Figure 2.10.



Figure 2.10: The Harley-Davidson Sport Glide, picture taken from Autoliv.

2.2.5 Sport/naked

Sport bikes are designed for performance, with high top speeds and efficient aerodynamics, where the fairings (the outer body panels) are one example of a design intended to improve aerodynamics.

Naked bikes can be considered a stripped-down version of a sport bike, where the major difference lies in the fact that the naked style minimize fairings. Most of the motorcycle parts are also exposed, which is done in order to reduce the weight and improve the power-to-weight ratio and overall performance (Viking Bags, 2025). The naked MC, a KTM Duke 390, can be seen below in Figure 2.11, and is the MC used for this project.



Figure 2.11: The KTM Duke 390, picture taken from Autoliv.

2.2.6 Commuter

A commuter motorcycle is adjusted for conditions often encountered when commuting in day-to-day life. They are often made to be driven on paved roads, and the design is focused on comfort and other factors that matter, such as fuel efficiency. The rider is often situated like a driver in a car, with an upright posture. The engines often do not come close to the power and size of other types, such as adventure, sport, and naked MCs. These motorcycles are also often lightweight in order to keep fuel consumption low. In Figure 2.12, a picture of a Honda CB125F can be seen, which is a commuter bike and part of this project.



Figure 2.12: The Honda CB125F, picture taken from Autoliv.

2.2.7 Adventure (ADV)

ADV bikes are relatively versatile compared to the other types previously mentioned, as these bikes allow for good performance both on- and off-road and are considered to be among the more versatile bikes. ADV bikes have high ground clearance, with a suspension that allows the wheel to travel a relatively long distance compared to

other bikes. The ADV is also a heavier option than most other off-road bikes (Viking Bags, 2025). In Figure 2.13, a BMW GS1300 can be seen, which is an adventure MC and part of this project.



Figure 2.13: The BMW GS1300, picture taken from Autoliv.

2.3 Impact sequence for MCs

The frontal impact of an MC into an object can often be split into three sequences: initial contact with the object, rebound of the front wheel into the engine, radiator, or frame, and finally impact with the object caused by the engine forcing the wheel back into the object after the rebound (Rose, 2022). The sequence is illustrated below in Figure 2.14.

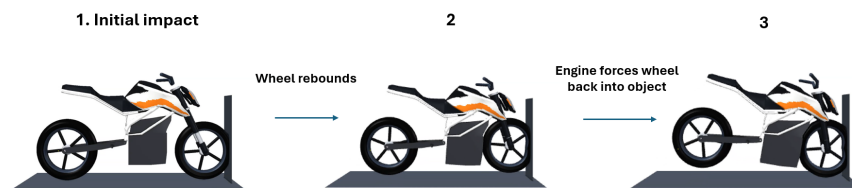


Figure 2.14: Typical frontal impact sequence for MCs.

2.4 Behaviour of materials

In this section, the project-specific material behaviour, and the modelling of these materials in LS-DYNA are introduced.

2.4.1 Elasto-plastic materials

An elasto-plastic material accounts for both the elastic and plastic behaviour of the material, with the transition between the two occurring at the yield point. After the yield point, the behaviour is no longer linear, and the material begins to undergo strain hardening until it reaches its ultimate strength and eventually fracture occurs (SIMSCALE, 2026).

This is illustrated below in Figure 2.15. The slope of the elastic region is described by the material's Young's modulus, while the non-linear behaviour between the yield stress and the ultimate stress is often simplified as the tangential modulus to describe the slope after the yield point.

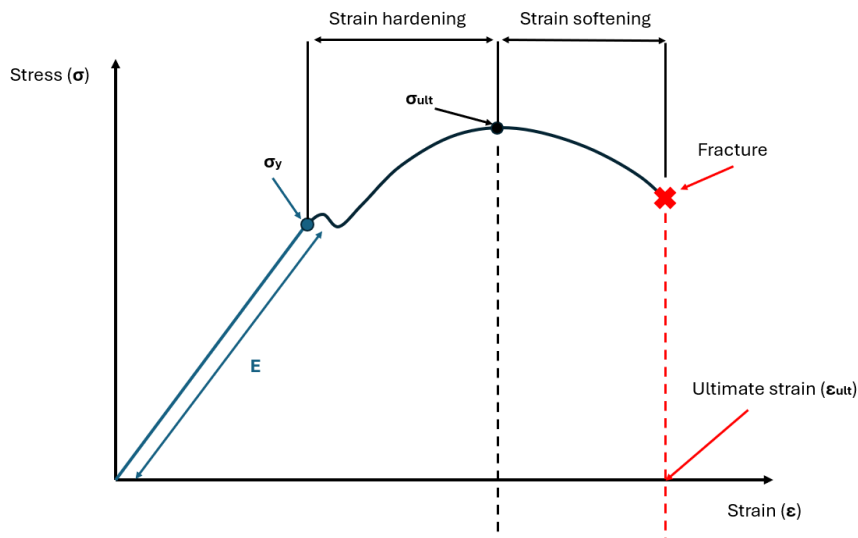


Figure 2.15: Engineering stress-strain curve for an elasto-plastic material.

2.4.2 Polymers

Some components of the MC are plastic polymers, such as the wind shield and saddle foam.

Polymers are long chains consisting of hydrocarbon molecules structured into smaller units called monomers. These structures give polymers a broad range of properties, one example being strength. The different properties allow polymers to be relatively versatile in their applications compared to many other materials.

Some of the most common types of polymers are natural polymers, synthetic polymers, thermoplastics, thermosetting polymers, and elastomers. The application of polymers in different settings often requires additional enhancement, which can be achieved through the use of plastic additives (EVERGREEN CHEMICAL, 2025) (WALKERRUBBER LTD, 2026).

2.4.3 Rubber

Rubber is used for the tires of the MC's wheels. Rubber is a polymer and belongs to the subclass of polymers known as elastomers. Because of elastomers' molecular chains, they can become twisted or straightened under tensile load, but return to their original state when the load is removed. Because the overall volume changes very little during tension or compression, they are close to incompressible. As a result of their behaviour and properties, they exhibit non-linear behaviour (Peksen, 2018). The behaviour itself is largely time-dependent, where the strain rate correlates strongly with the stiffness of the rubber, and may exhibit viscoelastic behaviour under certain loading conditions (Håkansson, 2000).

Rubber performs especially well in contexts where the focus lies on shock absorption and vibration absorption. Its ability to undergo large deformations without significant plasticity further emphasises its practical applications (Håkansson, 2000).

2.4.3.1 Rubber-modelling in LS-DYNA

There are different modelling strategies for rubber in LS-DYNA, where some of the models are the following: Ogden, Neo-Hooke, Mooney-Rivlin, and Arruda-Boyce. For this project, the Mooney-Rivlin model has been used consistently across models and will be presented below (Jonsson, 2020).

2.4.3.2 Mooney-Rivlin

The Mooney-Rivlin model is a popular choice when it comes to modelling the hyperelastic behaviour of rubber-like materials under large deformations. This model assumes that the material in question is isotropic and that the calculated deformation energy is determined solely by the strain (Fathi et al., 2024).

The strain-energy density function for Mooney-Rivlin can be seen below in Equation 2.5.

$$W = C_1 (I_1 - 3) + C_2 (I_2 - 3) \quad (2.5)$$

The constants C_1 and C_2 are material constants, and the variables I_1 and I_2 are strain invariants, which are written as a decomposition of the principal stretch ratios λ_1 , λ_2 , and λ_3 . These are illustrated below in Equations 2.6 and 2.7.

$$I_1 = \lambda_1^2 + \lambda_2^2 + \lambda_3^2 \quad (2.6)$$

$$I_2 = \lambda_1^2 \lambda_2^2 + \lambda_2^2 \lambda_3^2 + \lambda_3^2 \lambda_1^2 \quad (2.7)$$

Rubber is, as previously stated, considered to be incompressible, which the Mooney-Rivlin model captures through the criterion for incompressibility illustrated below in Equation 2.8 (Feng and Hallquist, 2026).

$$\lambda_1 \lambda_2 \lambda_3 = 1 \quad (2.8)$$

Given that the Mooney-Rivlin model performs particularly well under large deformations compared to other models, it is widely used in tire modelling (MACHE et al., 2026).

2.5 Filters for data

Data can, in many cases, be noisy and hence require post-processing to make it interpretable for comparison with other results. Filtering data is one type of post-processing that helps refine data sets and improve reliability and usability. A filter can be applied mainly with regard to three different aspects: noise reduction, relevance filtering, and data smoothing and transformation. Based on the data-processing method, different types of filters can be applied, one of which is frequency-based filtering. Types of filters using this method include low-pass, high-pass, bandpass, and bandstop (notch) filters (MathWorks, 2026).

In Figure 2.16, an example of the difference between unfiltered and filtered acceleration data using a CFC60 filter can be seen. A CFC (Channel Frequency Class) filter is a low-pass filter, and the frequency class of 60 is especially suitable for filtering acceleration data from vehicle simulations (DSP Development Corporation, 2026). The CFC60 filter, in accordance with the SAE J211 standard, has been used for filtering all the results in this project.

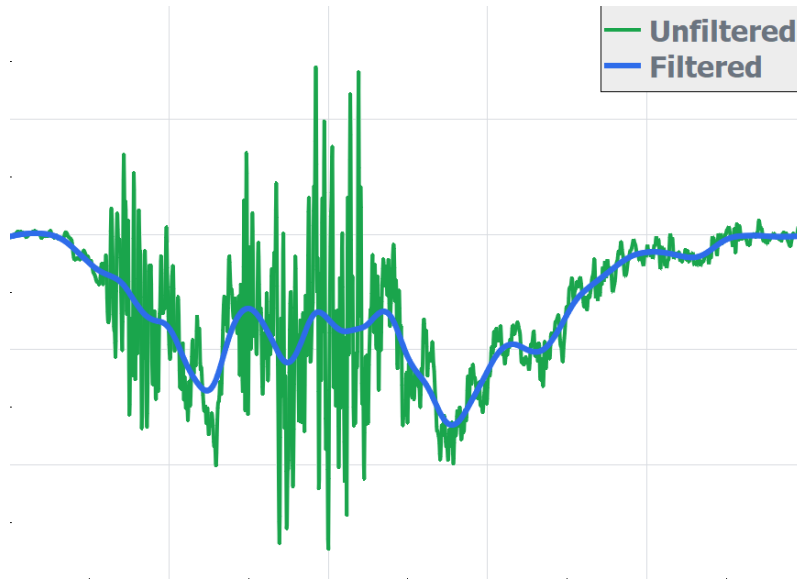


Figure 2.16: Example of an unfiltered and filtered acceleration signal using a CFC60 filter.

3

Method

The following chapter will describe each step of the method, from the initial design of the components to the calibration and validation of the models.

3.1 Generic base model design

The different steps in the design of the base model will be presented in this section.

3.1.1 Identifying crucial components for crash outcome

Some components that make up MCs are not crucial for the crash outcome or are too specialized for a specific MC model, meaning that the part either does not exist for other models or cannot be simplified to a generic part. Thus, some parts that exist on real MC models may be excluded from the generic FE model. To determine which parts were crucial for the crash outcome, several criteria were used. The first criterion was that the part would be in contact with the impact surface during the crash, the second was that the part would be in contact with the rider, and the last criterion was that the part would be needed to connect other parts. Through analysing the given 3D laser scans, pictures, and crash videos of the real MC models, parts that were shared between the models and were crucial for the crash outcome were identified.

The parts that were identified were as follows: the front and rear wheel, the front fork, the handlebars, the frame, the fuel tank, the saddle, the engine, the foot pegs, the wind shield and front light.

3.1.2 Design of base components through reverse engineering

For the FE model to be generic, its constituent parts must also be generic and able to be morphed into more specific shapes. In order to achieve this, the 3D scans of the MC models were used to measure the dimensions of the parts. Certain landmark features that are shared between the parts were measured for each part and documented. The landmark feature could, for example, be the width of the part. This resulted in a list of landmark features with lengths and angles for each part. The list was then used to calculate the mean of the largest and smallest measurements in order to obtain measurements of the landmark features for the generic parts. These

measurements were then used to create the generic parts.

In order for the user to morph the generic parts into model-specific parts, additional considerations had to be taken into account during the modelling process. One such consideration was that the KTM saddle wraps around the fuel tank, and the saddle geometry was therefore modelled to account for this feature. Another consideration was that the fuel tank geometry varied significantly between the models, with the Harley having a teardrop-shaped fuel tank, while the others wrap around the front fork. The generic parts were modelled with these morphing considerations in mind combined with the generic landmark feature measurements.

3.1.3 Assembly and meshing

The generic parts were modelled and then assembled using joints, nodal rigid bodies, extra nodes and rigid bodies. This resulted in the creation of the generic base model. The model was then meshed with a target element size of 5 mm, using ANSA's default criteria, which can be seen in Table 2.1, and additional criteria for the minimum length and the mass-scaling. The criterion for the minimum length was set to 1 mm and this was done to avoid generating small elements and needle elements. The criterion for the mass-scaling was set to 0.00033 ms since this was the chosen time step. This time step was chosen because it was deemed to be small enough to acquire accurate results from the simulation while minimizing computational time.

3.2 Morphing

The morphing strategy consisted of defining morph parameters in ANSA, which could then be used to morph the model. Morph parameters define which geometries should be affected, how they should be modified, which geometries should change shape, and where the boundary between affected and unaffected areas should be located. An example of such a morph parameter can be seen in Figure 3.1, where the purple areas change shape, the green areas are affected, the yellow arrow indicates the direction of the morphing, and the blue border denotes the boundary between affected and unaffected areas.

The specific MC models were then created by only morphing the generic base model, without addition or removal of parts. This was done by using the list of measurements, together with additional measurements obtained from the 3D scans and manufacturer specifications, as well as through visual comparisons.

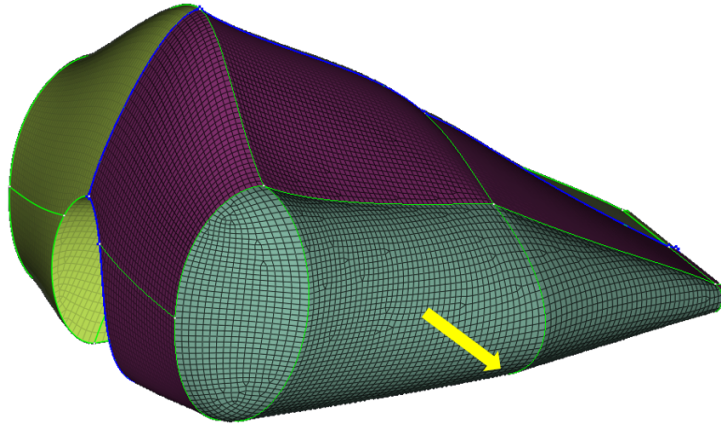


Figure 3.1: Morphing example.

3.3 Physical test and simulation setup

The physical tests that were conducted before the start of the project generated data from sensors on the barrier and on the MC. This data included accelerations and angular velocities at different points on the MC, as well as the force exerted on the barrier. This data had to be extracted, sorted, and filtered in order to compare the physical test results with the results from the FE models. A Python script was then created to extract and format the data from the physical tests. This resulted in the creation of CSV files that could be used by META (BETA CAE Systems, 2026b) to plot the data.

3.3.1 Physical test setup

The physical tests were conducted by Autoliv. Each MC was mounted in a fixture and accelerated to a velocity of 40 km/h. The MCs were then released from the fixture before impacting the barrier. The barrier consisted of concrete blocks, which were positioned at an angle of 10 degrees relative to the initial direction of travel of the MC. The load cell consisted of a metal plate mounted to the concrete blocks, with three sensors located between the plate and the concrete. This setup was used for the Harley-Davidson Sport Glide, the BMW GS1300, and the Honda CB125F, while the setup differed for the KTM Duke 390, for which the barrier was positioned perpendicular to the initial direction of travel of the MC.

3.3.2 Simulation setup

The RW simulations replicated the corresponding physical test setup for each MC. The setup in the simulation environment is illustrated in Figures 3.3 and 3.2. The MCs were positioned as close to the RW as possible to reduce the computational time before impact. The friction coefficient between the ground and the RW was

set to 0.65. An automatic single surface contact definition, including all parts of the MC, was specified. A gravity body load was applied to all MC parts during both the initialization and the transient analysis.

The coordinate system used in the simulations is shown in Figure 3.4. This definition was applied consistently across all four MC simulation setups.

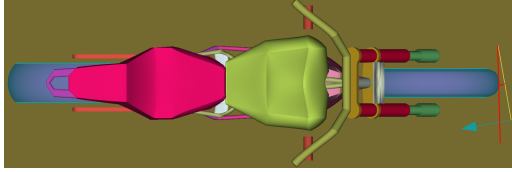


Figure 3.2: Top-view of the simulation setup for the Harley-Davidson sport glide.

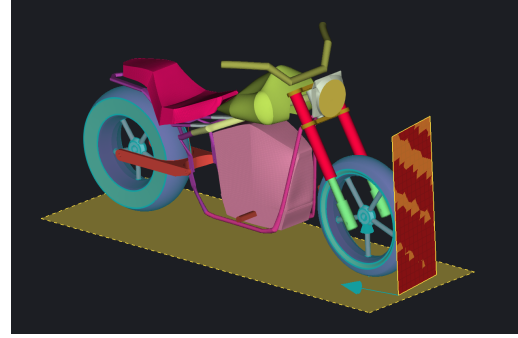


Figure 3.3: Isometric-view of the simulation setup for the Harley-Davidson sport glide.

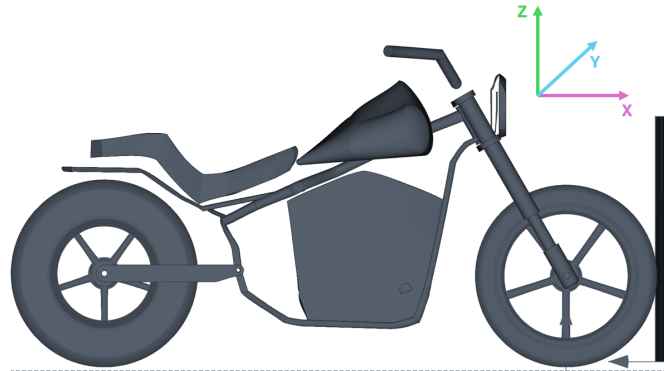


Figure 3.4: Definition of the coordinate system used in the simulation setup.

3.3.2.1 File structure

The simulations utilized the file structure illustrated in Figure 3.5. The Main.key file serves as the master input file and includes the remaining three files, thereby defining the complete simulation setup.

The control_card.k file governs the LS-DYNA control settings, which define the behaviour of the solver both before and during the simulation. An example of a pre-simulation process is dynamic relaxation, which allows the tire pressure solution to converge before the motorcycle is assigned its initial velocity. During the

simulation, this file controls settings such as contact handling, minimum time step size, and other solver parameters. The control card which was used was a Viva+ control card which was chosen because Viva+ is a representative open-source human surrogate to be used together with the MC model (John, 2024).

While `control_card.k` manages the solution process, the `database.k` file governs the simulation output. This file specifies which results are written and at what frequency. One example is the generation of `d3plot` files, which contain displacement information and other data required for post-processing and animation rendering.

The `MC_file.k` file contains the FE mesh, which represents the geometry, and the majority of the component and material definitions used to define the motorcycle model.

Finally, the `Main.key` file contains the simulation setup external to the motorcycle model, including the rigid wall, ground definition, and relevant contact definitions. The purpose of this file is to contain definitions that are not part of the motorcycle itself, as well as definitions that describe the interaction between the motorcycle and other components within the simulation environment, such as the ground and rigid wall.

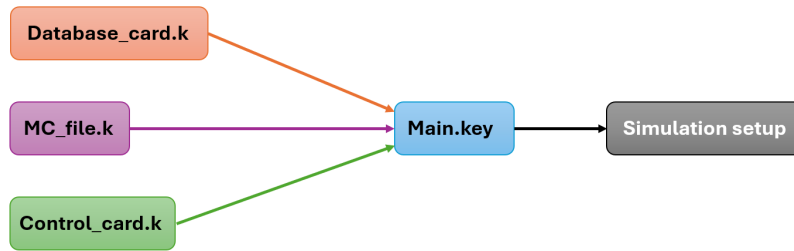


Figure 3.5: File structure for the simulation setup.

