



CHALMERS
UNIVERSITY OF TECHNOLOGY



Truck front underrun protection parameter and identification

Bachelor's thesis in mechanical engineering

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BACHELOR'S THESIS IN MMSX25

Truck front underrun protection parameter and identification

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Göteborg, Sweden 2026

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Abstract

Frontal collisions between heavy goods vehicles and passenger cars remain a major traffic safety challenge due to differences in mass, stiffness, and structural geometry. Although FUP systems are intended to improve structural interaction and reduce underride risk, limited information is available regarding how modern heavy goods vehicles front structures geometrically interact with passenger car load paths.

This thesis investigates geometric parameters of modern European heavy goods vehicles front structures with focus on frontal crash compatibility with passenger cars. A Truck Front Component Measurement database was established from collected vehicle measurements and compared with passenger car structural data and historical truck data from the VC-Compat project. The analysis considered vertical, transverse, and longitudinal positioning of selected frontal structural components and simplified overlap configurations.

The result indicate that alignment between truck frontal structures and passenger car load-carrying components is limited in several investigated configurations, particularly when overlap is reduced.

The findings suggest that geometric compatibility should be considered together with strength and deformation requirements when evaluating FUP performance. However, the study is based on simplified geometric analysis and publicly available or manually collected data, and therefore does not provide a complete assessment of crash performance. Further work involving validated simulations or physical testing is required to assess the dynamic effects of the identified parameters.

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Preface

This bachelor thesis was conducted at Chalmers University of Technology during the spring of 2026 in collaboration with Volvo Trucks.

The authors would like to thank everyone who contributed to the project, including supervisors and employees at both Chalmers University of Technology and Volvo Trucks, for their guidance and support throughout the work. Special thanks are directed to everyone who contributed with valuable data and technical insights that supported the project.

AI-assisted tools were used for translation support and grammatical refinement during the writing process. All technical content and conclusions presented throughout the report were developed and reviewed by the authors.

Abbreviations

Abbreviation	Description
AEBS	Advanced Emergency Braking System
A2MAC1	Vehicle benchmarking and teardown company
COE	Cab Over Engine
ECAS	Electronically controlled air suspension
Euro NCAP	European New Car Assessment Program
FUP	Front Underrun Protection
FUPD	Front Underrun Protection Device
HDV	Heavy Duty Vehicle
HGV	Heavy Goods Vehicle
ICCT	The International Council on Clean Transportation
N1	Vehicle category N1 / Light trucks ≤ 3.5 tons
N2	Vehicle category N2 / Medium trucks 3.5 - 12 tons
N3	Vehicle category N3 / Heavy trucks > 12 tons
Overlap	Percentage of the passenger car front width involved in the collision
OLC	Occupant Load Criterion
PC	Passenger car
P1	Outer test points of FUP where forces are applied
P2	Test point of FUP with 350-600 mm offset from center
P3	Test center point of FUP where force is applied
TFCM	Truck Front Component Measurement
UNECE	United Nations Economic Commission for Europe
VRU	Vulnerable Road User (pedestrians, cyclists, etc.)

1 Introduction

HGVs are an important type of vehicles for different purposes such as construction, transportation and more within and outside of EU. Since HGVs are driving on the same roads as private passenger cars, the safety of these vehicles is of great importance. PCs and HGVs are different in many ways and may cause great dangers when met in frontal collisions. This chapter outlines the context, scope and limitations of this project that will look into the structural design and safety aspects of frontal collision between HGVs and PCs.

1.1 The Problem

Frontal collisions between PCs and HGVs are among the most severe vehicle-to-vehicle crashes. Previous research indicates that between 43% and 73% of frontal car-to-truck crashes result in fatalities (Krusper, 2014), primarily due to incompatibilities in mass, stiffness, and geometry. The large mass of HGVs results in high kinetic energy levels, often causing great deformation and high acceleration pulses in the passenger compartment of the PC.

Geometrical mismatch between the vehicle types further aggravates the problem. Differences in frontal structural layout may prevent proper engagement between the vehicles' energy-absorbing load paths, increasing the likelihood of structural intrusion and frontal underrun, where the PC slides beneath the truck front. To mitigate this problem, many HGVs are equipped with FUPDs intended to improve structural interaction and prevent underride by providing a lower load-bearing structure at the truck front.