3.4 Calibration of morphed models

Having similar geometry to the real MC model is only one part of obtaining a representative FE model; the FE model must also exhibit deformation behaviour similar to that of the real MC. To compare the behaviour of the FE model with that of the real MCs, a Python script was created to extract and format data from history nodes in the model. The history nodes were positioned to correspond to the sensor locations on the physical test MC. This script also loads and plots the simulation data together with the physical test data in META. The quantities compared were the acceleration of the COG in x-direction, the force exerted on the RW and the rotational velocity around the y-axis. Exceptions were made for the BMW GS1300 and the KTM Duke 390. The y-axis angular velocity data from the physical test were not captured for the BMW. For the KTM Duke 390, belonging to a different

test series, neither the x-acceleration of the COG nor the angular velocity of the COG about the y-axis was collected during the physical test. The quantities used for the comparison of the KTM Duke 390 was the displacement of the COG in x- and z-direction, the linear acceleration of the frame in x-direction, and the force exerted on the RW.

The FE models were then tuned using the above-mentioned plots and by visually analysing the 3D plots of the MCs. The tuning was performed by changing different material and structural parameters in the model. The main behaviour that could be tuned was the structural stiffness and, consequently, which parts absorb the crash energy and how much the parts deform. The tuning was performed by changing Young's modulus, thickness, tangential modulus, yield stress, or, for the rubber models of the wheels, the Mooney-Rivlin parameters C_1 and C_2 . The spring and damper of the front fork and rear suspension were also tuned using spring and damper strength parameters, respectively.

The FE models were then simulated using LS-DYNA MPP single precision R12.2.2, on Chalmers' Vera cluster. Manual, incremental modifications were introduced between simulations, and the resulting plots were analysed and compared with the plots and videos from the physical MC crashes. This iterative process continued until a sufficiently similar crash outcome was obtained or if the time constraints of the project were reached.

3.5 Validation

The tuning of the FE models introduces the risk of overfitting, which occurs when a model has been adapted too closely to the training data. This may limit the model's applicability to other load cases, and validation is therefore needed to assess the credibility of the model's results.

Three physical crash tests using the KTM Duke 390 were conducted before the start of the project. These tests were performed by impacting a KTM Duke 390, with a Post Mortem Human Subject (PMHS) positioned on the motorcycle, at 50 km/h into a Honda Accord positioned at an angle of 30 degrees relative to the initial direction of travel of the MC. The test setup is fully described in (Carrol, Bolte, 2024). A previously used simulation setup was used to replicate these tests (Autoliv Development AB, 2025-2027). The FE model of the KTM Duke 390 developed in this project was then integrated into this setup and simulated. The results pertaining to the MC of this simulation were then compared with the results from the physical crash tests in order to validate the MC model's response. The quantities compared were the acceleration of the COG of the MC in the x-, y-, and z-directions, respectively, as well as the resultant acceleration of the MC.

4

Results

The following chapter presents the developed FE models and the results obtained from the simulations, together with comparisons to the corresponding physical test results.

4.1 Base model

This section presents the base components and base model from which the specific MC models were generated through morphing.

4.1.1 Base components

As previously stated, a collection of MC components were identified and modelled, and the developed geometries are shown in Figures 4.1–4.10. All components are modelled using shell elements, except for the saddle foam, which was modelled using solid elements.

The front and rear wheel can be seen in Figures 4.1 and 4.2. The two wheels are very similar, as the rear wheel was constructed from the same base model as the front wheel. The main difference between the models is that the rear wheel is wider, which was achieved through morphing. The wheels consist of five parts: the tire tread, the tire walls, the rim, the spokes, and the hub. The tire tread and walls use the Mooney-Rivlin material model (MAT27 in LS-DYNA), since these are made of rubber, while the rim, spokes, and hub use the piecewise linear plasticity model (MAT24 in LS-DYNA), since these are made of metal. An airbag (AIRBAG_SIMPLE_PRESSURE_VOLUME in LS-DYNA) has been defined inside each tire to represent the internal air volume and pressure. The pressure is set to 2.5 bar for both tires, and puncture of the front tire can be defined by altering its pressure curve.

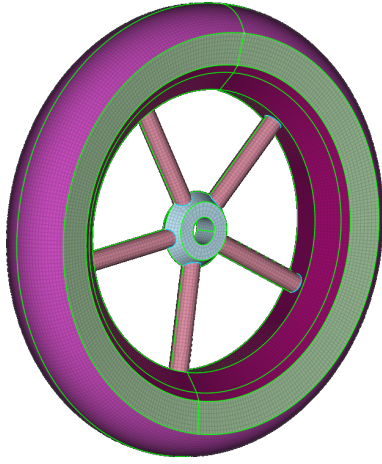


Figure 4.1: Base model front wheel

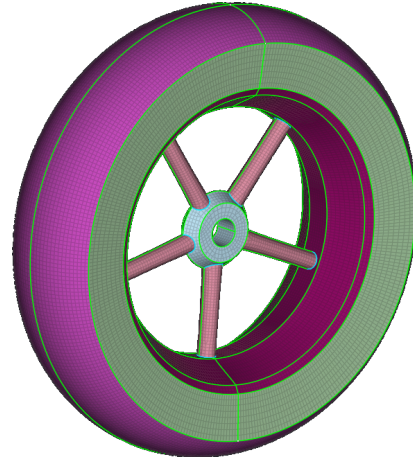


Figure 4.2: Base model rear wheel

The front assembly can be seen in Figure 4.3. The front assembly was constructed to accommodate the large variability in geometry between MC models, where the wind shield has a rectangular shape. Furthermore, if the real MC does not include a wind shield, this component can be folded down to represent only the headlight. The front assembly consists of four parts: the headlight, the wind shield, the plastic side trims, and the metal back and front. All parts use the piecewise linear plasticity material model.

The front fork can be seen in Figure 4.4. It consists of four parts: the bottom tube, the top tube, and the bottom and top clamps. All parts use the piecewise linear plasticity material model and are made of metal. The bottom and top parts are connected with a spring and a damper, modelled in parallel, in each tube. The spring is modelled as a beam element using BEAM_ELFORM_6 and MAT67 in LS-DYNA, while the damper is modelled as a discrete element using MAT_S05 in LS-DYNA. Cylindrical joints are also defined in each tube, allowing the top and bottom parts to translate along and rotate about a shared axis. The front suspension of the MCs can then be modelled using these springs and dampers. The shown front fork is an inverted fork, but it can be changed into a telescopic fork by reducing the top radius, increasing the bottom radius, and extending the bottom part over the top part using morphing.

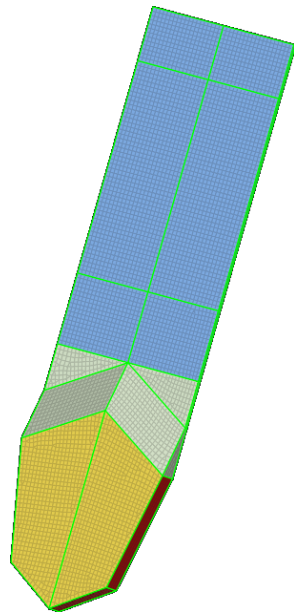


Figure 4.3: Base model front

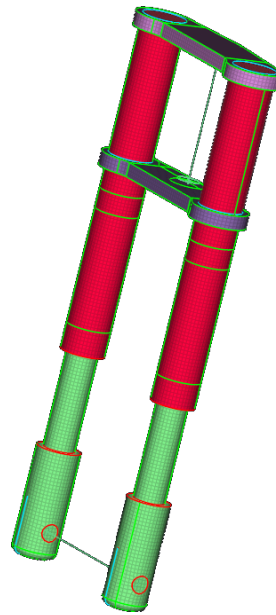


Figure 4.4: Base model front fork

The handlebars are shown in Figure 4.5. The handlebars are modelled as a single part and segmented to account for variations in handlebar curvature.

The frame is shown in Figure 4.6. The frame consists of six parts: the front connector, which connects the frame to the front fork; the upper and lower sides; the crossbeam; the rear; and the swing arm. The piecewise linear plasticity model is used for all parts except the rear and swing arm, which use a rigid material model (MAT20 in LS-DYNA), since these parts are not expected to deform. All parts of the frame are modelled as metallic components. The frame contains one revolute joint, which connects it to the swing arm. This joint enables rotation of the swing arm. An additional joint stiffness can be applied, if needed, to increase the rigidity of the joint. This joint works in combination with a spring-damper system, which is also connected between the frame and the swing arm. The spring and damper use the same element types and material models as the front fork springs and dampers. This was done to model the rear suspension. Mass elements are also defined on the rear part of the frame and can be used to adjust the mass of the MC and the location of the MC's COG.

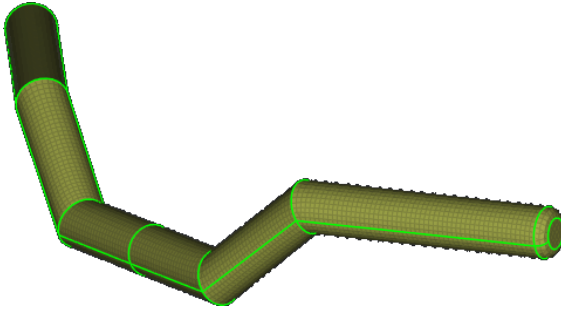


Figure 4.5: Base model handle-bars.

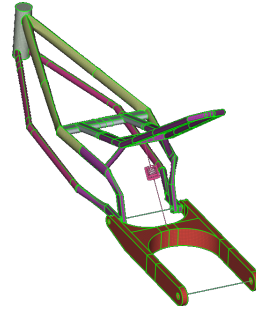


Figure 4.6: Base model frame.

The fuel tank can be seen in Figure 4.7. It consists of one part, which has been modelled using the piecewise linear plasticity material model and is made of metal. It has been modelled to account for the large geometric variability between fuel tanks. This is reflected by the relatively rounded shape of the fuel tank and the absence of features that would make it too model-specific. The model also has an indent along the underside of the fuel tank. This was included because some MC model frames are positioned within this region.

The saddle can be seen in Figure 4.8. It consists of two parts, one for the outside saddle fabric and one for the inner saddle foam. The outside uses a fabric material model (MAT34 in LS-DYNA) and is modelled using shell elements, while the inside uses the Fu-Chang foam material model (MAT83 in LS-DYNA) and consists of an unstructured tetrahedral volume mesh. The material parameters for the foam were taken from the OpenVT Open Access Front Seat Model (OpenVT, 2026).

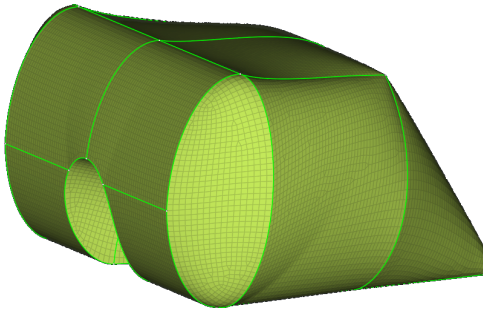


Figure 4.7: Base model fuel tank.

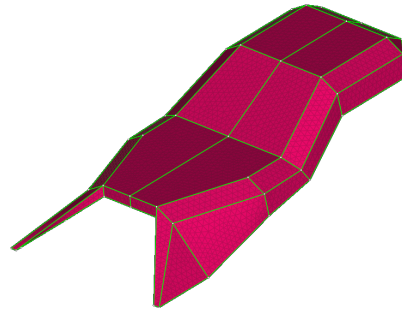


Figure 4.8: Base model saddle.

The engine is shown in Figure 4.9. It is modelled using a rigid material model and includes additional mass elements that can be used to adjust the overall mass of the MC and the location of its COG.

The pedals are shown in Figure 4.10. The pedals use a rigid material model and are designed to be easily morphed into the desired shape.

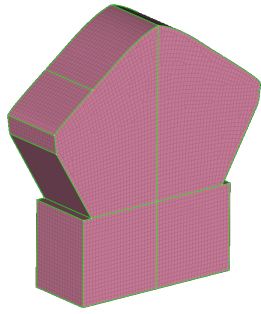


Figure 4.9: Base model engine.

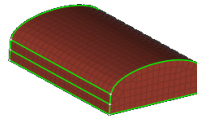
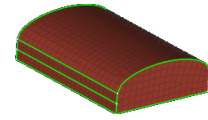


Figure 4.10: Base model pedals.



4.1.2 Base model assembly

The above-mentioned base components were assembled into the full base FE model, which is shown in Figure 4.11. As previously stated in the method, the parts were assembled using constraints such as joints, rigid bodies, nodal rigid bodies, and extra nodes.

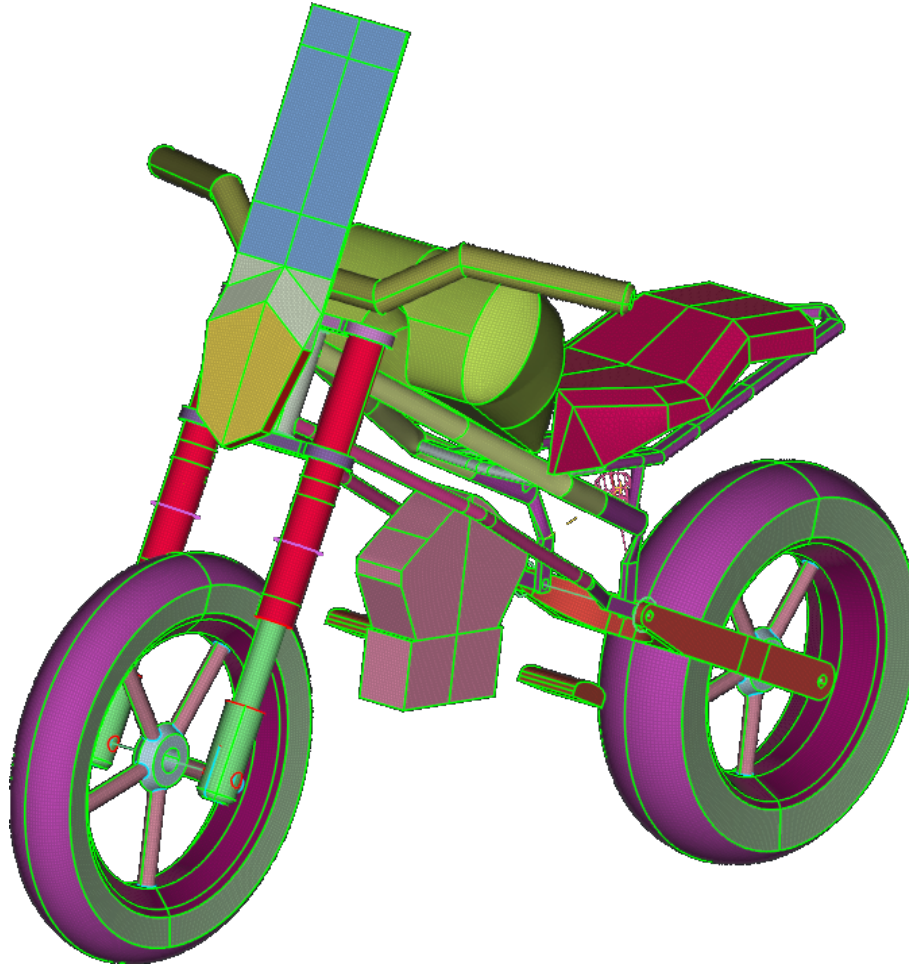


Figure 4.11: Assembled base FE model.

Five revolute joints are used to connect the components. One joint connects the front wheel hub to the bottom of the front fork. Two of the joints connect the frame's front connector to the front fork clamps, one for the top clamp and one for the bottom clamp. The fourth joint connects the swing arm to the rear wheel. The fifth joint is the frame–swing arm joint, which connects the frame to the swing arm. These joints allow the parts to remain connected while still being able to rotate independently. An example is the front wheel–front fork joint, which can be seen in Figure 4.12. The rotation allowed by these joints is shown in Figure 4.13, where the red arrows illustrate the allowed rotation.

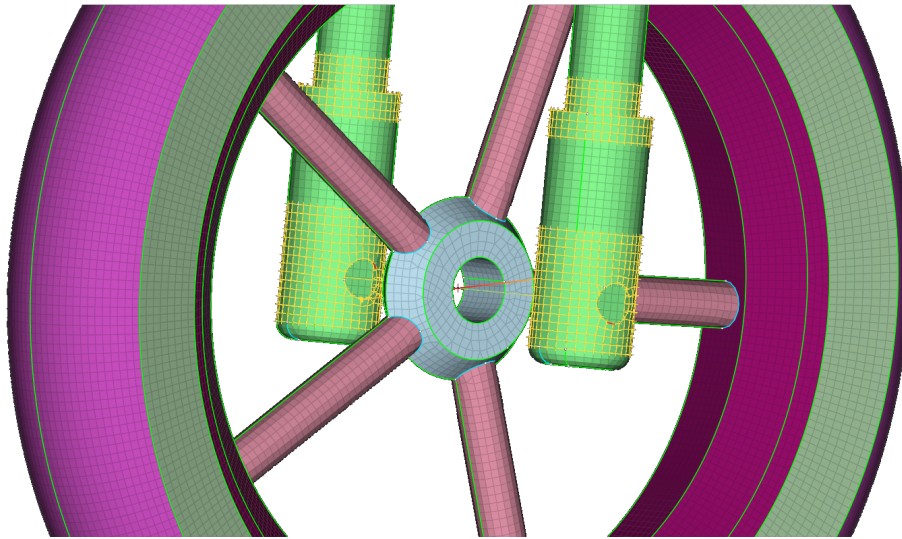


Figure 4.12: Front wheel to front fork revolute joint.

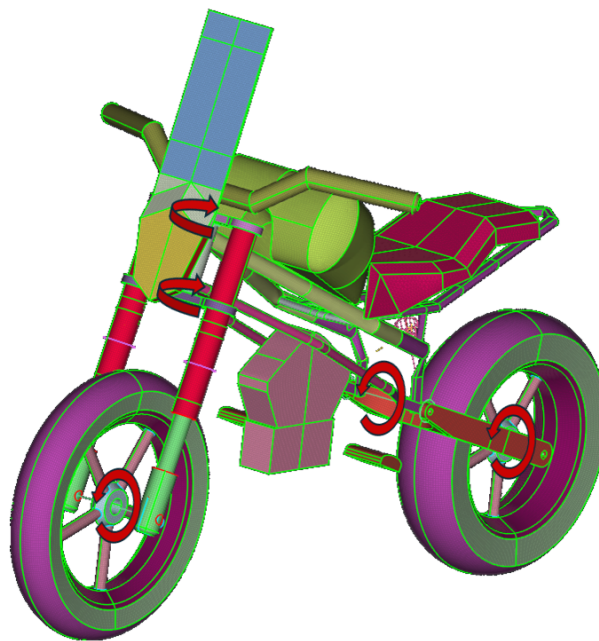


Figure 4.13: Base model with revolute joints illustrated with red arrows.

Several nodal rigid bodies and extra nodes are used to connect the different components, as shown in Figures 4.14–4.18. Extra nodes are used to connect deformable parts to rigid parts. This can be seen in Figure 4.14, where the engine is connected to the lower sides of the frame and the middle of the top frame bar. Another such connection is shown in Figure 4.15, where the underside of the saddle fabric is connected to the rear of the frame. The nodal rigid bodies are coloured yellow, while the extra nodes are assigned the colour of the corresponding rigid part.

The fuel tank has four nodal rigid bodies, which connect the underside of the fuel tank to the upper sides of the frame. It has two connections at the front and two

at the rear, with one on each side. This ensures that the fuel tank has a stable connection and moves with the frame. This is shown in Figure 4.16.

The front assembly has been connected to the front fork using three nodal rigid bodies: one on the lower clamp and two on the upper clamp, as shown in Figure 4.17. This three-point connection allows the front assembly to deform slightly while still maintaining a robust connection.

The handlebars have been connected to the front fork using two nodal rigid bodies, one located at the top of each front fork tube. This is shown in Figure 4.18.