However, several studies indicate that the effectiveness of current FUP designs may be limited. Crash tests with real world energy levels have shown that FUP systems fulfilling current regulations may still exhibit insufficient stiffness and deformation characteristics during frontal collisions (Thomson et al., 2023). In addition, limited focus is placed on the energy-absorbing capability of the FUPD. This indicates that further research is needed regarding how HGV front structures and FUP systems influence crash compatibility with PCs.

1.2 Relevance

The study of frontal compatibility between HGVs and PCs has become increasingly relevant due to the development of modern vehicle architectures and the growing demand

for road freight transportation throughout Europe. Despite continuous advancements in vehicle safety, collisions involving HGVs remain associated with disproportionately severe consequences for PC occupants and VRUs.

Current FUP regulations, including UNECE Regulation No. 93, were introduced in the 1990s and may therefore not fully reflect the front-end structures of modern HGVs operating on European roads today. Since then, both HGV and PC structures have undergone substantial changes regarding geometry, structural packaging, electrification, and crash safety requirements, potentially influencing structural interaction and compatibility performance during frontal collisions.

One of the most comprehensive European compatibility research initiatives, VC-Compat, was conducted more than two decades ago. Since then, relatively limited large-scale compatibility investigations focusing on modern HGV frontal structures and current underrun protection systems have been conducted. Understanding how current FUP regulations relate to modern vehicle architectures therefore remains highly relevant for improving compatibility in frontal HGV–PC collisions.

1.3 Gap

Much of the available compatibility research involving HGVs was conducted using vehicle data collected several years ago and may therefore not fully reflect the characteristics of modern HGVs. At the same time, openly available structural data for modern HGVs remains limited compared to PCs, where more extensive databases and crash investigations are available. The modular and highly variable nature of HGV designs further complicates comprehensive compatibility assessment. In addition, limited crash testing and compatibility-focused evaluation of HGVs may contribute to a reduced understanding of structural interaction and crash outcomes in frontal HGV–PC collisions.

1.4 Objective

This study aims to improve the understanding of HGV structures influencing compatibility and investigate the compatibility between HGVs and PCs in order to identify structural incompatibilities associated with current truck front designs during frontal collisions.

Particular focus is on frontal HGV components influencing crash interaction with PCs. Among these, the FUPD is of particular interest, as its geometry, stiffness, mounting, and placement influence structural interaction and the engagement of PC crumple zones and

energy-absorbing load paths.

1.5 Limitations

This study is limited to frontal collisions between PCs and HGVs, with focus on the structural compatibility of FUP systems installed on HGVs. The analysis primarily considers simplified geometrical aspects of the FUP system, including positioning, dimensions, and coverage range. Real-world crash behavior is influenced by several operational and vehicle-dependent factors, such as braking conditions, vehicle load state, ride height variation, suspension characteristics, and overall vehicle configuration. These aspects are therefore acknowledged and discussed where relevant throughout the study.

Since frontal HGV–PC collisions primarily affect the structural interaction with the PC occupant compartment, the study focuses on how HGV front structures may be modified to improve compatibility with PCs. Although several crash configurations may occur in real traffic environments, the present analysis is restricted to frontal impact scenarios considering overlap while excluding varying impact angles. Furthermore, the study focuses on structural components associated with the FUP system and does not consider other vehicle safety systems such as airbags, seatbelts, or advanced driver assistance systems.

Another limitation concerns the availability of detailed data and validated vehicle models. The analysis relies primarily on existing literature and video-documented crash tests, limiting the possibilities for comprehensive verification of the findings. Since the study is largely based on publicly available data, the range of investigated FUP designs and structural configurations is inherently limited.

The recommendations presented in this study are derived from measured vehicle geometries, available literature, and compatibility-related observations rather than from physical crash testing or numerical impact simulations. Consequently, the identified trends and proposed design considerations should be interpreted as indicative rather than definitive. While the results may support future compatibility assessments and FUP development, their applicability to real-world crash performance remains subject to further validation.

This project was conducted in collaboration between Chalmers University of Technology and Volvo Trucks. As some information obtained from Volvo Trucks is confidential, the final published report is limited to information that can be publicly shared.

2 Theoretical Framework / Background

Frontal collisions between HGVs and PCs remain a major traffic safety challenge due to structural misalignment, differences in mass and stiffness, energy absorption characteristics, and underrun risk.

This chapter presents previous research, regulatory frameworks, and relevant theoretical concepts in order to establish the theoretical foundation of the study.

2.1 Literature Review

This section presents an overview of previous studies regarding crash compatibility between HGVs and PCs in order to provide relevant background information and findings from earlier research.

2.1.1 FUP/FUPD

A key challenge in frontal HGV–PC collisions is preventing intrusion into the PC occupant compartment. Structural incompatibilities between the vehicle types increase the likelihood of deformation of the PC A-pillars, potentially reducing the effectiveness of occupant protection systems. Frontal underrun, where the PC slides beneath the HGV during a collision, may further aggravate crash outcomes. To reduce this risk, most HGVs manufactured since August 2003 are equipped with a FUPD (Krusper & Thomson, 2008), intended to prevent PC underride during frontal impacts. The protective effect of the FUPD is illustrated in Figure 1.

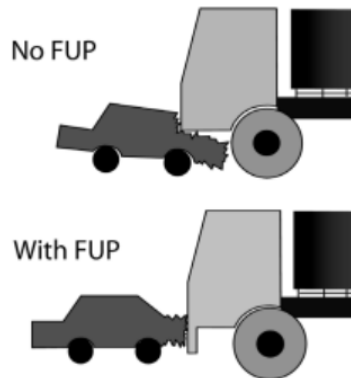


Figure 1: Outcome difference with/without FUPD installed. (adapted from Krusper, 2014)

To improve road safety within the EU, the UNECE World Forum for Harmonization of Vehicle Regulations has established regulations governing the structural characteristics, geometrical requirements, and performance criteria of FUPDs. The primary objective of the FUPD is to improve structural alignment with PCs in order to achieve more effective engagement of occupant protection systems and energy-absorbing load paths (Främby & Lantz, 2010).

2.1.2 Regulations

In order to understand the current design of FUP systems in HGVs, it is essential to consider the regulatory framework defining their design constraints and performance requirements. While some regulations directly influence the FUP system, others affect its design indirectly.

UNECE R93

The most important regulation governing the design and performance of FUPDs is UNECE Regulation No. 93. The regulation establishes minimum safety requirements intended to improve compatibility between HGVs and smaller vehicles during frontal collisions. Certain vehicle categories, such as military vehicles, are exempt from these requirements, but due to their limited presence in general traffic they are not considered relevant for this study.

UNECE Regulation No. 93 defines both geometrical and performance requirements for FUP systems on N2 and N3 vehicles. Key requirements include limitations regarding FUP beam dimensions and positioning, such as minimum beam heights of 100 mm for N2 vehicles and 120 mm for N3 vehicles, together with a maximum ground clearance of 400 mm. The regulation also limits the rearward positioning of the FUP relative to the vehicle front to 400 mm during testing, see Figure 2. During quasi-static testing, the regulation further limits vertical displacement to 50 mm and specifies load application points not exceeding 445 mm above ground level.

According to UNECE Regulation No. 93, prescribed loads shall be applied as rapidly as possible and maintained for at least 0.2 seconds. In the quasi-static test, forces are applied at points P1, P2, and P3 illustrated in Figure 3, with upper load limits of 80 kN at P1/P3 and 160 kN at P2. Despite the rapid loading procedure, the test is classified as quasi-static since it evaluates structural strength under controlled conditions rather than dynamic impact loading.