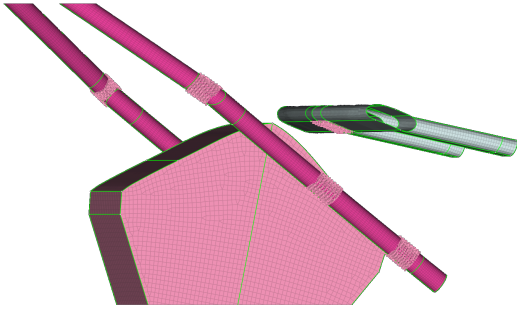


Figure 4.14: Engine to frame extra nodes.

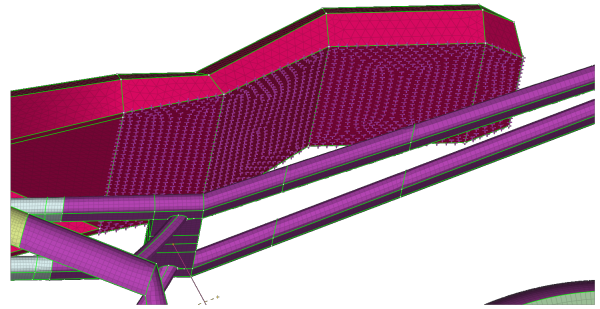


Figure 4.15: saddle to frame extra nodes.

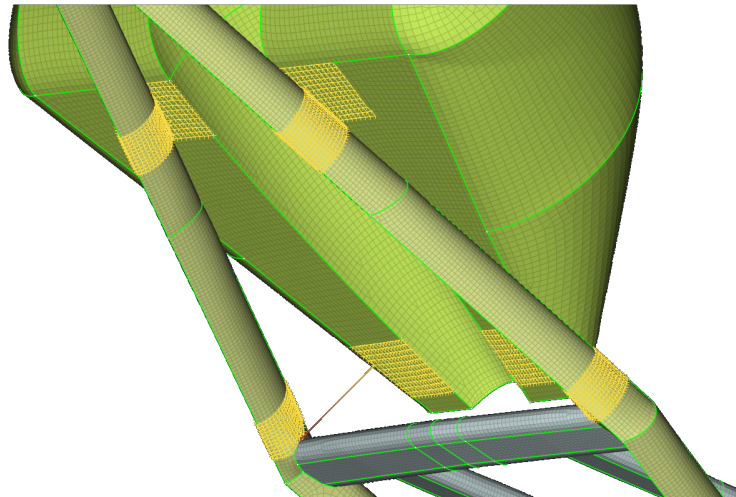


Figure 4.16: Fuel tank to frame nodal rigid body.

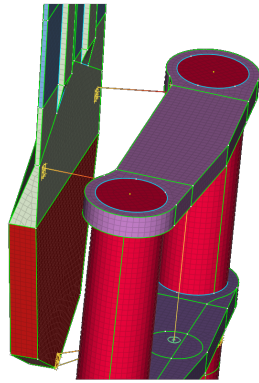


Figure 4.17: Front to front fork nodal rigid body.

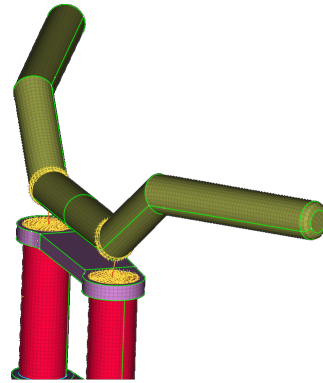


Figure 4.18: Handlebars to front fork nodal rigid body.

4.2 Morphed models

This section presents the geometries that were morphed from the base model. The morphed models were compared with the 3D scans using measurements taken during the morphing process and model overlays. Some of these measurements can be seen in Table 4.1 with the locations of the measurements shown in Figure 4.19. It can be seen in this table that the FE-model and 3D scan measurements are very similar for every model, with the maximum deviation being measurement J for the Honda where the measurement in the FE-model is 5.45% larger than that of the same measurement in the 3D scan. It should be noted that many more measurements were taken but are not shown in this table. The overlay comparisons between the morphed models and the 3D scans are shown in the figures below, where the red-coloured model represents the morphed model and the yellow model represents the 3D scan.

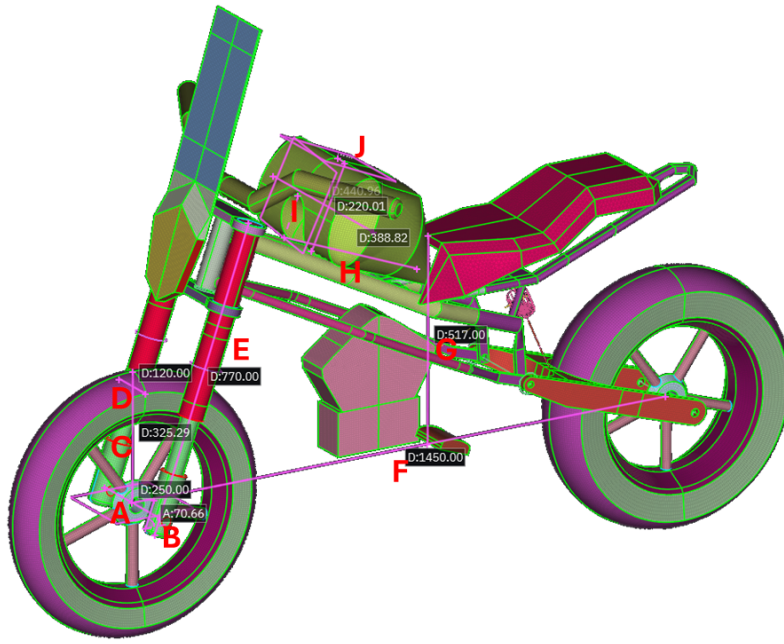


Figure 4.19: Measurements with accompanied letter.

Table 4.1: Feature measurements, FE-model and 3D scan.

	Feature measurements											
	Harley			BMW			Honda			KTM Duke		
	FE	Scan	%	FE	Scan	%	FE	Scan	%	FE	Scan	%
A	250mm	253mm	98.81	208mm	207mm	100.48	160mm	160mm	100	206mm	206mm	100
B	60°	58°	103.45	63.8°	63.8°	100	66°	65°	101.54	64.7°	64.45°	100.39
C	320mm	316mm	101.27	325mm	326mm	99.69	309mm	303mm	101.98	291mm	296mm	98.31
D	130mm	129mm	100.78	113mm	113mm	100	80mm	77mm	103.9	95mm	100mm	95
E	770mm	775mm	99.35	890mm	890mm	100	701mm	701mm	100	744mm	749mm	99.33
F	1625mm	1630mm	99.69	1518mm	1510mm	100.53	1295mm	1291mm	100.31	1389mm	1370mm	101.39
G	456mm	448mm	101.79	597mm	598mm	99.83	513mm	519mm	98.84	559mm	548mm	102.01
H	485mm	475mm	102.11	746mm	749mm	99.6	647mm	644mm	100.47	654mm	663mm	98.64
I	440mm	423mm	104.02	650mm	650mm	100	419mm	420mm	99.76	448mm	467mm	95.93
J	325mm	313mm	103.83	397mm	412mm	96.36	290mm	275mm	105.45	265mm	268mm	98.88

4.2.1 Harley-Davidson sport glide

The morphed Harley-Davidson model is shown in Figure 4.20. The comparison between the FE model and the 3D scan is shown in Figures 4.21–4.23. These figures show that the FE model is geometrically similar to the 3D scan. However, some discrepancies can be observed. One example is that the 3D scan has a higher rake angle. The model was morphed in accordance with the manufacturer's specification, which specified a rake angle of 30°, while the 3D scan has a rake angle of 31.5°.



Figure 4.20: Representative FE-model of the Harley-Davidson sport glide.

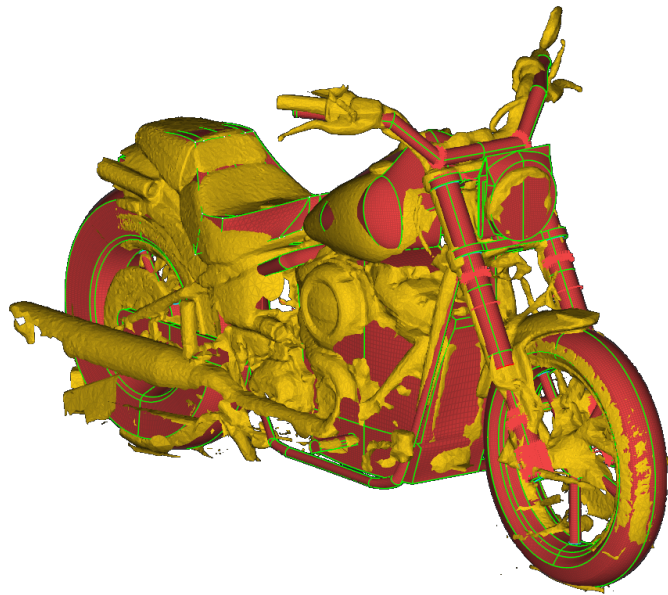


Figure 4.21: Harley-Davidson sport glide FE-model and 3D scan comparison, isometric view.

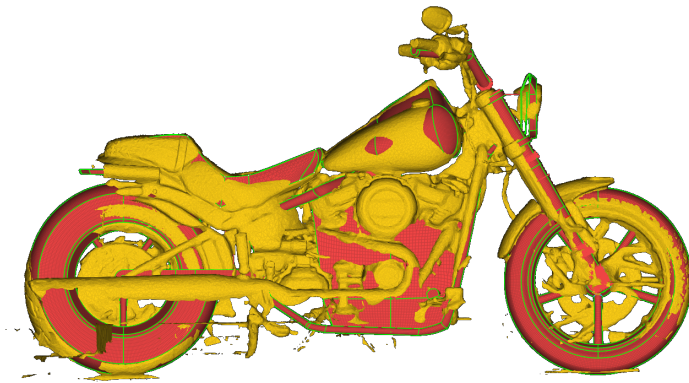


Figure 4.22: Harley-Davidson sport glide FE-model and 3D scan comparison, side view.

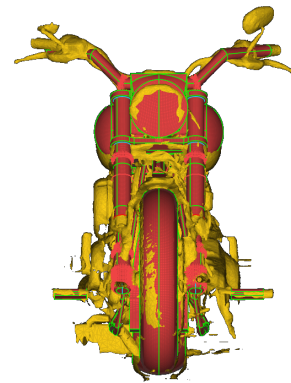


Figure 4.23: Harley-Davidson sport glide FE-model and 3D scan comparison, front view.

4.2.2 BMW GS1300

The morphed FE model of the BMW GS1300 can be seen in Figure 4.24. It should be noted that the real MC differs from the other MCs in two main ways. The first difference is that the suspension is neither telescopic nor inverted. This is because the BMW GS1300 has a unique suspension system that is not located inside the front fork but behind the triple clamps. The placement of this suspension system made it impossible to geometrically replicate through morphing of the base model. The suspension inside the front fork was therefore locked. The second difference is that the front assembly is connected to the fuel tank instead of the front fork. This has been replicated in the FE model by changing the nodal rigid body which connects the front assembly to the front fork to instead connect the front assembly to the fuel tank.

Comparisons between the FE model and the 3D scan are shown in Figures 4.25–4.27. The close agreement between the overlaid geometries indicates that the FE model provides a good representation of the scanned motorcycle geometry.

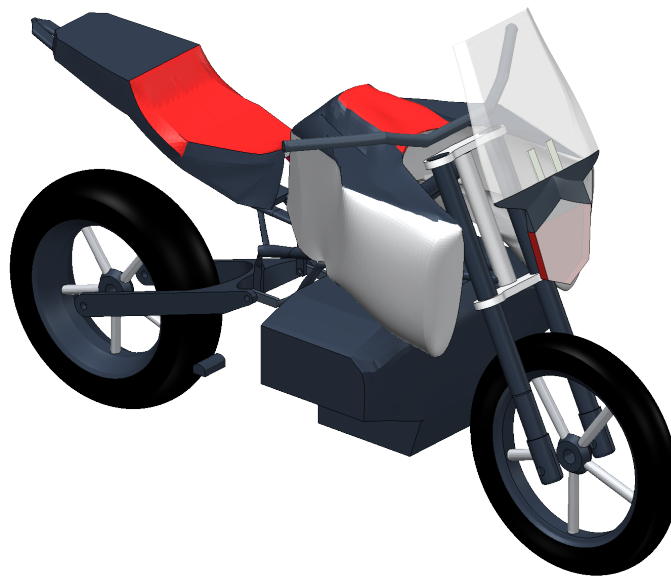


Figure 4.24: Representative FE-model of the BMW GS1300.

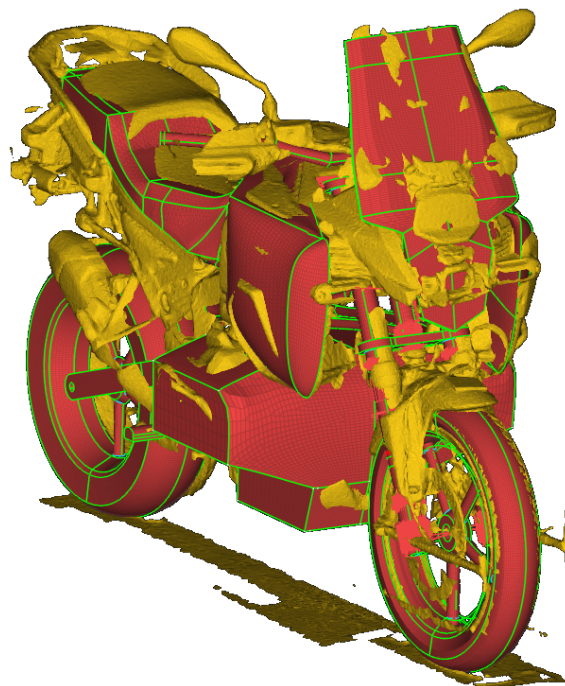


Figure 4.25: BMW GS1300 FE-model and 3D scan comparison, isometric view.

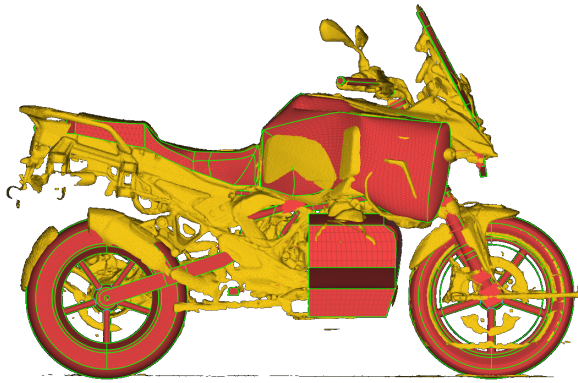


Figure 4.26: BMW GS1300 FE-model and 3D scan comparison, side view.

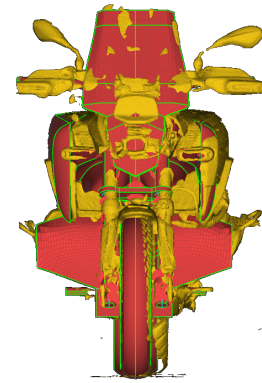


Figure 4.27: BMW GS1300 FE-model and 3D scan comparison, front view.

4.2.3 Honda CB125F

The morphed Honda FE model is shown in Figure 4.28, and the comparison between the FE model and the 3D scan is shown in Figures 4.29–4.31. These figures show that the FE model geometry corresponds well to the 3D scan geometry.

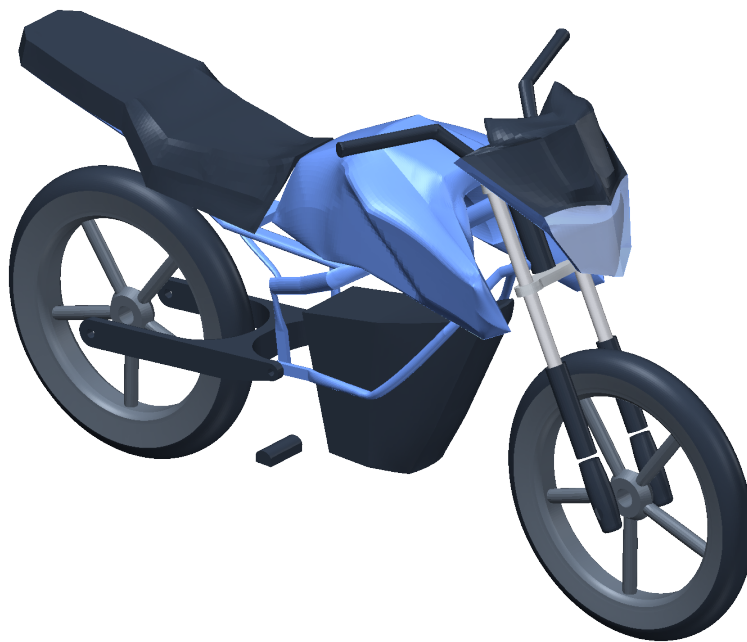


Figure 4.28: Representative FE-model of the Honda CB125F.

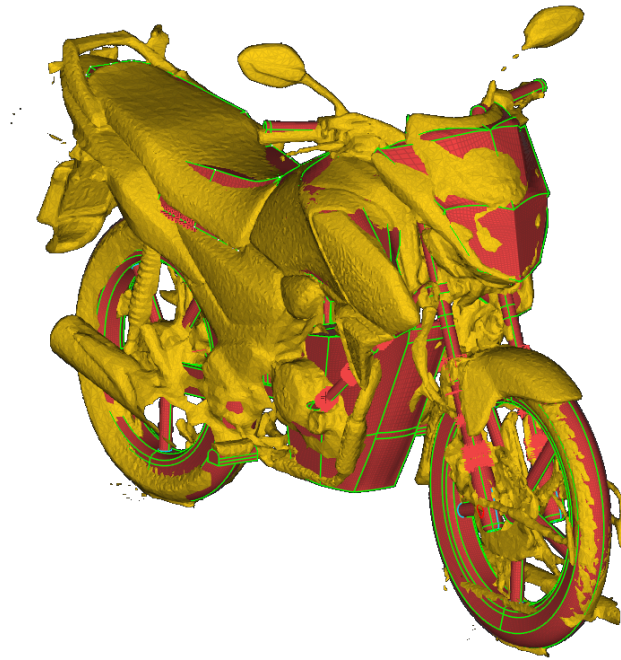


Figure 4.29: Honda CB125F FE-model and 3D scan comparison, isometric view.

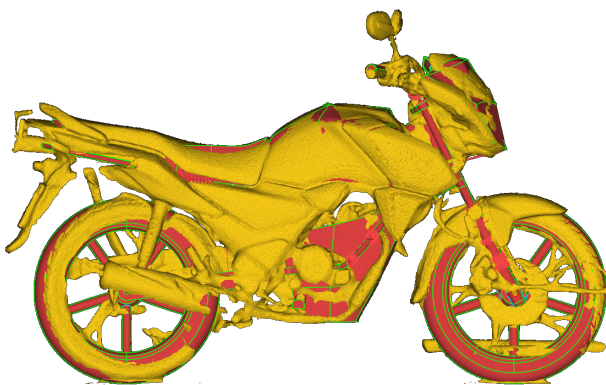


Figure 4.30: Honda CB125F FE-model and 3D scan comparison, side view.

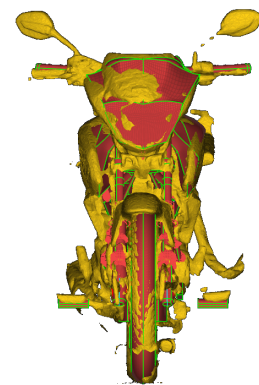


Figure 4.31: Honda CB125F FE-model and 3D scan comparison, front view.

4.2.4 KTM duke 390

The morphed KTM Duke 390 model is shown in Figure 4.32, and comparisons between the 3D scan and the morphed model are shown in Figures 4.33–4.35. It should be noted that the real MC is resting on its kickstand in the 3D scan, which has caused the MC to be tilted and the steering column to be rotated relative to the frame. To account for this, the FE model has been tilted in the comparison figures. As a result, some parts of the models do not coincide exactly. Nevertheless, when this difference is taken into consideration, the geometries of the two models are shown to largely coincide.



Figure 4.32: Representative FE-model of the KTM Duke 390.