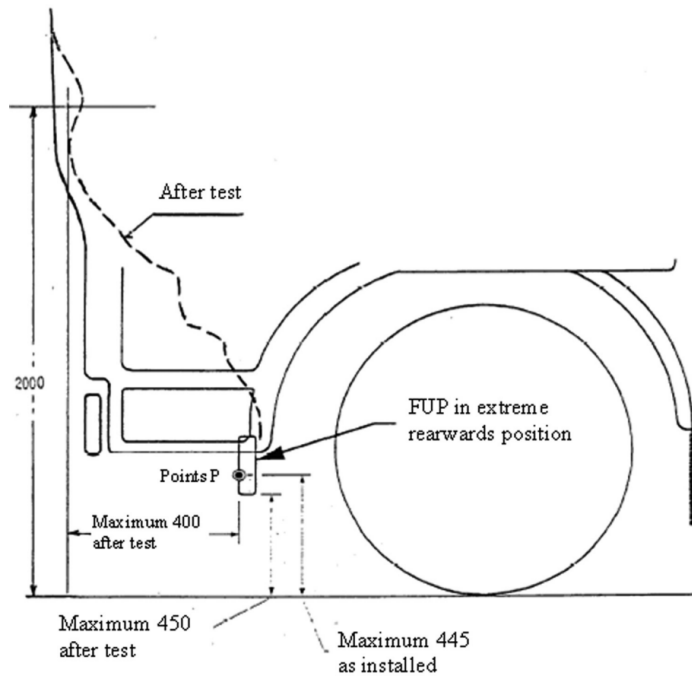


Figure 2: Explaining the FUP longitudinal positioning before and during testing, reproduced from UNECE Regulation No. 93.

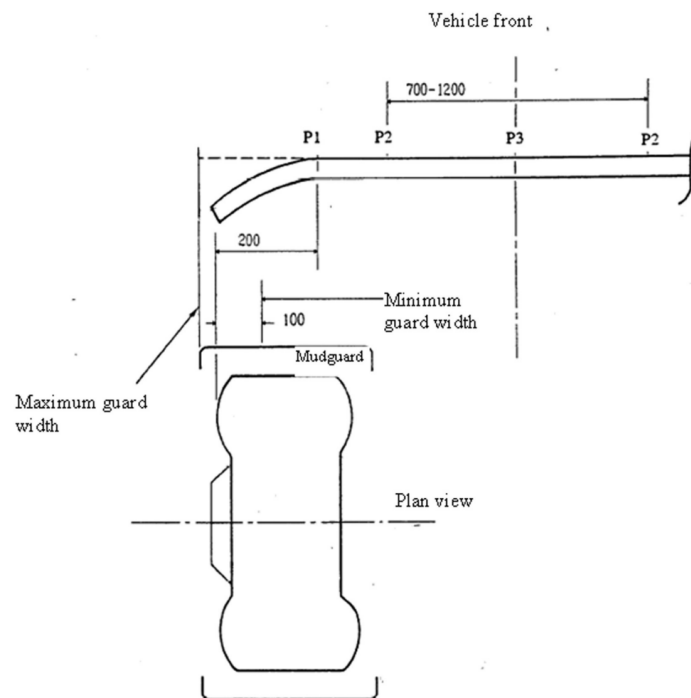


Figure 3: Illustration of load application points P1, P2 and P3, reproduced from UNECE Regulation No. 93.

Directive 96/53/EC

Directive 96/53/EC, which defines the maximum permissible vehicle dimensions and weights within the EU, has historically influenced HGV front-end design by promoting

cab-over-engine configurations, see Figure 4.



Figure 4: Illustration of long nose design (left) and COE design generated by AI for visualization purposes.

Since the total allowable vehicle length includes both the cab and cargo space, manufacturers have minimized cab length in order to maximize cargo capacity by positioning the cab directly above the engine. However, this compact design creates challenges regarding crashworthiness and energy absorption due to limited frontal deformation space and the close placement of rigid structural components near the truck front.

The compact front structure increases the importance of the FUPD, as it often becomes the first load-bearing structure engaged during frontal collisions and contributes to preventing underride. At the same time, the limited available deformation distance at the truck front may reduce the potential for progressive energy absorption, increasing the demand on the PC occupant protection structures.

Directive (EU) 2015/719

Historically, strict overall vehicle length limitations within the EU promoted highly compact COE designs. While this configuration maximized cargo capacity and operational efficiency, it also resulted in compact front structures with limited deformation space before interaction with rigid structural components during frontal collisions.

Directive (EU) 2015/719 was introduced as an amendment to Directive 96/53/EC, allowing increased overall vehicle length if improved safety or aerodynamics could be achieved, provided that maneuverability and safety requirements are fulfilled.

Extended front designs may improve frontal impact characteristics by increasing the available deformation space. Through more progressive structural interaction and energy absorption, improved compatibility and occupant protection for PCs may be achieved.