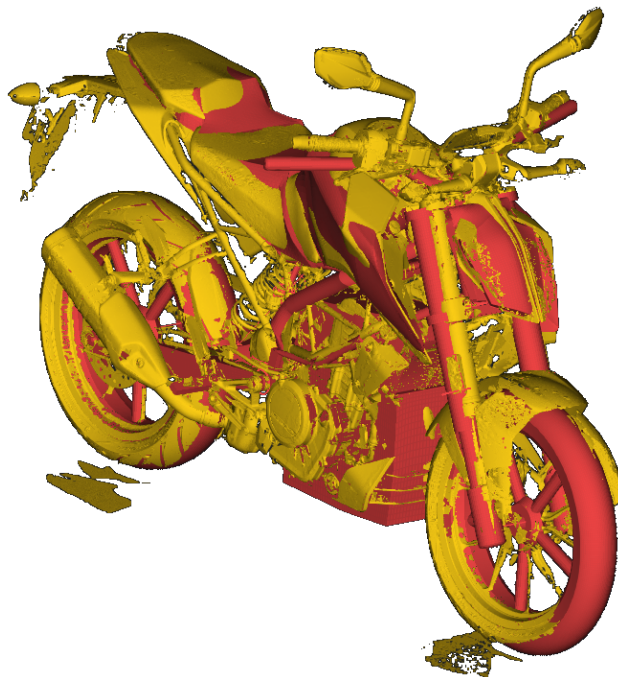


Figure 4.33: KTM duke 390 FE-model and 3D scan comparison, isometric view.

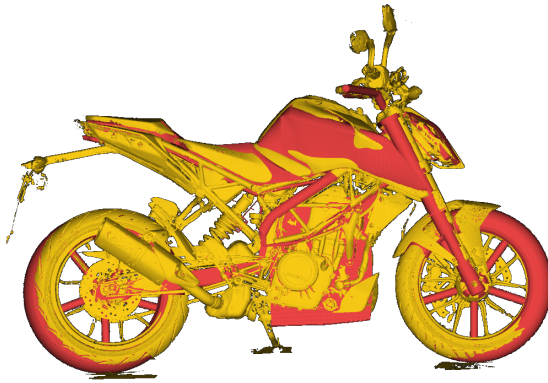


Figure 4.34: KTM duke 390 FE-model and 3D scan comparison, side view.

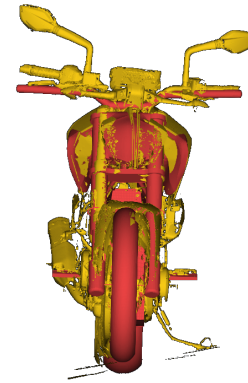


Figure 4.35: KTM duke 390 FE-model and 3D scan comparison, front view.

4.3 Mesh quality

The mesh quality for each of the four MCs is presented in this section. In the following figures, elements that fail one or more quality criteria are highlighted using colours other than grey. Grey elements satisfy the applied criteria. Yellow indicates that the aspect ratio is outside the accepted range, green indicates skewed elements, and light blue indicates warped elements, meaning that one or more nodes do not lie in the same plane. Purple indicates elements that will be mass-scaled during the simulation, red indicates elements that are too small, and dark blue indicates elements with a Jacobian that is too small or negative.

These colours were assigned based on, as previously stated, ANSA's default criteria together with the chosen criteria for the minimum length and mass-scaling. These criteria are shown in Table 2.1. It should be noted that an element highlighted in one colour may also fail additional criteria.

The mesh for each FE model consists of 192,736 shell elements and 38,431 solid elements.

4.3.1 Harley-Davidson sport glide

The mesh quality of the complete Harley-Davidson FE model is shown in Figure 4.36. Most of the elements that fail the quality criteria are located on the bottom of the frame and on the saddle. These areas are shown separately in Figures 4.37 and 4.38. The saddle contains several elements with aspect ratio and skewness issues. The frame shows similar element issues, in addition to mass scaling issues.

In total, the Harley-Davidson FE model contains 36,399 quality violating elements, 15,578 of which are shells and 20,810 of which are solids.

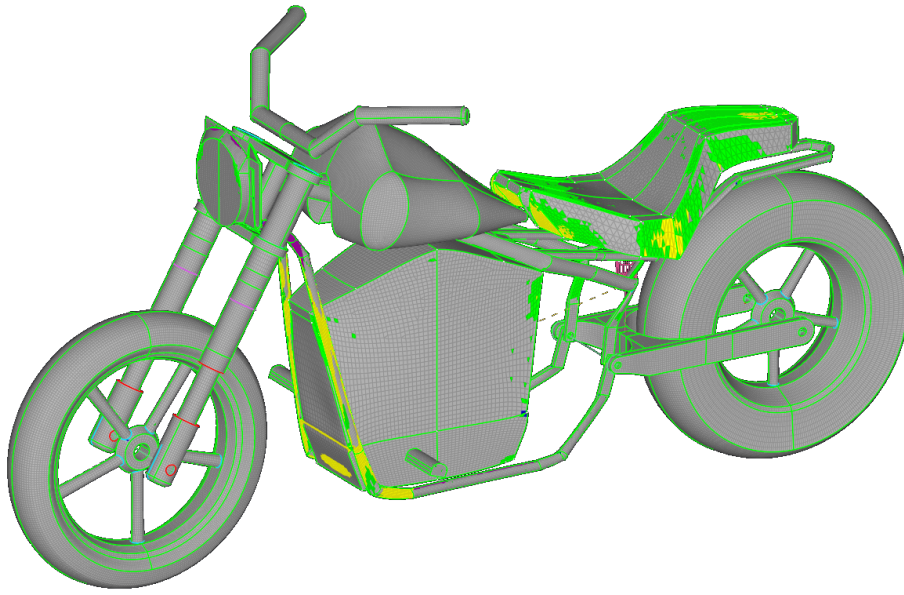


Figure 4.36: Harley-Davidson mesh quality, isometric view.

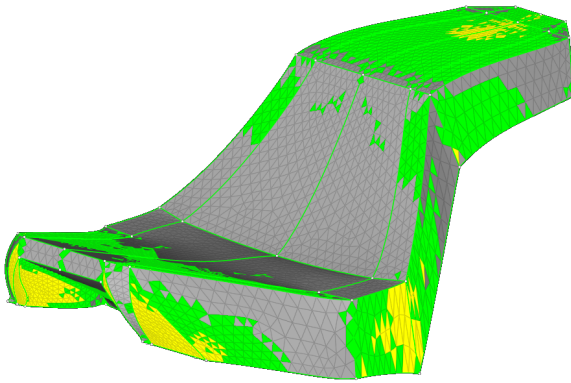


Figure 4.37: Harley-Davidson mesh quality, saddle.

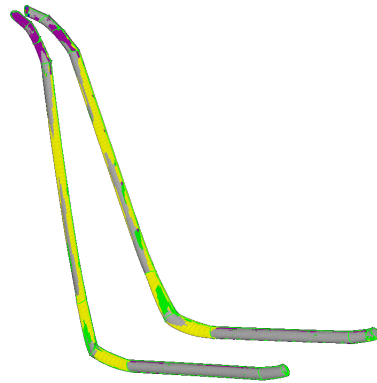


Figure 4.38: Harley-Davidson mesh quality, frame.

4.3.2 BMW GS1300

The isometric view of the BMW mesh quality is shown in Figure 4.39. This figure shows several areas with elements that fail the quality criteria, including the fuel tank, the saddle, and the front. Similar areas are also present on the frame, as shown in Figure 4.41. It should also be noted that some elements on the engine fail the quality criteria; however, since the engine is rigid and cannot deform, the element quality is less critical. A detailed view of the saddle is shown in Figure 4.40, while detailed views of the fuel tank are shown in Figures 4.42 and 4.43.

In total, the BMW GS1300 FE-model has 49,053 quality violating elements, where 15,916 are solid elements and 33,137 shell elements.

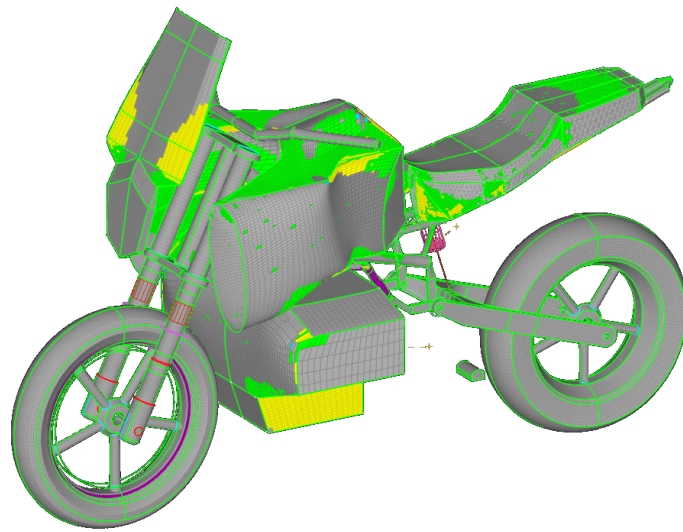


Figure 4.39: BMW GS1300 mesh quality, isometric view.

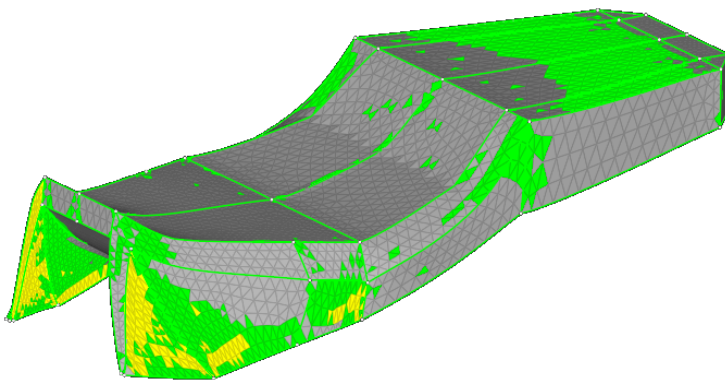


Figure 4.40: BMW GS1300 mesh quality, saddle.

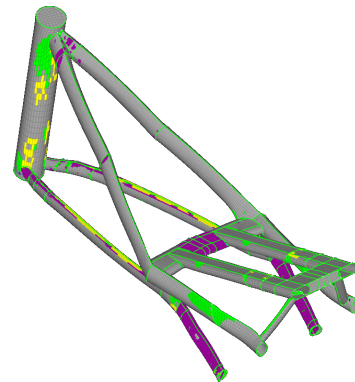


Figure 4.41: BMW GS1300 mesh quality, frame.

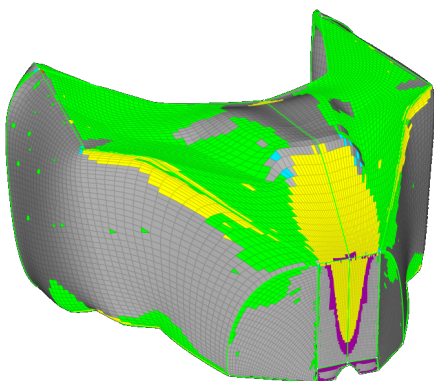


Figure 4.42: BMW GS1300 mesh quality fuel tank, isometric view.

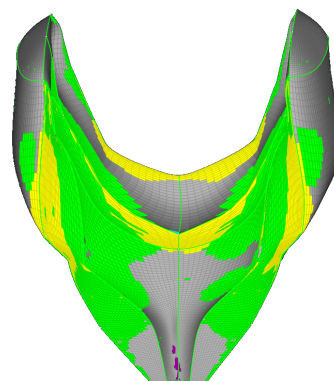


Figure 4.43: BMW GS1300 mesh quality fuel tank, underside.

4.3.3 Honda CB125F

The mesh quality of the complete Honda FE model is shown in Figure 4.44. Similar to the BMW model, the main areas with elements that fail the quality criteria are located around the front, fuel tank, saddle, frame, and engine. Since the engine is rigid, these elements can be considered less critical. The other components are shown separately in Figures 4.45–4.48. Many elements on the tank and saddle are stretched and skewed, while several elements on the frame have mass scaling issues.

In total, the Honda CB125F FE-model has 51,363 quality violating elements, 30,911 of which are shells and 20,452 of which are solids.

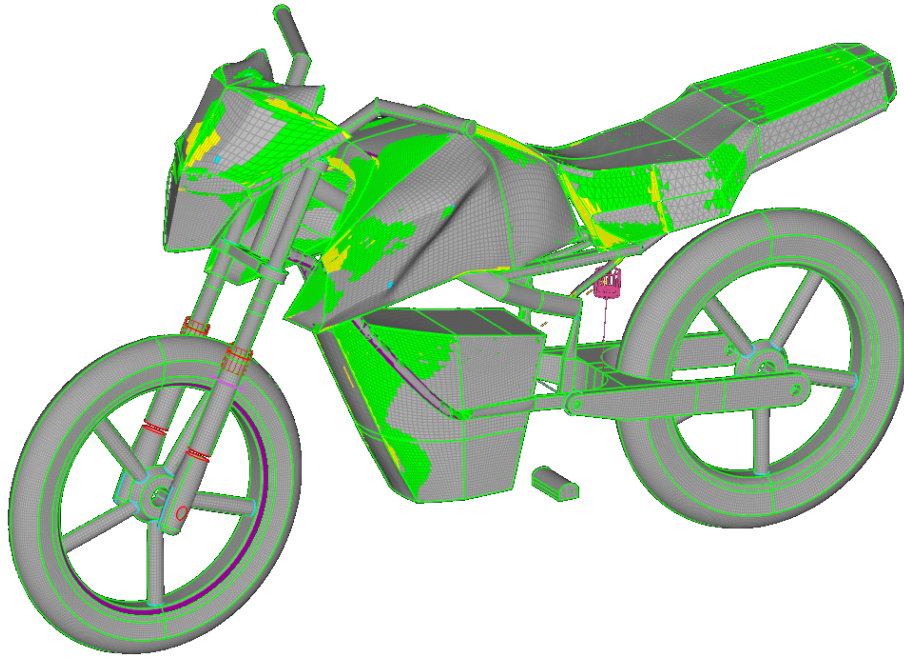


Figure 4.44: Honda CB125F mesh quality, isometric view.

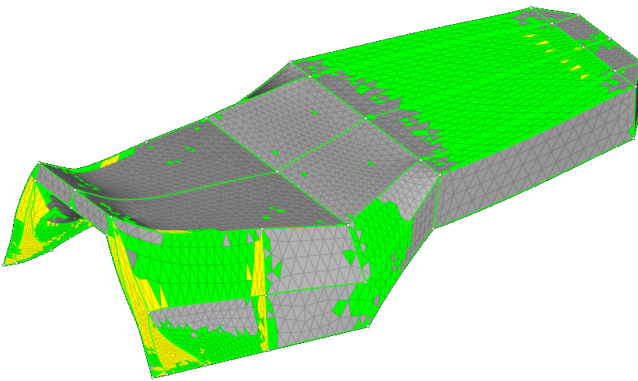


Figure 4.45: Honda CB125F mesh quality, saddle.

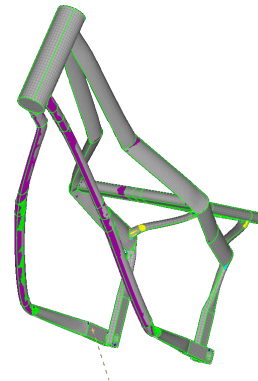


Figure 4.46: Honda CB125F mesh quality, frame.

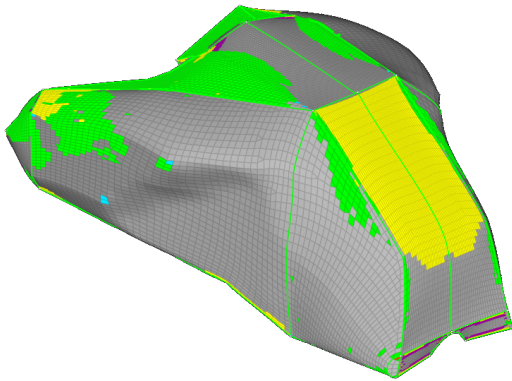


Figure 4.47: Honda CB125F mesh quality fuel tank, isometric view.

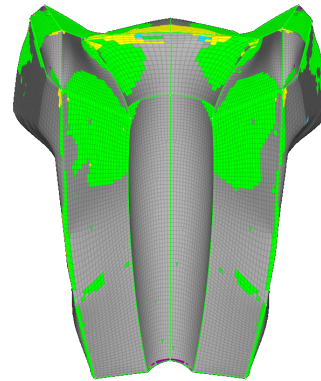


Figure 4.48: Honda CB125F mesh quality fuel tank, underside.

4.3.4 KTM Duke 390

The mesh quality of the complete KTM Duke 390 FE model is shown in Figure 4.49. Similar to the other FE models, the fuel tank and frame contain most of the elements that fail the quality criteria. Some elements on the saddle also fail the quality criteria. These parts are shown separately in Figures 4.50–4.53. These figures show that the saddle and fuel tank contain elements with aspect ratio and skewness issues, although most of the elements that fail the quality criteria on the saddle are located on the pillion saddle. The frame also contains elements affected by mass scaling.

In total, the KTM Duke 390 FE-model has 55,845 quality violating elements, where 14,479 are solid elements and 41,366 shell elements.

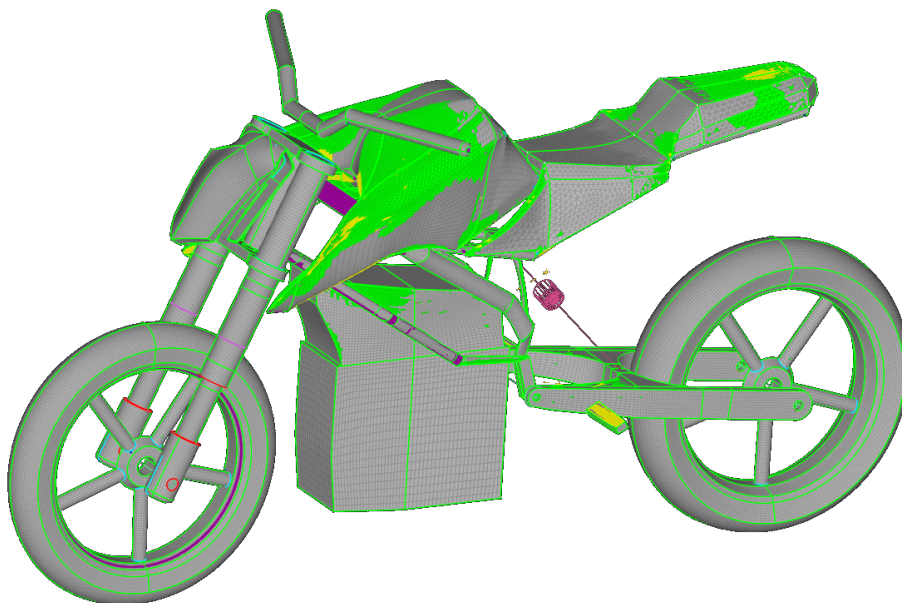


Figure 4.49: KTM duke 390 mesh quality, isometric view.

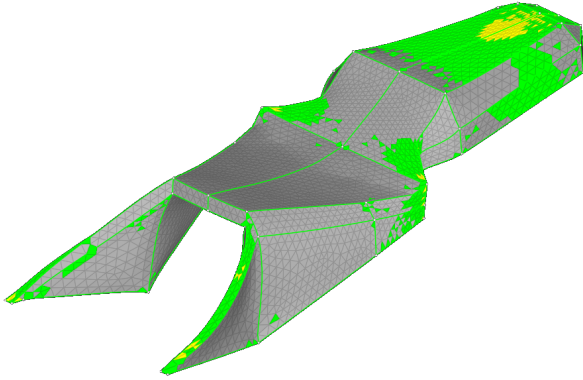


Figure 4.50: KTM duke 390 mesh quality, saddle.

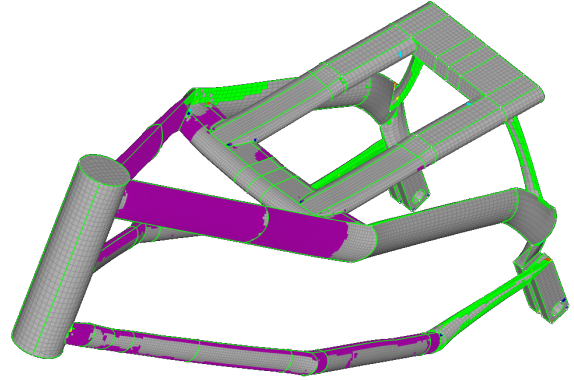


Figure 4.51: KTM duke 390 mesh quality, frame.