2.1.3 Euro NCAP

Euro NCAP is an independent organization that evaluates vehicle safety and publishes comparative ratings. Since 1997, it has contributed to increased safety standards by encouraging manufacturers to implement safety technologies beyond regulatory requirements. In November 2024, Euro NCAP released its first safety ratings for HGVs (Euro NCAP, 2024), initially focusing on Safe Driving, Crash Avoidance, and Post-Crash Safety. A fourth category, Crash Protection, is planned for 2031 and will include occupant protection and crash compatibility (Euro NCAP, n.d.).

In multi-vehicle collisions involving HGVs, fatalities among occupants of the opposing vehicle outnumber HGV occupant fatalities by five to one (Euro NCAP, n.d.), primarily due to differences in mass and geometry between HGVs and PCs. This makes the development of compatibility-focused assessment methods particularly relevant. In addition, Euro NCAP ratings strongly influence both market perception and vehicle purchasing decisions, encouraging manufacturers to improve safety performance beyond minimum regulatory requirements. By incorporating crash compatibility into its assessment framework, Euro NCAP may influence future HGV design standards and compatibility-focused safety development in HGV-PC collisions.

2.1.4 VC-Compat

An earlier European research consortium investigating frontal crash compatibility between PCs and HGVs was VC-Compat. Funded by the European Commission, the project focused on improving structural interaction between HGVs and PCs based on the recognition that vehicle incompatibility represents a major traffic safety challenge.

The project emphasized that the geometrical positioning and stiffness characteristics of frontal vehicle structures strongly influence compatibility performance by affecting structural interaction and impact force distribution during collisions. Positional measurements of frontal structures in both PCs and HGVs were therefore collected in order to evaluate structural compatibility.

VC-Compat also investigated the advantages and limitations of energy-absorbing FUPDs and how different FUP concepts influence intrusion into the PC occupant compartment.

Since this thesis investigates structural compatibility between modern HGVs and PCs, the VC-Compat project provides important background regarding frontal crash interaction and compatibility-related parameters.

2.1.5 EU cars database_2026

The purpose of the *Car Structure Parameter Identification for CAE Barrier Model Development_Master Thesis Report_2026* project was to investigate how the frontal structural characteristics of modern PCs influence crash compatibility in frontal collisions with HGVs, and to translate these characteristics into simplified CAE models for improved safety assessment and future FUPD-related analysis.

Since the project includes measurement data related to frontal structures of the modern European PC fleet, the resulting database, “EU Cars Database_2026”, is considered suitable for establishing a comparative representation between the current HGV and PC fleets.

2.2 Limitations to Existing Literature

This study relies on the availability of detailed structural information for both PCs and HGVs. For PCs, sources such as A2MAC1 provide access to comprehensive structural data that can be used to obtain detailed information about vehicle components. In contrast, equivalent structural information for HGVs is considerably more limited and, in some cases, outdated.

While manufacturers may collect detailed data on their own vehicle fleets, such information is often kept confidential, making it difficult to access data beyond general publicly available sources or non-detailed descriptions. To gain a broad database worthy of representing an EU fleet, the result is limited to information gained by manufacturers willing to collaborate. Consequently, obtaining detailed information about HGV structures often requires independent data collection.

When analyzing the effect of different crash configurations such as impact angle, ride height, brake-squatting, velocity, load and overlap etc., there are limitations to how much data that can be found, making it difficult to give a realistic evaluation of all configuration differences.

3 Methodology

In order to fulfill the objective of this study, relevant data must be collected, analyzed, and systematically evaluated to enable reliable conclusions to be drawn. This chapter presents the methodologies, data collection procedures, and analytical approaches used throughout the study, as well as how the collected data was processed and assessed to ensure its relevance and applicability to the investigation of HGV–PC crash compatibility.

3.1 Analyzing Regulations

Although several updates have been made since its introduction, the core design and testing framework of UNECE Regulation No. 93 remains largely unchanged. An amendment introduced in 2017 allows the FUP geometry to follow the truck front structure, provided that deformation from the load application plane does not exceed 400 mm from the tested surface.

The prescribed test setup represents controlled loading conditions with perpendicular and uniform force application across the FUP structure, while real-world collisions often involve concentrated loads, varying overlap conditions, asymmetric interaction, multi-directional force components, and dynamic impact behavior not represented within the quasi-static framework.