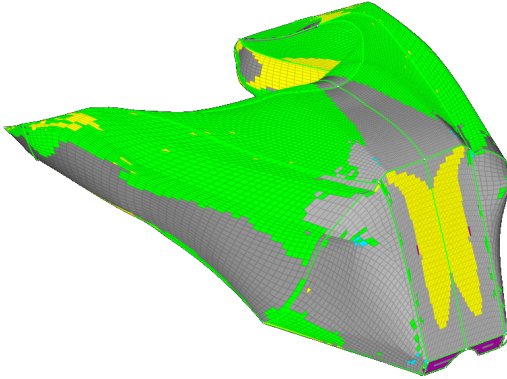


Figure 4.52: KTM duke 390 mesh quality fuel tank, isometric view.

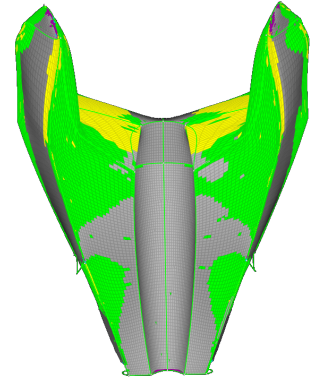


Figure 4.53: KTM duke 390 mesh quality fuel tank, underside.

4.3.5 Quality violating elements

Elements can have multiple quality issues but can only be displayed using one colour. The total number and percentage of off-elements were therefore tabulated by criterion and model and are shown in Tables 4.2 and 4.3. The percentages are calculated relative to the total number of elements of the same type. These tables show that the solid elements have a higher percentage of off-elements than the shell elements. The percentage of shell off-elements is relatively small for all models, with skewness being the most common issue. Skewness is also the most common issue for the solid elements. It should be noted that all solid elements are located in the saddle.

Table 4.2: Shell off-elements by criteria and model.

Model	Off-criteria											
	Skewness		Jacobian		Min length		Aspect		Warping		Crash	
	N	%	N	%	N	%	N	%	N	%	N	%
Harley	8715	4.522	257	0.133	57	0.03	5411	2.808	103	0.007	6130	3.181
BMW	15386	7.983	92	0.048	658	0.341	7195	3.733	310	0.161	16345	8.481
Honda	16970	8.805	93	0.048	230	0.119	3859	2.002	323	0.168	15322	7.95
KTM	16841	8.738	236	0.122	404	0.210	4551	2.361	278	0.144	25600	13.282

Table 4.3: Solid off-elements by criteria and model.

Model	Off-criteria											
	Skewness		Jacobian		Min length		Aspect		Warping		Crash	
	N	%	N	%	N	%	N	%	N	%	N	%
Harley	20147	52.424	3	0.008	0	0	6021	15.667	0	0	25	0.065
BMW	15292	39.791	2	0.005	0	0	4002	10.413	0	0	9	0.023
Honda	19895	51.768	13	0.034	0	0	4682	12.183	0	0	23	0.06
KTM	13700	35.648	0	0	0	0	2842	7.395	0	0	0	0

4.3.6 Mass-scaling and COG placement

The mesh quality figures for the FE models, together with Tables 4.2 and 4.3, show that some elements had mass scaling issues, but these do not show the magnitude of the mass scaling. The mass scaling has therefore been quantified and tabulated, as shown in Table 4.4. This table shows that the mass scaling is relatively small, with the largest amount occurring in the KTM Duke 390.

Table 4.4: Mass-scaling for each FE-model.

Mass-scaling table			
FE-Model	Net Mass [kg]	Scaled mass [kg]	Mass Increase [%]
Harley-Davidson	298	298.425	0.14244
BMW GS1300	254	255.443	.568241
Honda Cb125F	107	108.113	1.04012
KTM duke 390	160	163.532	2.51942

The masses and placements of the COG for the physical MCs and the FE models has been tabulated and can be seen in Table 4.5. It should be noted that the origin is located in the middle of the front wheel axle. It should also be noted that there was no data for the y-placement of the COG for the physical MCs and has thus been assumed to be located at $y = 0$.

Table 4.5: Masses and COG placements of the physical MCs and the FE models.

	COG placement (x,y,z) [mm]		Mass [kg]	
MC	Physical	FE-model	Physical	FE-model
Harley-Davidson	(-920, 0, 180)	(-920.25, -0.64, 180.566)	298	298
BMW GS1300	(-780, 0, 270)	(-779.49, -1.50, 271.83)	254	254
Honda CB125F	(-580, 0, 181.37)	(-580.53, -0.5, 181.1)	107	107
KTM Duke 390	(-672, 0, 214)	(-670.32, 0.08, 215.54)	160	160

4.4 Simulation and physical crash test

This section presents the results of the simulations and compares them with the physical crash test results.

The simulations generated d3plot files, enabling visualisation of the response at specified time intervals. Motion snapshots were extracted at corresponding time instances from both the physical test recordings and the simulation results. These snapshots were then compared side by side to assess the correlation between the simulated and physical test responses. The resulting comparisons are presented in this section for each MC.

The compared quantities from the tests and the simulations have been plotted together in graphs and are presented in this section as well.

Additionally, the material parameters and the corresponding LS-DYNA material definitions can be found in appendices A.1, A.2, A.3, and A.4 for all MCs.

4.4.1 Comparison of calibrated models and physical test with a rigid wall

In this section, the results of the physical crash tests and simulations are compared for each MC model. It should be noted that the graphed data were filtered using a CFC60 filter.

4.4.1.1 Harley-Davidson Sport Glide

The side-by-side simulation results are shown in Figure 4.54. The figure shows that the model behaves similarly to the real MC up to approximately 60 ms, after which the front wheel turns considerably more in the physical test than in the FE model. This also causes the MC to rebound earlier in the simulation than in the test. It can also be seen that both sides of the front wheel rim are deformed in the simulation, while only the left side has deformed and failed in the physical test.

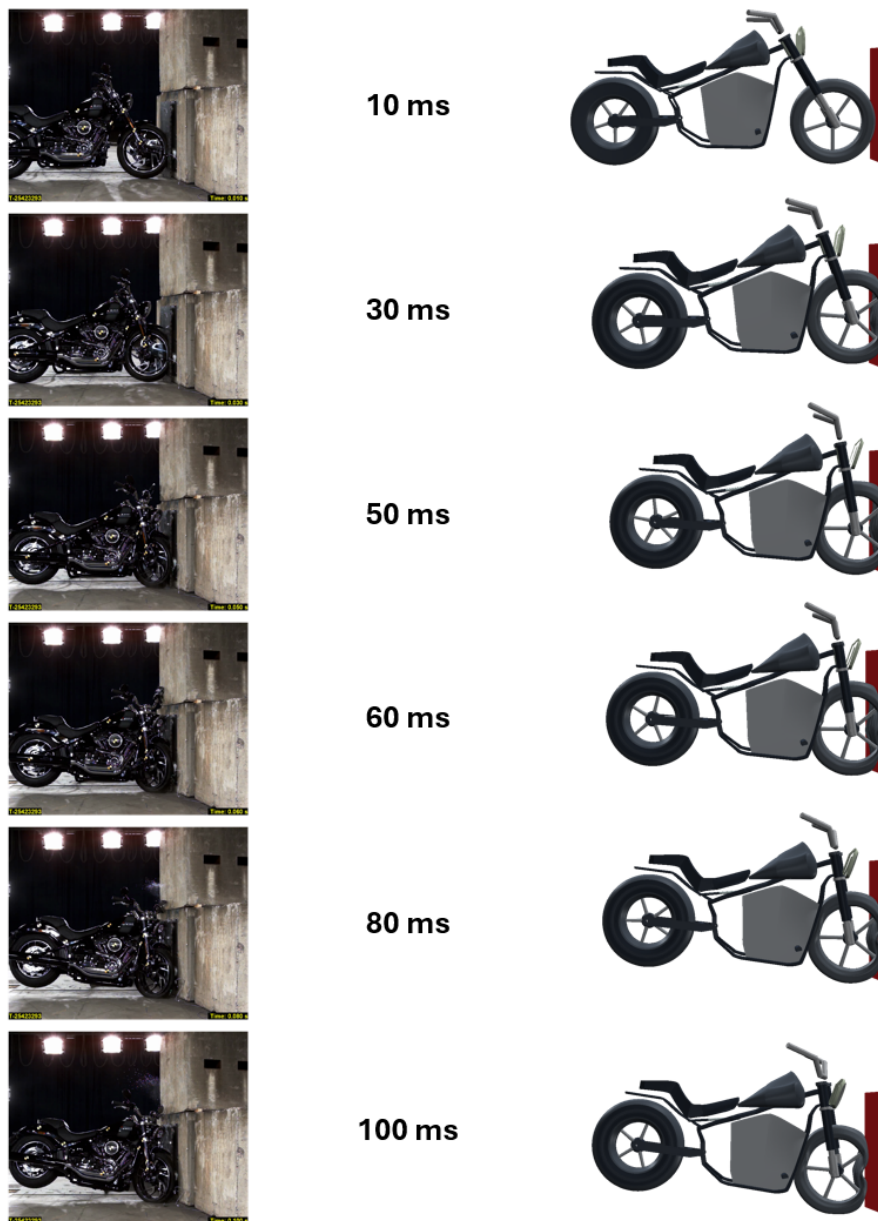


Figure 4.54: Side by side comparison between the physical crash test and the simulation at selected time stamps, Harley-Davidson sport glide.

The results of the rigid-wall simulations using the Harley-Davidson model are shown in Figures 4.56–4.57. Figure 4.55 shows that the physical test result contains three distinct force peaks, with local minima between them. The first peak is caused by the initial impact of the front wheel, the second by the engine impacting the rear side of the front wheel and pushing it into the barrier, and the third by the front of the MC impacting the barrier.

The total force on the RW can be seen in Figure 4.55. It can be seen that the force in the test decreases momentarily after each peak, which is because the energy is absorbed by the MC. Although the general shape of the curve from the simulation

results is similar to the test results, the simulation results lack this unloading of the RW. This is because of the low thickness and yield stress of the rim of the front wheel. The rim cannot absorb energy to the same extent in the simulation as in the test before plasticizing, which lessens the unloading of the RW. The reason for having a low thickness and yield stress was because it was observed that the front wheel would not rotate around the steering axis if it did not plasticize. It can also be seen that the second and third peaks differ in magnitude and are shifted to the left. The second peak differs by approximately 3 kN in magnitude and is shifted by 6 ms, while the third peak differs by approximately 12 kN in magnitude and is shifted by 12 ms.

The acceleration in the x-direction at the COG is shown in Figure 4.56. The figure shows that the curves agree well, except during the time interval between the engine impacting the front wheel and the front wheel impacting the RW again. This discrepancy is because the front wheel does not rotate around the steering axis to the same extent in the simulation as in the test, which causes the MC to decelerate more in the simulation than in the test. This is reflected in Figure 4.55, where the force in the RW increases during this time interval.

The rotational velocity around the y-axis is shown in Figure 4.57. The simulation and physical test results agree well up until 20 ms, which is just before the engine impacts the front wheel. However, the rotational motion is not momentarily reversed in the simulation as it is in the physical test. This discrepancy is also caused by the front wheel not turning around the steering axis to the same extent in the simulation as in the test. It can also be observed that the rotational velocity is considerably lower after the second impact.

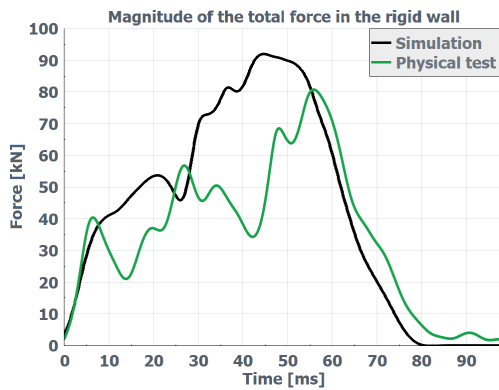


Figure 4.55: Harley-Davidson Sport Glide: Total force in the rigid wall from the physical test and the simulation.

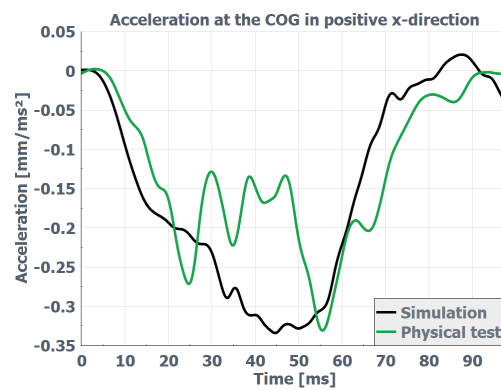


Figure 4.56: Harley-Davidson Sport Glide: Acceleration data at the COG in the positive x-direction from the physical test and the simulation.

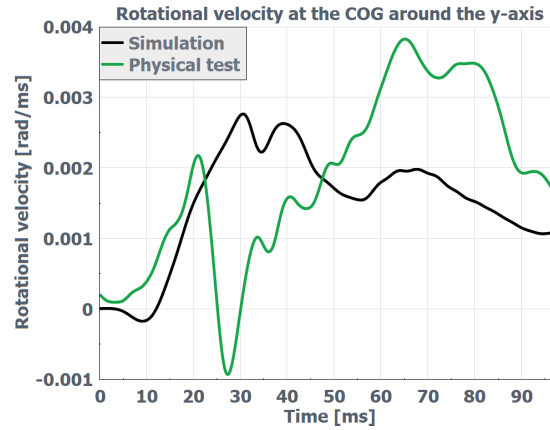


Figure 4.57: Harley-Davidson Sport Glide: Rotational velocity data around the y-axis at the COG from the physical test and the simulation.

4.4.1.2 BMW GS1300

The side by side BMW results are shown in Figure 4.58. The simulation and physical test results exhibit similar overall behaviour, with the most noticeable difference being that the pitch of the real MC is greater than that of the FE model. This difference is most likely caused by differences in the geometry and structural behaviour of the front wheel.

The BMW wheel design is characterised by numerous thin spokes connecting the rim to the hub. In contrast, the generic FE wheel contains five spokes that were intentionally weakened to reproduce the crushing behaviour observed in the physical wheel. Consequently, the FE wheel has fewer load paths between the rim and the hub, which likely reduces the stiffness of the front wheel assembly. This reduction in stiffness alters the effective pivot point of the motorcycle during impact, resulting in a lower pitch response compared with the physical test.



Figure 4.58: Side by side comparison between the physical crash test and the simulation at selected time stamps, BMW GS1300.

The results of the physical test and the simulations are plotted in Figure 4.59. In the plotted data, there are two distinct peaks, with the first corresponding to the initial impact of the tire with the RW. In the physical test data, the smaller peaks after the distinct first peak result from the wheel starting to deform considerably. This behaviour does not generate as many peaks and local minima in the simulation, but the overall shape still matches the physical test relatively well. Lastly, the

motorcycle pitches into the wall and generates the final, largest peak.

The first peak is less pronounced than in the physical test data, although its amplitude still corresponds well. For the final peak, the amplitude agrees well with the physical test data. A phase difference between the physical test and simulation data is particularly noticeable for the last peak, where the simulation data are shifted to the left by 10 ms. This indicates that the event generating the peak occurs earlier in the simulation than in the physical crash test. This difference can be explained by the physical crash test response, where the fourth peak is caused by the front plastic impacting the RW and failing, while the final peak is caused by the metal connected to the windshield colliding with the RW. Since the FE model does not include failure modelling, these two peaks merge into one in the simulation, as the force is transferred directly from the MC to the RW when the front section impacts it.

The acceleration of the COG in the x-direction is shown in Figure 4.60. The acceleration corresponds reasonably well with the experimental data, although deviations are observed from approximately 40 ms onward. Nevertheless, the overall trend is still captured.

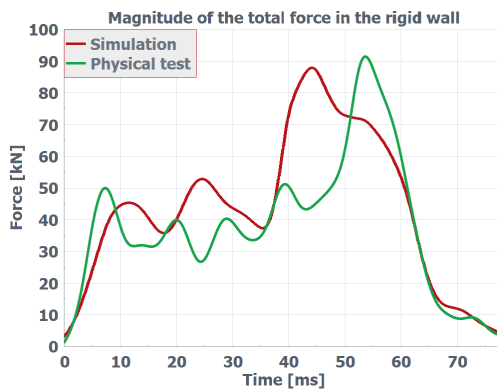


Figure 4.59: BMW GS1300: Total force in the rigid wall from the physical test and the simulation.

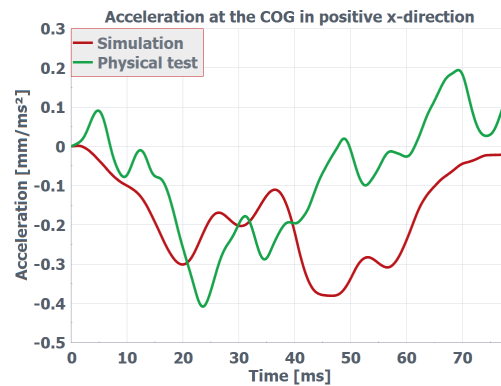


Figure 4.60: BMW GS1300: Acceleration data at the COG in the positive x-direction from the physical test and the simulation.

4.4.1.3 Honda CB125F

Figure 4.61 shows snapshots of the FE model side by side with the corresponding video snapshots from the physical test. The figure shows that the FE model behaves similarly to the real MC, with the largest discrepancy occurring at 30 ms, where the front wheel is positioned further rearward in the physical test. This is caused by the front fork folding more rapidly in the physical test than in the simulation.

Another difference is that both sides of the front wheel rim are deformed in the simulation, whereas the right side of the rim remains seemingly intact and undeformed in the physical test.

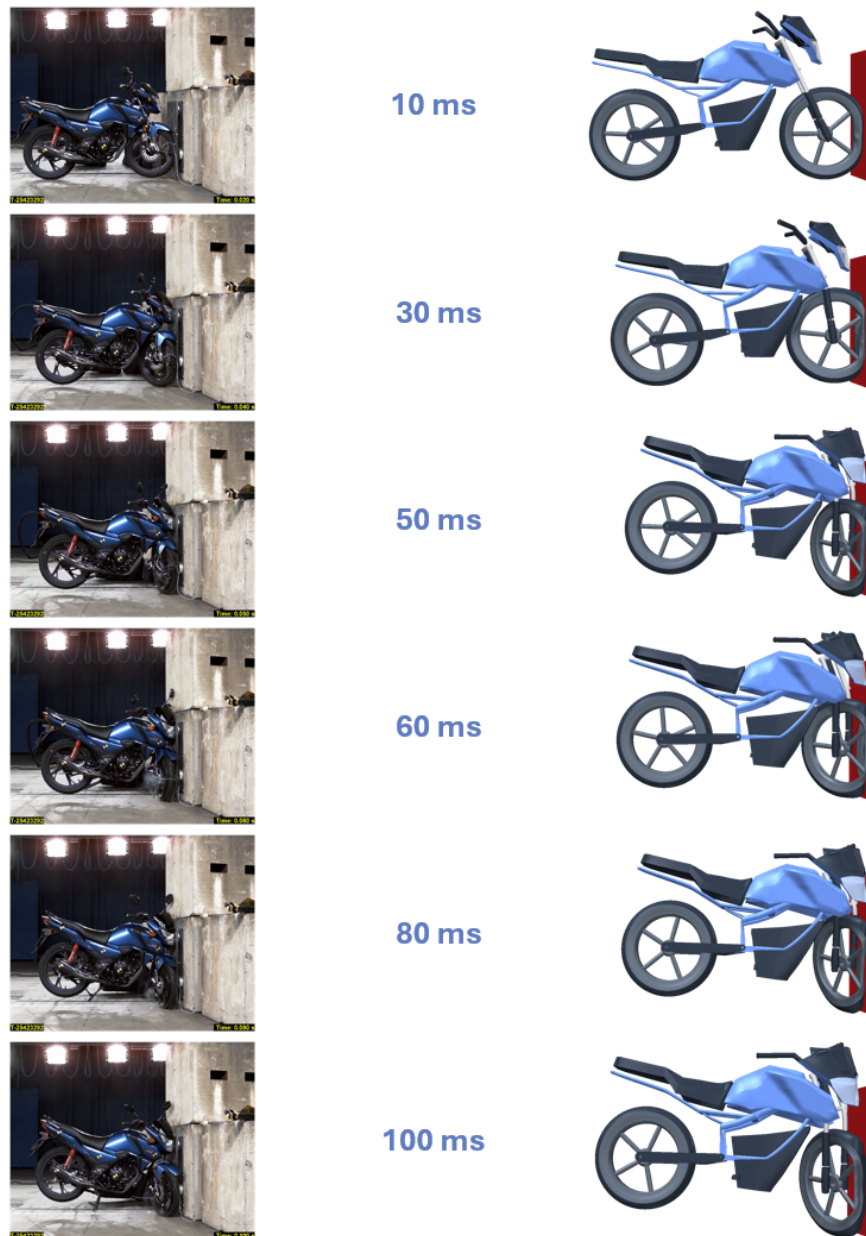


Figure 4.61: Side to side comparison between the physical crash test and the simulation at selected time stamps, Honda CB125F.