Furthermore, each load case is evaluated independently without accounting for accumulated deformation between load applications. Since the FUP structure may be restored or replaced between tests, cumulative damage effects and mounting degradation are not represented within the assessment procedure

3.2 Vehicle structural layout

An important initial step in the investigation of crash compatibility and the influence of different parameters on frontal collision outcomes is the identification and definition of the structural components considered most relevant to the study. Since the same component may be referred to by different terms depending on the source, clearly defining the terminology is important. Both PCs and HGVs frontal structures are relevant, as the interaction between these structures largely determines compatibility performance during a collision.

Since the PCs data representing the present-day fleet was obtained from EU Cars Database_2026,

the terminology used throughout this report follows the definitions adopted within that database. Figure 5 illustrates the analyzed PC structural components.

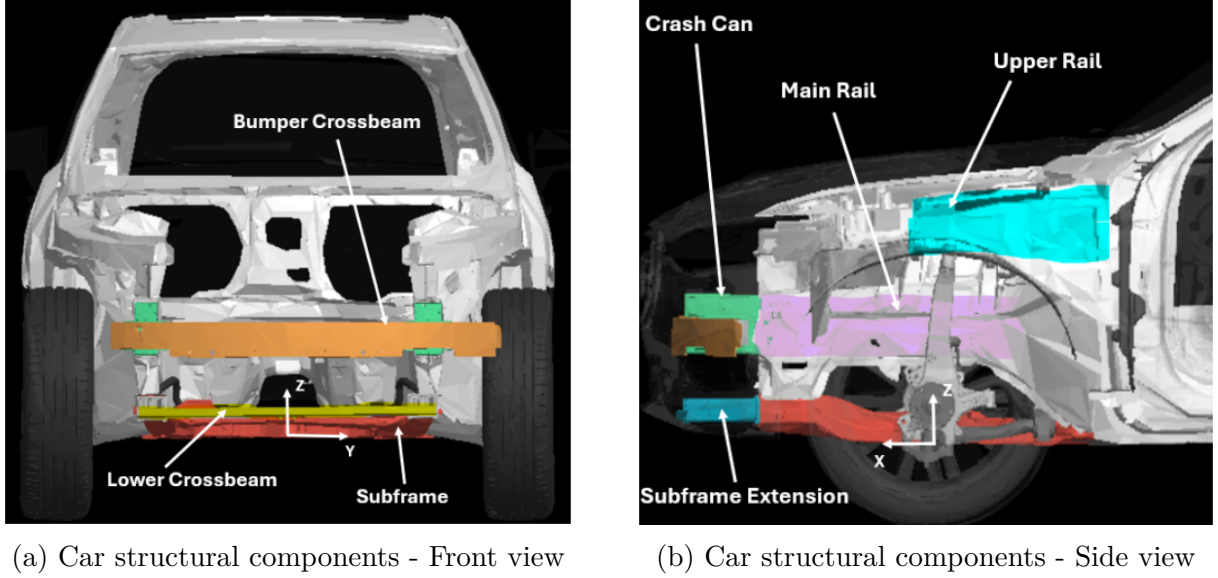


Figure 5: Frontal components and names of PCs

The terminology used in the TFCM database differs slightly from the terminology adopted within the VC-Compat study. The component definitions used for the HGV frontal structures in this report are presented in Figure 6 and further defined in table 1.