In Figures 4.63 and 4.62, the x-direction acceleration at the COG and the total force in the RW are illustrated using plots comparing the simulation data with the physical test data. The sensor data from the test in Figure 4.62 show four distinct force peaks. The first peak occurs when the wheel impacts the RW, followed by an unloading as the wheel rebounds into the engine. The engine then pushes the wheel back toward the RW, generating a second force peak. As the wheel rotates, the RW is momentarily unloaded. The third peak is caused by the front of the MC impacting the RW, while the deformation of the front results in the third unloading. Finally, the fourth peak occurs when the front fork reaches the RW, which is made possible by the rotation of the wheel around the steering axis. After this peak, the motorcy-

cle rebounds from the RW, thereby unloading it, which is reflected by the force drop registered by the sensors on the RW. The simulation matches the amplitude of the last peak, but is phase shifted to the left by 10 ms, meaning that the largest force peak occurs earlier for the simulation than for the physical test. This is because, in the simulation, the front section and front fork impact the RW with less delay compared to the test. This causes the third and fourth force peaks to merge into a single peak in the simulation. Regarding the response before the final peak, the simulation captures the overall behaviour but fails to capture all of the individual peaks.

The acceleration of the COG in the x-direction can be seen in Figure 4.63, which shows that the simulation and test results match well. The physical test has a higher maximum deceleration magnitude of -0.5 mm/ms^2 compared to approximately -0.38 mm/ms^2 for the simulation.

The rotational velocity of the COG is plotted for both the physical test and the simulation in Figure 4.64. The largest difference between the simulation and the physical test data occurs at the largest peak in the simulation. Overall, the physical test data appear to follow the trend of the simulation, with the time-span of 0-30 ms showing good agreement, as does the interval from 65 ms onward. However, during this later interval, the simulation has a lower rotational velocity than indicated by the physical test data.

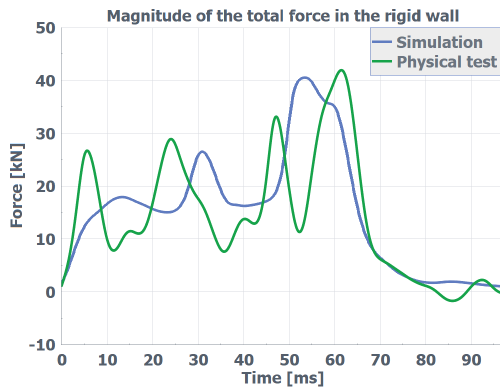


Figure 4.62: Honda CB125F: Total force in the rigid wall from the physical test and the simulation.

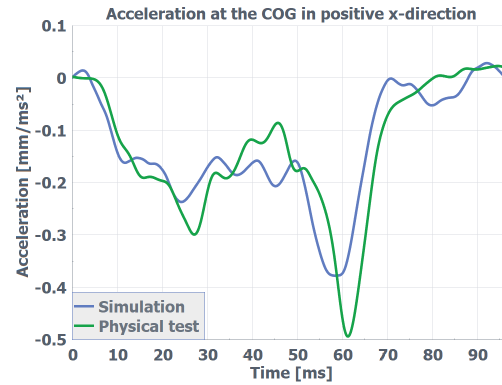


Figure 4.63: Honda CB125F: Acceleration data at the COG in the positive x-direction from the physical test and the simulation.

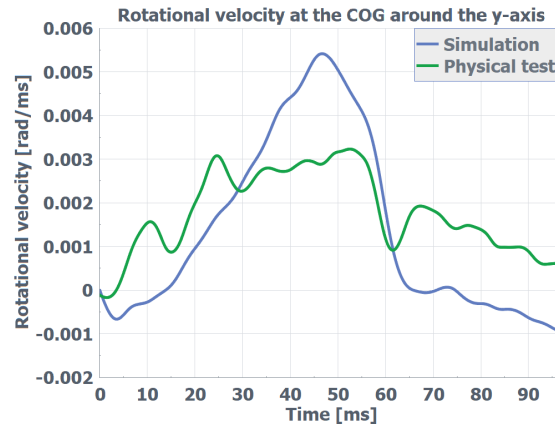


Figure 4.64: Honda CB125F: Rotational velocity data around the y-axis at the COG from the physical test and the simulation.

4.4.1.4 KTM Duke 390

In Figure 4.65, the video from the physical test is presented side by side with the simulation results and divided into six distinct snapshots at different points in time.

A closer examination of Figure 4.65 shows similar behaviour at 9.7 ms, where the wheels have made initial contact with the RW. At 21.7 ms, the front wheel of the MC in the simulation is positioned higher in the z-direction than in the physical test, and the rear part of the MC has started to pitch. In contrast, the rear wheel in the physical test remains in contact with the ground. This difference in rear-end motion continues at 31.3 ms, where the rear wheel in the simulation has moved further in the positive z-direction compared with its position at 21.7 ms, while the rear wheel in the physical test is still in contact with the ground. At this time, both front wheels have moved downward and appear to align more closely in the z-direction compared with the snapshot at 21.7 ms. The front wheel in the physical test is also in contact with the ground at 31.3 ms.

At 41.4 ms, both front wheels are in contact with the ground, and the MC in the physical test shows a distinct pitch, with the rear wheel no longer in contact with the ground. At 60.7 ms, the MC in both the simulation and the physical test is moving away from the RW. At the same time, the rear pitch has increased in both cases compared with the snapshot at 41.4 ms. Finally, at 98 ms, neither MC is in contact with the wall, and the rear pitch appears to align more closely than at 60.7 ms. However, a distinct difference in the x-direction position between the two MCs is visible at 98 ms, as well as across all snapshots except at 9.7 ms, where the MCs appear to align in both the x- and z-directions.

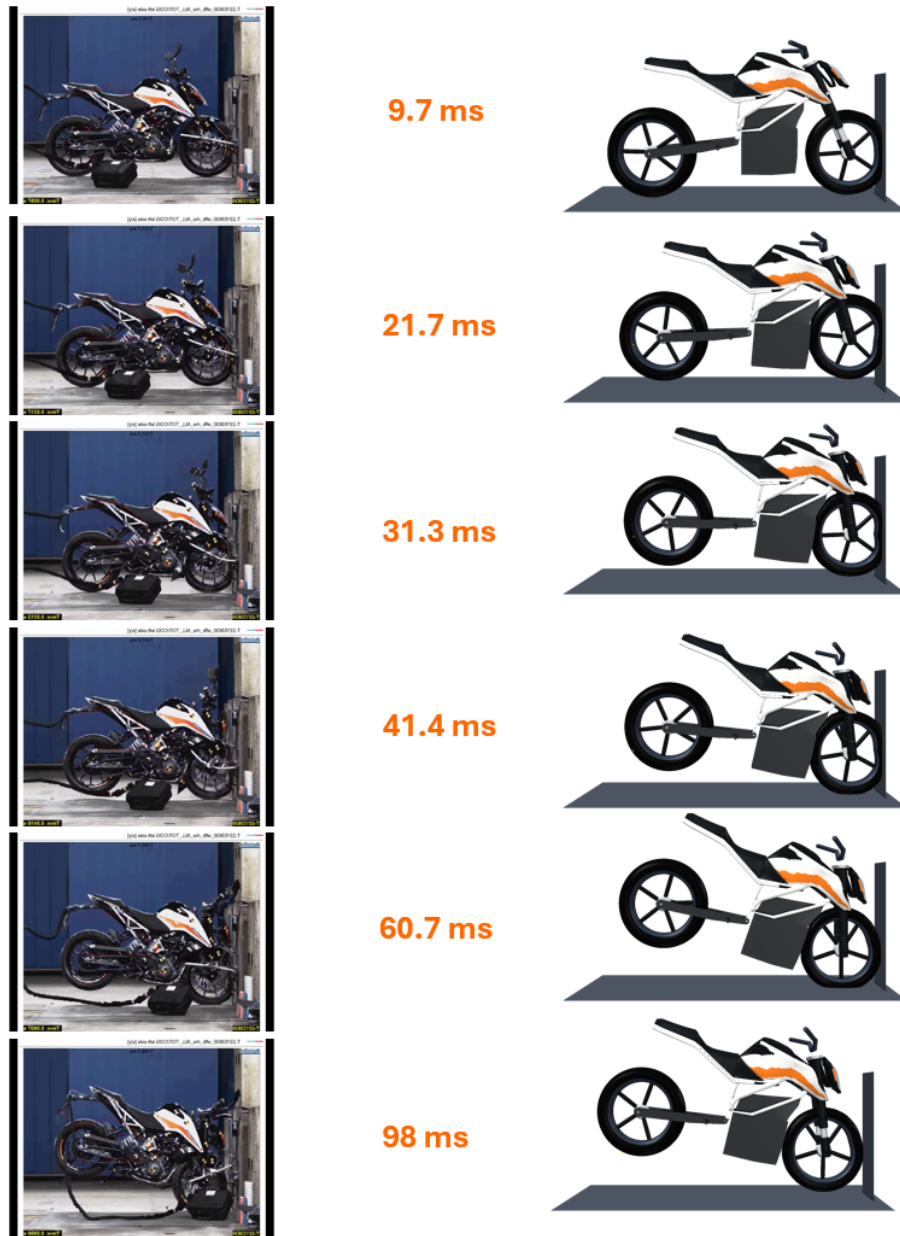


Figure 4.65: Side by side comparison between the physical crash test and the simulation at selected time stamps, KTM Duke 390.

The COG displacement in the x- and z-directions is illustrated in Figures 4.66 and 4.67, respectively. The figures show similar behaviour in terms of shape, although the results are shifted for both displacement components. A closer examination of the x-displacement in Figure 4.66 shows that the simulation and physical test outputs follow a similar displacement trend, with comparable values up to approximately 15 ms, after which the results start to deviate. The peak x-displacement in the simulation occurs approximately 10 ms before the peak in the physical test and has a lower magnitude. The deviation between the x-displacement results appears to reach its maximum at around 50 ms and remains approximately constant until the end of both the simulation and the physical test data.

Upon closer examination of Figure 4.67, it can be observed that the displacement outputs show similar values up to approximately 10 ms, after which the results start to deviate. After 10 ms, the COG in the simulation begins to move in the negative z-direction and reaches its lowest value at around 30 ms. In contrast, the COG in the physical test moves in the positive z-direction, with a small peak occurring at approximately the same time as the minimum value in the simulation. The difference in direction and magnitude between the simulation and physical test is explained by the front suspension of the MC having a preload while the front suspension in the simulation does not, resulting in a negative z-displacement and causing the absolute difference in displacement between the simulation and physical test to increase considerably. After roughly 30 ms, the z-displacement in the simulation increases and appears to approach the displacement observed in the physical crash test. This trend is visible from approximately 30 ms until the end of both the simulation and the physical test data.

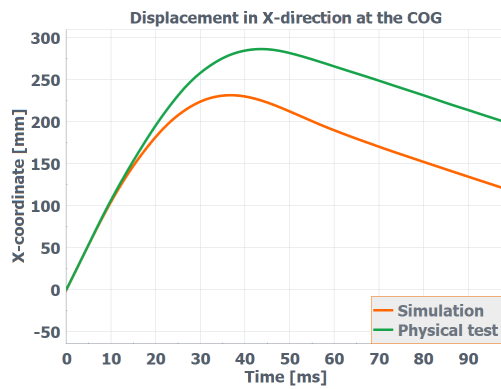


Figure 4.66: KTM Duke 390: Displacement data at the COG in the positive X-direction from the physical test and the simulation, normalized at time 0.

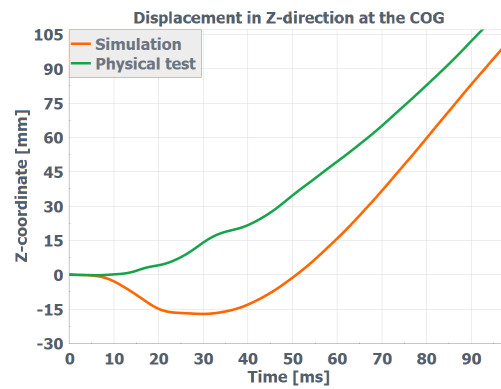


Figure 4.67: KTM Duke 390: Displacement data at the COG in the positive Z-direction from the physical test and the simulation, normalized at time 0.

The force on the barrier and the linear acceleration are shown in Figures 4.68 and 4.69, respectively.

A closer examination of the total force on the RW shows two distinct peaks in the simulation results. The first peak occurs as the MC impacts the wall. After the first peak reaches its maximum, the wheel rebounds into the engine, reducing the contact force between the RW and the MC and thereby unloading the wall. This is reflected by the following local minimum in the simulation.

The second and final peak arises as a continuation of this rebound sequence. After the wheel impacts the engine, it rebounds back toward the wall, generating the final force peak. Once this peak is reached, the load from the MC begins to decrease as the MC starts to pitch. This is illustrated by the final force drop following the last peak.

The force registered in the physical test shows similar behaviour in terms of the force peaks, particularly for the first peak and subsequent unloading. However, the physical test data show a sharper peak than the simulation, and this peak also occurs earlier. The first local minimum, corresponding to the unloading caused by the wheel rebounding from the wall, is also sharper and occurs earlier than in the simulation. The region after the first local minimum, starting at approximately 17 ms, contains several peaks and local minima and corresponds to the phase where the wheel is in contact with the engine and rebounds back toward the wall after the initial rebound indicated by the first local minimum at around 12 ms.

The acceleration registered on the frame in the simulation appears to show similar behaviour to the physical test data, as illustrated in Figure 4.69. In the first segment, from the start of the impact until approximately 30 ms, the peaks and local minima appear to align well, although some differences in magnitude can be observed. After 30 ms, the peaks and local minima are less distinct, but the general shape of the simulation and physical test curves remain relatively similar.

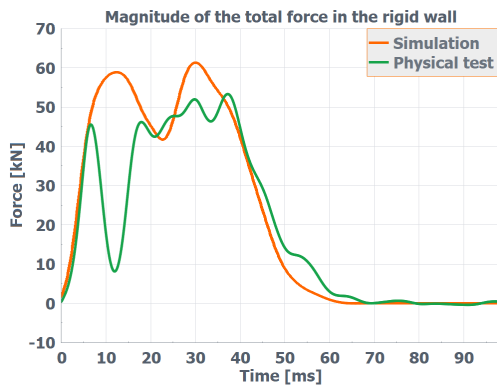


Figure 4.68: KTM Duke 390: Total force in the rigid wall from the physical test and the simulation.

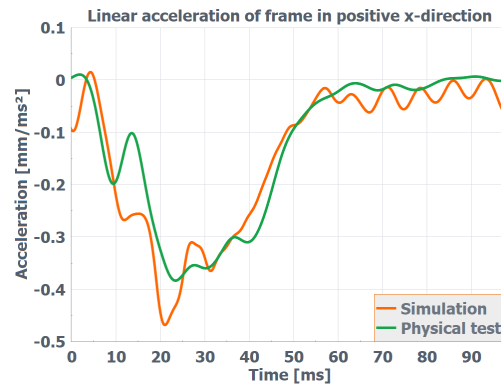


Figure 4.69: KTM Duke 390: Linear acceleration of the frame in x-direction from the simulation and physical test.

4.4.2 Validation of KTM Duke 390

In Figures 4.70 and 4.71, the resultant acceleration and x-direction acceleration of the MC from the simulation and the physical test are presented.

An analysis of the resultant acceleration in Figure 4.70 shows that the simulation and the physical test agree relatively well. In several regions, the simulation either aligns well with the physical test or deviates slightly from the test mean while remaining close to or within the test corridor. The most noticeable difference can be seen between roughly 25 ms and 40 ms, where the simulation response appears to occur earlier than that of the physical test. This can be seen in the difference in time between the first local minimum in the physical test and simulation, followed by the second peak and second local minimum for both results. The magnitude of the first local minimum, second peak, and second local minimum also differs between the

two results. After 40 ms, the results realign, and the simulation appears to capture the overall behaviour and magnitude of the physical test.

An analysis of the x-acceleration in Figure 4.71 shows that the results from the simulation and physical test align well, and the simulation captures the general behaviour of the physical test while aligning the force peaks and local minima relatively well. After roughly 40 ms, the simulation remains, for the most part, within the physical test corridor. The largest differences occur between just before 25 ms and 40 ms. The first local minimum right before 25 ms is lower for the simulation compared to the physical test, but it aligns well in terms of time. The following force peak is slightly earlier for the simulation, which is also true for the subsequent force peak, which occurs at around 30 ms for the simulation and slightly later for the physical test. Around 40 ms, the following local minimum can be seen for both the simulation and the physical test, and these appear to align well in terms of time, although a difference in magnitude is observed.

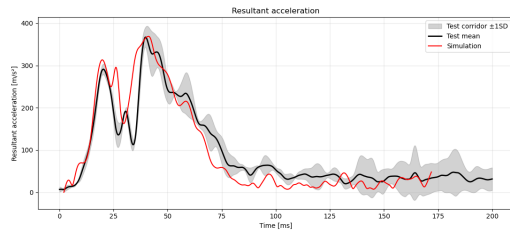


Figure 4.70: Resultant acceleration of the MC from the simulation and physical test.

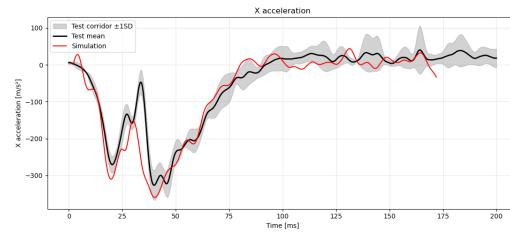


Figure 4.71: X-acceleration of the MC from the validation simulation and the physical test.

The y- and z-acceleration from the simulation and the physical test are presented in Figures 4.72 and 4.73.

A closer examination of the y-acceleration in Figure 4.72 shows that the overall behaviour agrees well. The similarity between the simulation and physical test is most apparent in the regions where the response is most pronounced, as both results contain a comparable number of peaks and local minima. These regions mainly occur from the start of the simulation to approximately 100 ms. The timing of these peaks and local minima also corresponds relatively well with the physical test, although some occur slightly earlier or later in the simulation. One example is the peak around 38 ms, which occurs slightly earlier in the simulation.

The magnitudes of the peaks and local minima also differ between the simulation and the physical test during the first 100 ms. After 100 ms, the comparison becomes slightly less distinct, but the simulation generally remains within the physical test corridors, with despite some deviations outside the corridors.

Further examination of the z-acceleration in Figure 4.73 shows that the results differ in magnitude, especially during the period between roughly 20 ms and 40 ms. The peaks and local minima during that same period appear to align in terms of time,

4. Results

with the local minimum at 25 ms being the first pronounced local minimum during that period. From 40 ms to roughly 115 ms, the peaks and local minima are less noticeable, but the general behaviour of the two data sets follows the same pattern, except for the period from approximately 85 ms to 110 ms, where the simulation result fluctuates around 0, while the physical test result has a noticeable local minimum and increase during that same period. The next segment extends from roughly 115 ms onward, where the simulation result stays within the physical test corridor.