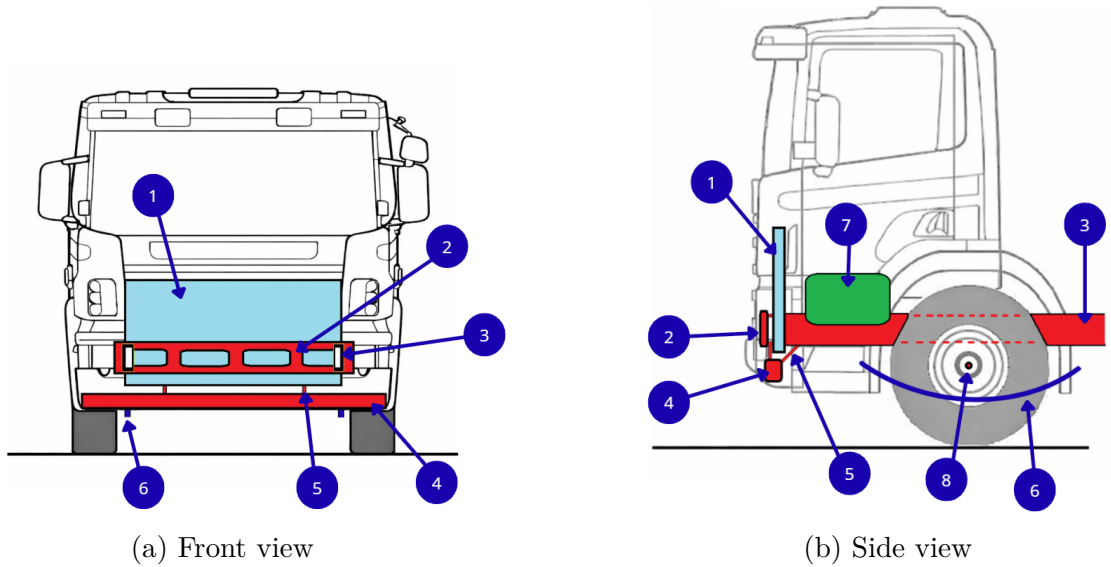


Figure 6: Frontal components and names of HGVs from front view (a) and side view (b).

Table 1: Truck front components

Component	Name
1	Radiator
2	Front closure / bumper beam
3	Frame rails
4	FUPD
5	FUP brackets
6	Leaf springs
7	Engine
8	Front axle

In total, seven structural components were selected and measured to establish the Truck Front Components Measurements database. Component selection was based on previous research, structural positioning, and expected load-path interaction. The front axle was used as the primary longitudinal reference point.

3.3 Crash compatibility and influencing factors

Frontal collisions represent complex crash scenarios involving several factors influencing structural interaction and compatibility performance. When collisions involve different vehicle categories, the complexity further increases due to differences in mass, geometry, stiffness, and structural layout. To improve the understanding of frontal HGV-PC compatibility, several influencing parameters were investigated throughout the study.

3.3.1 Brake Squatting

Modern trucks are typically equipped with AEBS, which automatically initiates braking when a potential collision is detected. Consequently, frontal HGV-PC collisions may occur while the truck is undergoing heavy braking due to either AEBS intervention or driver reaction.

During braking, longitudinal load transfer increases the vertical load on the front axle, compressing the front suspension and causing vehicle squat. The magnitude of the squat varies depending on vehicle configuration and several operating conditions.

The amount of longitudinal load transfer can be estimated using

$$\Delta F = \frac{mah}{L} \quad (1)$$

where ΔF is the load transfer, m is the vehicle mass, a is the braking deceleration, h is the centre-of-gravity height, and L is the wheelbase.

The resulting ride-height change may depend on suspension type. For leaf spring suspension, ride height generally decreases as vehicle load increases. Air suspension systems may partly compensate for ride-height variations during operation through electronically controlled suspension systems (Volvo Trucks, 2023). The same documentation states that unloaded and loaded ride height may remain similar due to automatic height adjustment.

However, the ability of air suspension systems to counteract squat during braking may be limited by the response time required to adjust air pressure. In some ECAS implementations, active height regulation may also be suspended during braking. According to the ECAS maintenance manual, the ECU freezes ECAS control activities while the brakes are applied (ZF CV Systems North America, 2022). Due to these interacting factors, it is difficult to define a representative ride-height change applicable to all vehicles.

According to Volvo internal information, HGVs may squat approximately 70–80 mm during braking. The compatibility implications of this behavior are further discussed in the discussion section.

3.3.2 Glance-Off

A *glance-off*, where the PC is deflected away from the truck front instead of fully engaging structurally, may be beneficial in frontal HGV–PC collisions. In such cases, the contact surfaces slide against each other rather than fully engaging, allowing part of the vehicle velocity and kinetic energy to remain as post-impact motion instead of being transformed into structural deformation energy (Krusper, 2014). A successful glance-off may therefore reduce A-pillar deformation and OLC values.

Achieving a glance-off requires sufficient FUP structural integrity, favorable friction conditions, compatible alignment between the FUP and PC load paths, and an angled or curved geometry capable of promoting lateral deflection and geometric energy redirection.

As shown in Figure 7, an angled back plate, meaning the rear supporting plate behind the energy-absorbing structure in the studied model, reduced A-pillar deformation and firewall intrusion at lower overlaps, indicating that angled geometries may be beneficial when the crash configuration allows a glance-off response.