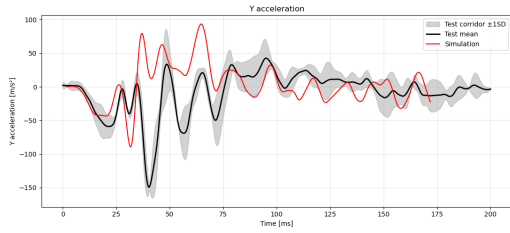


Figure 4.72: Y-acceleration of the MC from the validation simulation and the physical test.

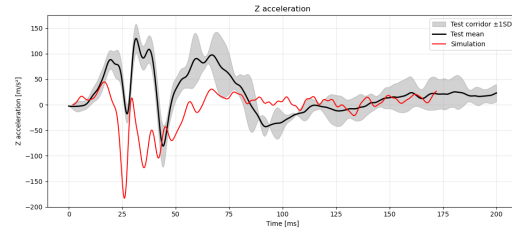


Figure 4.73: Z-acceleration of the MC from the validation simulation and the physical test.

Figure 4.74 presents a side-by-side comparison of the physical test and simulation results, with the simulation results shown in the rightmost column. At 30 ms, the front wheel has rotated around the steering axis a similar amount in both the physical test and the simulation. However, the simulation snapshot indicates that the MC has begun to pitch earlier than in the physical test.

At 45 ms, the front wheel has rotated around the steering axis further in both the simulation and the physical test. This is particularly apparent in the simulation, where the side of the car appears to have been unloaded. In addition, the MC in the simulation appears to have started reducing its pitch, as indicated by the lower position of the engine compared with its position at 30 ms.

At 60 ms, the front wheel has rotated around the steering axis is less pronounced than between 30 ms and 45 ms. The wheel has also moved further into the car compared with the snapshot at 45 ms, which is indicated in the simulation by the increased deformation of the car structure. The deformation of the car is less apparent in the physical test than in the simulation. However, when comparing the wheel position between 45 ms and 60 ms in the physical test, only a small change is observed. The pitch has increased in the simulation at 60 ms compared with 45 ms, whereas the pitch in the physical test appears to remain relatively unchanged.

At 75 ms, the wheel rotation remains limited in both the physical test and the simulation, while the pitch has increased in both results compared with 60 ms. A similar trend is observed between 75 ms and 90 ms, and this trend appears to continue when comparing the snapshots at 90 ms and 105 ms.

At 120 ms, the MC in the physical test appears to have reached its maximum pitch. In the simulation, the pitch remains relatively unchanged compared with the

result at 105 ms. The wheel rotation also remains relatively unchanged in both results compared with the snapshots at 105 ms.

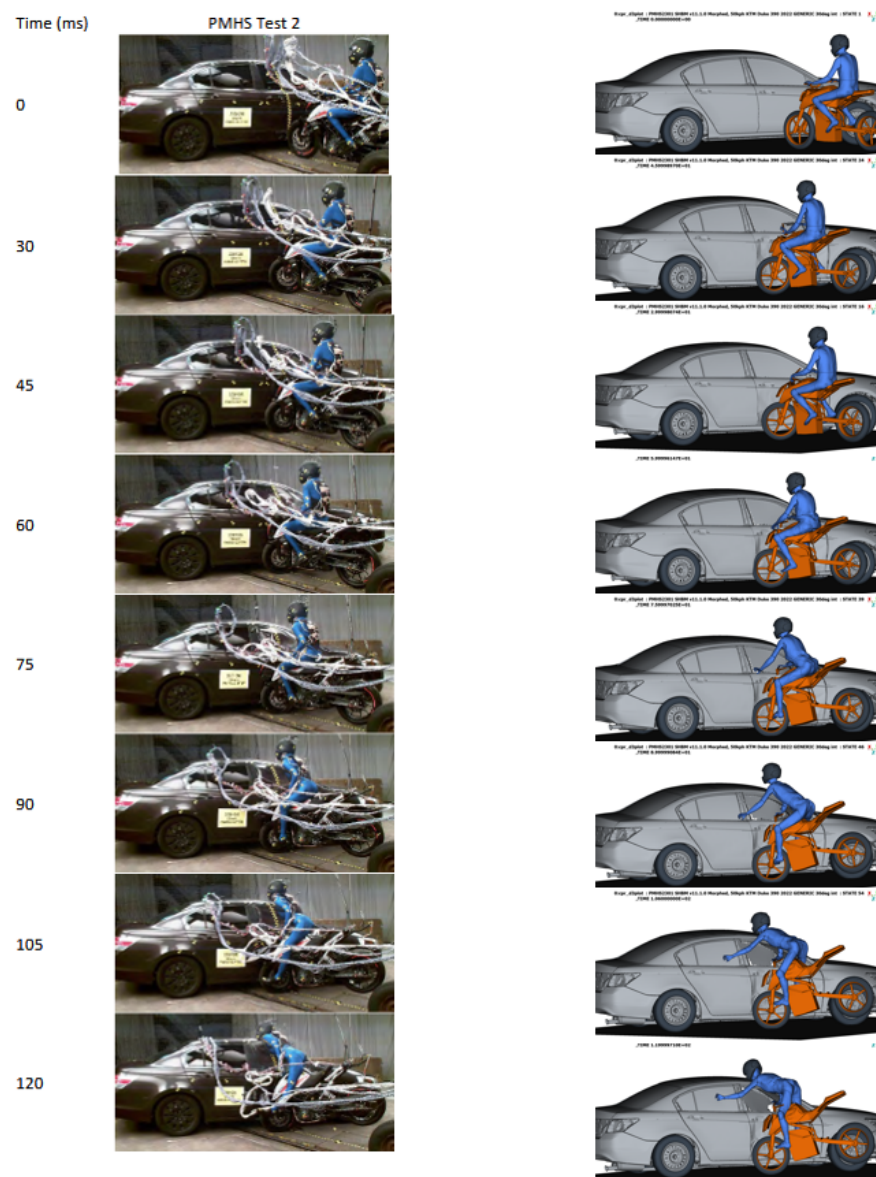


Figure 4.74: Side-to-side comparison between the physical crash test and the simulation at selected time stamps, KTM Duke 390.

5

Discussion

The previous section presented the generated motorcycle geometries and the results obtained in simulated motorcycle crashes, which were compared with the corresponding experimental results. These comparisons highlighted both similarities and differences between the numerical predictions and the test outcomes.

This section discusses the most important sources of error that may explain the observed discrepancies, the arguments for using a generic morphable base model, and the modelling challenges encountered during the project. Furthermore, potential directions for future development, the validation results, and the representativeness of the model are discussed.

5.1 Sources of error

There are some differences between the results from the FE simulations and the real MC crash tests. Some of these differences can be reduced through additional calibration, but this would require more time. The other sources of error will be discussed in this section.

The wheel geometry has a significant influence on the crash result, and which varies from MC to MC. The main difference between the MCs that the FE model cannot capture is the variability in the number of spokes in the different wheels. During the tuning, it was found that the rotation of the front wheel could have a large influence on the crash outcome. This is because, if the wheel is impacted where a spoke is located, then the wheel acts stiffer, and the energy is transferred through the wheel without substantial deformation. It also affects the turning of the front wheel and front fork because the spoke makes the connected rim stiffer, which leads to less rim deformation and was observed to lead to less turning.

The wheel material parameters are also important for the crash outcome. During tuning, it was found that stiffer tire rubber leads to better unloading after the first impact, but the stiffness of the tire inhibits the front wheel's ability to turn. This happens because, for the front wheel to turn, the tire rubber must be compressed and stretched, which it cannot do if the material is too stiff. It should also be noted that the wheel tire of real MCs have a composite structured which the simplified generic model might not be able to replicate.

The rim of the front wheel and tire walls are connected via node-coupling, which, together with the absence of failure modelling, prevents the tire from separating from the rim. This behaviour was observed in the physical crash tests, but it was not modelled in the FE model because it was deemed too complex and difficult to represent accurately. This led to the tire rubber restraining the front wheel rim, thereby limiting or slowing its rotation.

The physical crash tests showed that some components experienced failure and fracture. The areas that exhibited failure in the tests were the sides of the front wheel rim and the front fork of the BMW. It was decided that the FE model would not include failure modelling due to its complexity. As a result, the model cannot capture this behaviour in the same way. Instead, this behaviour was approximated by adjusting the material yield stress and tangential modulus.

The adjustment of the material yield stress and tangential modulus partly improved the response, but did not fully reproduce the physical behaviour. Areas that failed in the tests remained present in the simulation, meaning that the load path was maintained. The difference is that if an element fails, then additional energy cannot be absorbed by the element, nor can it be transferred to connected elements. However, if the element plasticizes, then, if the element is unloaded, it can again absorb energy and deform elastically until the yield stress is reached.

A consequence of not modelling failure was that it made the rim initially too weak due to an effort to compensate for the failure occurring in the physical test. In some tests, the rim deformed on the left side during the initial impact, which caused the left side of the rim to plasticize and partly fail. The second impact then caused the left side of the rim to fully fail. This led to unbalanced forces acting on the front wheel, causing it to rotate around the steering axis.

This was observed in the simulations, but to a lesser degree. Similar behaviour was observed in the simulations, the difference being that both sides of the rim deformed during the initial impact, but one side deformed more. During the second impact, the wheel turned, but both sides became significantly deformed. Since both sides were deformed, the imbalance of the forces acting on the wheel was reduced, which led to a decrease in the rotational velocity or angle of the front wheel around the steering axis.

The z-displacement of the KTM Duke 390 initially decreased before increasing in the simulations, and this discrepancy may be attributed to the front fork not having a prescribed pre-tension. This is also applicable to the other FE models and may therefore represent an additional source of error. This can be addressed by additional calibration.

The FE models exhibited mesh quality issues stemming from distortion of the base mesh. These quality issues could lead to incorrect stress and strain calculations and stress distributions on the elements. This, in turn, could lead to incorrect de-

formation. The effect of this incorrect deformation can be inhibited by calibrating the structural stiffness. For example, if the deformation of certain part of the mesh with quality violating elements is too great then the thickness of this part can be increased in order to inhibit this behaviour. Table 4.2 shows that there were relatively few shell elements with mesh quality issues and this together with the fact that calibration can solve the incorrect deformations shows that the impact of the quality violating elements is likely insignificant, although it may still have some effect. Table 4.3 shows that a high proportion of the solid elements have mesh quality issues, but since these are all located in the saddle and that calibration can also solve this problem, these issues are deemed insignificant. It should also be noted that the quality violating elements of the final meshes did not cause premature termination of the simulations which also points to the effect of these elements to be insignificant. The influence of these mesh quality issues could be evaluated by re-meshing the models and rerunning the simulations. However, this was not investigated within the scope of this project due to time constraints.

5.2 Base model

One of the goals of the project was to create a generic base model. This was achieved partly by determining mean geometric values across different MC types, but also by ensuring that all morphed models retained the same mesh, with an identical number of nodes and elements.

One could argue that simply re-meshing each morphed model to the user's preferences would be relatively straightforward and would avoid the distorted elements that result from the morphing process. In many cases, this would likely produce a more accurate mesh. However, re-meshing would remove the definitions inherited from the base setup. These definitions include carefully established constraints and settings that are required for a functional MC model. Having to recreate all of these definitions for every re-meshed model would make the workflow significantly less accessible and require additional effort from the user. Since the morphed meshes have been shown to produce comparable results, retaining the same mesh structure is a practical solution. It ensures that all definitions remain consistent across all morphed models, regardless of MC type.

Additionally, if a user generates several morphed models and discovers that a particular definition is missing from all of them, setting up the definition in a single model is often sufficient. Because the node numbering remains identical across all models, the definition can then be copied directly to the remaining models with minimal effort.

It can also be argued that each FE model could be created separately, which could result in more accurate geometries and, in turn, more accurate results. However, this approach would be more time-consuming, as the user would have to start from the beginning each time a new FE model is to be created, rather than modifying a base geometry with predefined connections. The process of creating an MC FE

model is therefore made more accessible and less time-consuming through the use of the generic morphable MC model.

5.3 Modelling challenges

This section will discuss the model-specific challenges associated with each of the four MCs.

5.3.1 Harley-Davidson sport glide

The Harley-Davidson weighs 298 kg, which is considerably heavier than the other MC models; for comparison, the Honda weighs 107 kg. This, combined with a rake angle of 30° , made it challenging to tune the model so that the front wheel would turn. During tuning, it was found that if the front wheel rim or the tire rubber was made too stiff, then the front wheel would bounce and rotate until it became orthogonal to the wall instead of sliding along it. This behaviour led to lowering the thickness, yield stress, and tangential modulus of the rim material, as well as the stiffness of the rubber, which made the wheel turn more but worsened the unloading of the barrier.

5.3.2 HONDA CB125F

The test result showed that the front wheel turned almost 80 degrees, becoming almost parallel to the wall. The front wheel could turn this much because the front fork folding was caused by the wheel bouncing upwards whereas for the other models, it did so by the front fork moving downwards. This is most likely because the front fork is an inverted fork for this model. This made the wheel airborne, and when the engine hit it, it could turn without hitting the ground. This made tuning the model challenging, as many parameters influenced how the wheel behaved, such as the damper and spring stiffness, the stiffness of the front wheel rim and tire rubber, and the stiffness of the frame. It was also made challenging by the fact that, in the test, the tire separated from the rim, and this could not be modelled.

5.3.3 BMW GS 1300

The BMW had several modelling challenges, but much of its behaviour upon impacting the RW could be controlled by adjusting parameters related to the wheel. One aspect that made the BMW stand out from the rest of the models was how the front tire and wheel as a whole deformed. This can be seen in Figure 4.58 from 30 ms onward, where the spokes deform to such an extent that the front of the rim makes contact with the hub, caused by the substantial deformation of the spokes. In the physical test, the spokes also penetrated several points of the tyre itself, a behaviour that was not captured in the simulation. Because the spokes seemed to govern much of the behaviour of the wheel, calibration efforts were allocated in particular to these. It was noted that the spokes in the physical MC were relatively thin compared with both the other MCs and how the geometry was defined in the

generic MC model, where the geometry was significantly larger in terms of diameter compared with the real MC. To replicate its geometric features and stiffness, the behaviour was largely controlled through adjusting the shell thickness of the spokes.

The top part of the forks also deformed significantly during the physical test, to the extent that the outer part of the front fork cracked on the left side, where most of the force naturally ended up due to the rotation of the wall. To try to replicate this behaviour, the stiffness was adjusted through reduced shell thickness as well as an adjusted yield stress. The timing of this failure plays a crucial role in the ability to match the rotation of the MC into the wall, as the bottom of the front fork was assumed to remain undeformed.

In Figure 4.58, the side-by-side comparison between the simulation and the physical test is presented. It can be seen that the rear wheel in the simulation at around 40 ms is positioned further ahead in the x-direction than in the physical test, as a result of differences in wheel deformation and steering. At this same time, the MC in the physical test has pitched more than in the simulation. This could result from differences in wheel deformation, but also from the ability of the upper part of the front fork to carry force, where a stiffer upper part of the fork might have resulted in the MC in the simulation being able to pitch more. Efforts to calibrate this further were dropped as a result of time limitations.

Additionally, one important difference between the physical test and simulation lies in the way that the front wind shield and headlight and the tank impact the RW. In the physical test, there is an increased peak at around 40 ms, which is illustrated in Figure 4.59. This peak is caused by the plastic details in front of the tank impacting the RW, while the following peak is caused by the tank impacting the RW. The replication of these two events proved difficult and would have required more time to establish a functional replication mechanism.

The wind shield and the front plastic parts in the MC model are connected using rigid body definitions between the tank and these plastic details. Since the material at the front was not calibrated to fracture as the plastic details do in the physical test, the momentum of the MC and rotation of the tank are instead directly transferred through the rigid body connection and into the wall upon initial impact of the wind shield and corresponding plastic details in the model. This explains why the simulation results in its largest peak earlier than the physical test.

5.3.4 KTM Duke 390

The setup and calibration of the KTM were relatively different compared to the three other models, with one difference lying in the rotation of the wall. This introduced its own modelling challenges, as the force upon impact was more directly transferred to the wall and, consequently, to the MC itself. This was further influenced by the KTM's limited steering as a result of the RW having no rotation. Prior to this, a KTM model from another project had been calibrated using the same RW

setup (Gowda et al, 2025). The material definitions, together with the shell definitions, were therefore copied to the model used in this project in an attempt to accelerate the calibration process. One distinct difference between the current and previous model is the number of components. The previous KTM model had more components and was, overall, more complex. Therefore, simply transferring material definitions to comparatively simplified parts did not significantly accelerate the calibration process.

Another distinct difference is that the rim failed in the physical test. This behaviour proved difficult to replicate, and failure was applied in early iterations to the rim in order to attempt to replicate the failure. Ultimately, the failure mechanism was considered too complex to model, and for that reason the rim behaviour was largely calibrated through its material properties. Several aluminium configurations were tested in order to identify the best-performing material as a starting point. The behaviour was subsequently calibrated using the yield stress and hardening stress, which proved to be a successful calibration approach since much of the force response on the RW was governed by the deformation of the rim. During the final calibration iterations, material hardening was removed in an attempt to replicate the observed failure through the implementation of perfect plasticity.

In addition to the rim, the spokes add stability to the rim upon impact with the RW. It was observed that the wheel in the physical test was rotated so that the spokes aligned perpendicular to the RW upon impact, which led to the spokes being a direct force path upon impact, which affected other regions of the structure. The setup for the simulation, on the other hand, had the rim handle much of the direct impact on the RW, with stability provided only by the surrounding spokes. This particular setup is illustrated in Figure 4.65 at 9.7 ms.

5.4 Future development

There are certain areas in which the model could be improved. Most of these areas have been discussed as sources of error, such as improving the wheel geometry. Another possible improvement would be to include failure, but this would require additional material testing of the MC parts or tuning of the individual models. An area of possible improvement that has not been explored in this project is replacing the shell elements with solid elements for the wheel tires.

No additional crash tests have been conducted for the MC models other than the KTM Duke 390, which has limited the project to only one validated model. Validation of these models could thus be an area for future work if desired.

The tuning of the individual FE models was limited by time and could be further refined to improve the accuracy and representativeness of these models.

To further increase the base model's ability to be morphed and represent a wider range of motorcycles, future work could focus on developing a base model that also

allows morphing to scooter geometries, utilizing a step through frame. This was initially part of the project scope; however, because the scooter geometry differed too greatly and the available time was limited, it was deemed infeasible to morph the existing base model into a scooter geometry while maintaining the same quality as the other four MC types.

To enable morphing to scooter geometries, or other geometries similar to scooters, the current base model may either need to be updated to a new version capable of morphing into both scooters and the four existing MC types, or an entirely new base model may need to be developed that is capable of morphing into different scooter geometries or a limited set of other MC types.

5.5 Validation of the KTM Duke 390

The results from the validation simulation match well with the physical test with respect to the resultant acceleration and the x-acceleration (see Figures 4.70 and 4.71). The resultant acceleration is expected to match well if the x-direction acceleration matches well, considering that the x-direction is the governing direction for the overall behaviour. It is somewhat surprising that the validation result matches well for the resultant acceleration but relatively poorly for the force on the RW (see Figure 4.68). To further investigate this observation would potentially require additional validation cases for the KTM Duke 390. Furthermore, validation of the remaining MCs, which have been calibrated in a similar manner, could be used to investigate whether this behaviour is unique to the validation simulation or is exhibited by the other models as well.

5.6 Model representativeness

Comparing the results from the RW tests and the RW simulations shows that the FE models gave somewhat different results, but general trends that aligned with the experimental data could be found for all models. This, combined with the fact that the results from the validation are promising, shows that the models are representative of real MCs in crash scenarios involving frontal crashes. Additional calibration of the FE models is possible, but the results from the graphs and from the d3plots show that the models behave similarly to the real crash tests. Further tests would be needed to validate the representativeness of all the models, but the validation of the KTM Duke 390 shows that it is possible to use this generic FE model, morph it into a specific MC, tune it, and then get results that are similar to those from a crash test.

A comparison of the results from the FE models developed in this project with MC FE models developed in other projects shows that the FE models developed in this project show comparable accuracy (Gowda et al, 2025). The results from these other FE models also differ in terms of phase and magnitude when compared with the physical tests. An example of this is that the largest discrepancy in RW force magnitude for the KTM Duke 390 is around 10 kN, which is the same discrepancy

that can be seen in (Gowda et al, 2025). The discrepancies between simulation and test are generally greater for the FE models developed in this project, yet similar overall behaviour of the FE models compared with the tested MCs was achieved.

6

Conclusion

Many challenges were encountered during this project. Many of these stemmed from the main objective of the project: to create a generic FE model that can be morphed into a wide range of shapes while still using the same mesh. The resulting model has been shown to be representative not only geometrically, but also through front end stiffness.

The FE models developed in this project have been shown to be largely capable of representing real MCs. This has been demonstrated by comparing force, tracking data, angular velocity, and acceleration responses generated from both the simulations and physical tests. It has also been shown by comparing snapshots of the kinematic response from the tests and simulations. Differences between the tests and simulations remain; however, the overall behaviour is comparable and could be improved through further calibration.

Similar overall behaviour is shown by the validation simulation of the KTM Duke 390 model. This simulation demonstrates that it is possible to obtain results similar to those from real tests using the developed FE model. There are some areas in which the FE model is lacking, such as failure behaviour and wheel geometry. However, further development or calibration of the existing FE models could likely improve the model in these areas.

In summary, the model may serve as a useful tool for incorporating MC variability into rider injury risk assessment, motorcycle safety system development, and evaluation of existing safety systems in both academic and industrial contexts.

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A

Appendix

A.1 Structural properties BMW GS1300

Table A.1: BMW GS1300 LS-DYNA structural properties

Material	Name of part	T1 [mm]	RO (ρ) [kg/mm ³]	E [GPa]	PR (ν)	SIGY [GPa]	ETAN [GPa]
MAT24	FRONT-WHEEL-RIM	8.	2.67E-6	72.4	0.33	0.165	0.
MAT27	FRONT-WHEEL-TIRE-TREAD	8.	1.111E-6	—	0.495	—	—
MAT27	FRONT-WHEEL-SIDEWALL	6.	1.111E-6	—	0.495	—	—
MAT24	FRONT-WHEEL-SPOKES	.1	2.67E-6	72.4	0.33	0.165	2.1
MAT24	FRONT-WHEEL-HUB	1.5	2.67E-6	72.4	0.33	0.165	2.1
MAT24	FRONT-FORK-BOTTOM	12.	7.85E-6	210	0.3	0.9	0.6
MAT24	FRONT-FORK-TOP	2.	2.89E-6	71.7	0.33	3.	1.58
MAT67	FRONT-FORK-SPRING	—	1.	—	—	—	—
MATS05	FRONT-FORK-DAMPER	—	—	—	—	—	—
MAT24	FRONT-FORK-TRIPLE-CLAMPS-UPPER	4	2.89E-6	210	0.30	0.9	0.6
MAT24	FRONT-FORK-TRIPLE-CLAMPS-LOWER	4.	2.89E-6	210	0.30	0.9	0.6
MAT24	FRONT-WINDSHIELD	4	1.16E-6	30	0.37	0.07	0.
MAT24	FRONT-METAL	1	7.85E-6	205	0.3	0.9	0.6
MAT24	FRONT-ALUMINIUM	1	2.89E-6	71.7	0.33	0.221	1.58
MAT24	FRONT-HEADLIGHT	4	2.4E-6	62	0.19	0.1	0.555
MAT24	HANDLEBARS	3	7.85E-6	210	0.3	0.	0.
MAT24	FRAME-FRONT	4	7.85E-6	205.	0.3	0.9	0.6
MAT24	FRAME-FRONT-CONNECTOR	1.	7.85E-6	205	0.3	0.9	0.6
MAT24	FRAME-FRONT-BOTTOM	3.	7.85E-6	205	0.3	0.9	0.6
MAT24	FRAME-FRONT-TOP	2.5	7.85E-6	205	0.3	0.9	0.6
MAT24	FRAME-FRONT-TOP-BARS	3.	7.85E-6	205	0.3	0.9	0.6
MAT20	FRAME-REAR	3.	7.85E-6	205	0.3	—	—
MAT20	FRAME-SWINGARM	4.	2.89E-6	205.	0.3	—	—
MAT67	FRAME-REAR-SPRING	—	1.0	—	—	—	—
MATS05	FRAME-REAR-DAMPER	—	—	—	—	—	—
MAT24	FUEL-TANK	4.	7.85E-6	205.	0.3	0.9	0.6
MAT34	SEAT-SKIN	2.	9.0E-9	—	—	—	—
MAT83	SEAT-FOAM	—	4.8E-7	0.138	—	—	—
MAT20	PEDALS	1.	7.85E-6	205.	0.3	—	—
MAT20	ENGINE	4.	7.85E-6	205.	0.3	—	—
MAT27	BACK-WHEEL-TIRE-TREAD	10.	1.111E-6	—	0.495	—	—
MAT27	BACK-WHEEL-TIRE-SIDEWALL	4.	1.111E-6	—	0.495	—	—
MAT24	BACK-WHEEL-SPOKES	4.	2.67E-6	72.4	0.33	0.165	2.1
MAT24	BACK-WHEEL-HUB	3.	2.67E-6	72.4	0.33	0.165	2.1
MAT24	BACK-WHEEL-RIM	4.	2.67E-6	72.4	0.33	0.165	2.1

A. Appendix

Material model abbreviations: MAT24 = Piecewise Linear Plasticity; MAT20 = Rigid; MAT27 = Mooney-Rivlin Rubber; MAT67 = Nonlinear Elastic Discrete Beam; MATS05 = Nonlinear Viscous Damper; MAT34 = Fabric; MAT83 = Fu Chang Foam.

Mooney-Rivlin parameters:

Front-wheel tread: $A = 0.0104$, $B = 0.0101$

Front-wheel sidewall: $A = 0.0108$, $B = 0.0102$

Back-wheel tread: $A = 0.0104$, $B = 0.0101$

Back-wheel sidewall: $A = 0.0108$, $B = 0.0102$.

Spring and damper parameters:

Spring: $SPRING_SC = 0.0001$;

Damper: $DAMP_SC = 0.001$;

A.2 Structural properties Harley Davidson Sport Glide

Table A.2: Harley-Davidson Sport Glide LS-DYNA structural properties

Material	Name of Part	T1 [mm]	RO (ρ) [kg/mm ³]	E [GPa]	PR (ν)	SIGY [GPa]	ETAN [GPa]
MAT27	FRONT-WHEEL-TREAD	8	2.E-6	–	0.495	–	–
MAT27	FRONT-WHEEL-SIDEWALL	10	2.E-6	–	0.495	–	–
MAT24	FRONT-WHEEL-SPOKES	12	2.67E-6	72.3	0.33	0.165	2.1
MAT24	FRONT-WHEEL-RIM	4	2.67E-6	71.	0.33	0.165	0.35
MAT24	FRONT-WHEEL-HUB	12	2.67E-6	72.3	0.33	0.165	2.1
MAT24	FRONT-FORK-TOP	6	7.85E-6	210.	0.3	0.9	0.6
MAT24	FRONT-FORK-LOWER	8	7.85E-6	210.	0.3	0.9	0.6
MAT67	FRONT-FORK-SPRING	–	1.	–	–	–	–
MATS05	FRONT-FORK-DAMPER	–	–	–	–	–	–
MAT24	FRONT-FORK-TRIPLE-CLAMPS	12	2.89E-6	210.	0.3	0.9	0.6
MAT24	FRONT-WINDSHIELD	4	1.16E-6	30.	0.37	0.07	0.
MAT24	FRONT-PLASTIC	4	1.415E-6	1.	0.3	0.02	0.01
MAT24	FRONT-ALUMINIUM	4	2.89E-6	71.7	0.33	0.221	1.58
MAT24	FRONT-HEADLIGHT	4	1.4E-6	7.3	0.3	0.091	0.555
MAT24	HANDLEBARS	3	7.85E-6	210.	0.3	0.	0.
MAT24	FRAME-FRONT-BOTTOM	8	7.85E-6	205.	0.3	0.9	0.6
MAT20	FRAME-REAR	3	7.85E-6	205.	0.3	–	–
MAT20	FRAME-SWINGARM	4	2.89E-6	205.	0.3	–	–
MAT24	FRAME-FRONT-TOP	8	7.85E-6	205.	0.3	0.9	0.6
MAT67	FRAME-REAR-SPRING	–	1.	–	–	–	–
MAT24	FRAME-FORK-CONNECTOR	10	7.85E-6	205.	0.3	2.	1.8
MATS05	FRAME-REAR-DAMPER	–	–	–	–	–	–
MAT24	FRAME-TOP-BAR	3	7.85E-6	205.	0.3	0.9	0.6
MAT24	FUEL-TANK	3	7.85E-6	205.	0.3	0.9	0.6
MAT34	SEAT-SKIN	2	9.E-9	–	–	–	–
MAT83	SEAT-FOAM	–	4.8E-7	0.138	–	–	–
MAT20	PEDAL	1	7.85E-6	205.	0.3	–	–
MAT20	ENGINE	4	7.85E-6	205.	0.3	–	–
MAT27	BACK-WHEEL-SIDEWALL	4	1.111E-6	–	0.495	–	–
MAT27	BACK-WHEEL-TREAD	10	1.111E-6	–	0.495	–	–
MAT24	BACK-WHEEL-HUB	12	2.67E-6	72.4	0.33	0.165	2.1
MAT24	BACK-WHEEL-SPOKES	12	2.67E-6	72.4	0.33	0.165	2.1
MAT24	BACK-WHEEL-RIM	4	2.67E-6	72.4	0.33	1.65	21.

Material model abbreviations: MAT24 = Piecewise Linear Plasticity; MAT20 = Rigid; MAT27 = Mooney-Rivlin Rubber; MAT67 = Nonlinear Elastic Discrete Beam; MATS05 = Nonlinear Viscous Damper; MAT34 = Fabric; MAT83 = Fu Chang Foam.

Mooney-Rivlin parameters:

Front-wheel tread: $A = 0.0104$, $B = 0.0101$

Front-wheel sidewall: $A = 0.0108$, $B = 0.0102$

Back-wheel tread: $A = 0.0104$, $B = 0.0101$

Back-wheel sidewall: $A = 0.0108$, $B = 0.0102$.

Spring and damper parameters:

Spring: $SPRING_SC = 0.001$;

Damper: $DAMP_SC = 0.001$;

A.3 Structural properties Honda CB125F

Table A.3: Honda CB125F LS-DYNA structural properties

Material	Name of Part	T1 [mm]	RO (ρ) [kg/mm ³]	E [GPa]	PR (ν)	SIGY [GPa]	ETAN [GPa]
MAT27	FRONT-WHEEL-TREAD	10	1.111E-6	–	0.495	–	–
MAT27	FRONT-WHEEL-SIDEWALL	8	1.111E-6	–	0.495	–	–
MAT24	FRONT-WHEEL-SPOKES	7	2.67E-6	71.	0.33	0.095	0.35
MAT24	FRONT-WHEEL-RIM	1.5	2.67E-6	71.	0.33	0.13	0.35
MAT24	FRONT-WHEEL-HUB	12	2.67E-6	71.	0.33	0.095	0.35
MAT24	FRONT-FORK-TOP	18.5	2.89E-6	71.	0.33	0.165	0.35
MAT24	FRONT-FORK-MIDDLE	12	2.89E-6	71.	0.33	0.165	0.35
MAT67	FRONT-FORK-SPRING	–	1.	–	–	–	–
MATS05	FRONT-FORK-DAMPER	–	–	–	–	–	–
MAT24	FRONT-FORK-TRIPLE-CLAMP-UPPER	12	2.89E-6	71.	0.30	0.165	0.35
MAT24	FRONT-FORK-TRIPLE-CLAMP-LOWER	12	2.89E-6	71.	0.30	0.165	0.35
MAT24	FRONT-FORK-LOWER	4	2.89E-6	71.	0.33	0.165	0.35
MAT24	FRONT-WINDSHIELD	4	1.16E-6	30.	0.37	0.07	0.
MAT24	FRONT-PLASTIC	4	1.415E-6	1.	0.30	0.02	0.01
MAT24	FRONT-ALUMINIUM	4	2.89E-6	71.7	0.33	0.221	1.58
MAT24	FRONT-LIGHT	4	1.4E-6	7.3	0.30	0.1	0.555
MAT24	HANDLEBAR	3	2.67E-6	71.	0.30	0.165	0.165
MAT24	FRAME-FRONT-BOTTOM	2.5	7.85E-6	205.	0.30	0.9	0.6
MAT20	FRAME-REAR	3	2.89E-6	71.7	0.37	–	–
MAT20	FRAME-SWINGARM	4	2.89E-6	205.	0.30	–	–
MAT24	FRAME-FRONT-TOP	2.5	7.85E-6	205.	0.30	0.9	0.6
MAT67	FRAME-REAR-SPRING	–	1.	–	–	–	–
MATS05	FRAME-REAR-DAMPER	–	–	–	–	–	–
MAT24	FRAME-FORK-CONNECTOR	10	7.85E-6	205.	0.30	0.9	0.6
MAT24	FRAME-TOP-BAR	3	7.85E-6	205.	0.30	0.9	0.6
MAT24	FUEL-TANK	2	7.85E-6	205.	0.30	0.9	0.6
MAT34	SEAT-SKIN	2	9.E-9	–	–	–	–
MAT83	SEAT-FOAM	–	4.8E-7	0.138	–	–	–
MAT20	PEDAL	1	7.85E-6	205.	0.30	–	–
MAT20	ENGINE	4	2.89E-6	71.4	0.37	–	–
MAT27	BACK-WHEEL-SIDEWALL	10	1.111E-6	–	0.495	–	–
MAT27	BACK-WHEEL-TREAD	4	1.111E-6	–	0.495	–	–
MAT24	BACK-WHEEL-RIM	4	2.67E-6	72.4	0.33	0.165	2.1
MAT24	BACK-WHEEL-SPOKES	4	2.67E-6	72.4	0.33	0.165	2.1
MAT24	BACK-WHEEL-HUB	3	2.67E-6	72.4	0.33	0.165	2.1

Material model abbreviations: MAT24 = Piecewise Linear Plasticity; MAT20 = Rigid; MAT27 = Mooney-Rivlin Rubber; MAT67 = Nonlinear Elastic Discrete Beam; MATS05 = Nonlinear Viscous Damper; MAT34 = Fabric; MAT83 = Fu Chang Foam.

Mooney-Rivlin parameters:

Front-wheel tread: $A = 0.0065$, $B = 0.0065$;
 Front-wheel sidewall: $A = 0.0065$, $B = 0.0065$;
 Back-wheel tread: $A = 0.0065$, $B = 0.0065$;
 Back-wheel sidewall: $A = 0.0065$, $B = 0.0065$.

Spring and damper parameters:

Spring: $SPRING_SC = 0.0009$;
 Damper: $DAMP_SC = 0.0003$;

A.4 Structural properties KTM Duke 390

Table A.4: KTM Duke 390 LS-DYNA structural properties

Material	Name of part	T1 [mm]	RO (ρ) [kg/mm ³]	E [GPa]	PR (ν)	SIGY [GPa]	ETAN [GPa]
MAT27	FRONT-WHEEL-THREAD	8.	1.111E-6	–	0.495	–	–
MAT27	FRONT-WHEEL-SIDEWALL	9.	1.111E-6	–	0.495	–	–
MAT24	FRONT-WHEEL-SPOKES	4.	2.7E-6	71.	0.33	0.448	0.8
MAT24	FRONT-WHEEL-SPOKES-2	4.	2.7E-6	71.	0.33	0.448	0.8
MAT24	FRONT-WHEEL-HUB	3.	2.7E-6	71.	0.33	0.448	0.8
MAT24	FRONT-WHEEL-RIM	4.	2.7E-6	71.	0.33	0.8	0.
MAT24	FRONT-WHEEL-RIM-2	4.	2.7E-6	71.	0.33	0.8	0.
MAT67	FRONT-FORK-SPRING	–	1.	–	–	–	–
MATS05	FRONT-FORK-DAMPER	–	–	–	–	–	–
MAT24	FRONT-FORK-MIDDLE	18.	7.85E-6	210.	0.3	0.9	0.6
MAT24	FRONT-FORK-TOP	18.	2.89E-6	71.7	0.33	0.221	1.58
MAT24	FRONT-FORK-TRIPLE-CLAMPS-LOWER	14.	2.68E-6	70.	0.33	0.225	1.58
MAT24	FRONT-FORK-TRIPLE-CLAMPS-UPPER	14.	2.68E-6	70.	0.33	0.225	1.58
MAT24	FRONT-FORK-LOWER	18.	2.89E-6	71.7	0.33	0.221	1.58
MAT24	HANDLEBARS	2.25	7.85E-6	210.	0.3	0.	0.
MAT24	FRONT-HEADLIGHT	4.	1.4E-6	7.3	0.3	0.1	0.555
MAT24	FRONT-PLASTIC	4.	1.415E-6	1.	0.3	0.02	0.01
MAT24	FRONT-PLASTIC-OUTtoCONTACT	4.	1.415E-6	1.	0.3	0.02	0.01
MAT24	FRAME-FORK-CONNECTOR	12.	7.85E-6	213.	0.3	0.463	0.
MAT24	FRAME-BOTTOM-1ST	6.	7.85E-6	213.	0.3	0.463	0.
MAT24	FRAME-BOTTOM-2ND	4.	7.85E-6	213.	0.3	0.463	0.
MAT24	FRAME-BOTTOM-3RD	5.	7.85E-6	213.	0.3	0.463	0.
MAT24	FRAME-TOP-1ST	6.	7.85E-6	213.	0.3	0.463	0.
MAT24	FRAME-TOP-2ND	4.	7.85E-6	213.	0.3	0.463	0.
MAT24	FRAME-TOP-3RD	5.	7.85E-6	213.	0.3	0.463	0.
MAT24	FRAME-FRONT-TOP-BARS	1.75	7.85E-6	213.	0.3	0.463	0.
MAT20	FRAME-REAR	3.	7.851E-6	205.	0.3	–	–
MAT20	FRAME-SWINGARM	4.	2.89E-6	205.	0.3	–	–
MAT67	FRAME-REAR-SPRING	–	1.	–	–	–	–
MATS02	FRAME-REAR-DAMPER	–	–	–	–	–	–
MAT24	FUEL-TANK	0.81	7.85E-6	229.	0.3	0.396	0.
MAT34	SEAT-SKIN	1.	1.E-7	–	–	–	–
MAT57	SEAT-FOAM	–	1.18E-7	0.0049	–	–	–
MAT20	PEDALS	2.25	7.85E-6	210.	0.3	–	–
MAT20	ENGINE	4.	1.8E-6	210.	0.3	–	–
MAT24	BACK-WHEEL-HUB	3.	7.85E-6	210.	0.3	0.	0.
MAT24	BACK-WHEEL-RIM	4.	2.68E-6	70.	0.33	0.225	1.58
MAT24	BACK-WHEEL-RIM-2	4.	2.68E-6	70.	0.33	0.225	1.58
MAT24	BACK-WHEEL-SPOKES	4.	2.68E-6	70.	0.33	0.225	1.58
MAT24	BACK-WHEEL-SPOKES-2	4.	2.68E-6	70.	0.33	0.225	1.58
MAT27	BACK-WHEEL-SIDEWALL	0.6	1.111E-6	–	0.495	–	–
MAT27	BACK-WHEEL-TREAD	9.	1.111E-6	–	0.495	–	–

Material model abbreviations: MAT20 = Rigid; MAT24 = Piecewise Linear Plasticity; MAT27 = Mooney-Rivlin Rubber; MAT34 = Fabric; MAT57 = Low Density Foam; MAT67 = Nonlinear Elastic Discrete Beam; MATS02 = Viscous Damper; MATS05 = Nonlinear Viscous Damper.

Mooney-Rivlin parameters:

Front-wheel tread: $A = 0.009$, $B = 0.002$;

Front-wheel sidewall: $A = 0.018$, $B = 0.006$;

Back-wheel tread: $A = 0.009$, $B = 0.002$

Back-wheel sidewall: $A = 0.018$, $B = 0.006$.

Spring and damper parameters:

Spring: $SPRING_SC = 0.001$;

Damper: $DAMP_SC = 0.7$;



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