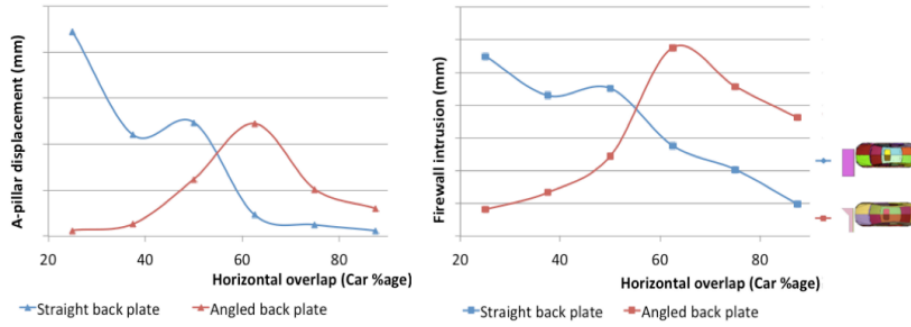


Figure 7: Effect of an angled back plate on PC deformation at different overlap conditions, adapted from Lefer and Reboloso (2012).

At higher overlaps, however, glance-off behavior becomes less likely as a larger portion of the PC front engages the truck structure. In such cases, a straight FUP design may instead provide more even force distribution across the engaged structures. Although the study shown in Figure 7 investigated an angled back plate rather than an angled FUP, similar tendencies have been indicated by internal simulation results.

3.3.3 Angled and Curved FUP Designs

As described in the previous section, a *glance-off* interaction may reduce the concentrated loading and poor structural alignment associated with small-overlap collisions by deflecting the PC away from the truck front. One possible approach to promote such behavior is the implementation of angled or curved FUP geometries, which generate a lateral force component encouraging the PC to slide along the truck front, as seen in figure 8

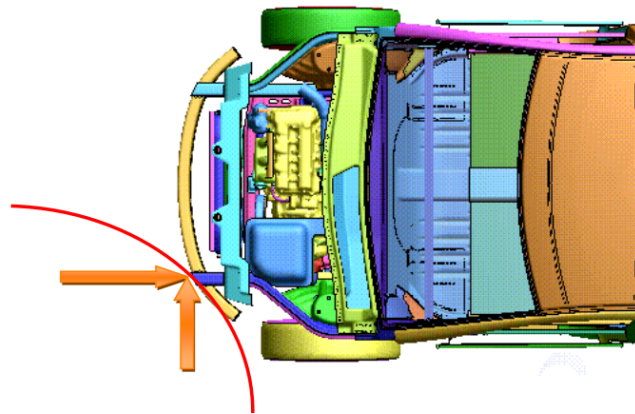


Figure 8: Illustration of a glance-off interaction between a PC and an angled truck front structure, reproduced from Knight (2016).

Curved FUP designs represent one approach for introducing this interaction and may be combined with long-nose vehicle concepts. However, highly curved geometries may create

compatibility challenges in higher-overlap collisions. If initial contact occurs in a narrow curved region, loads may be transferred into rigid PC components before the primary load paths have engaged, potentially increasing occupant compartment intrusion. High load concentrations may also increase the likelihood of FUP cross-member failure (Knight, 2016).

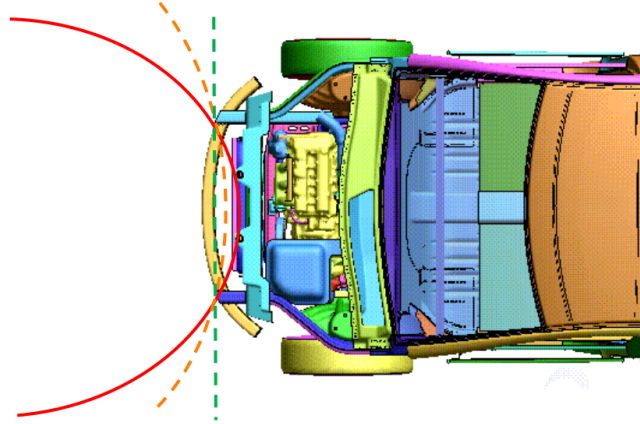


Figure 9: Example of a curved front-end concept for a truck front, adapted from Knight (2016).

For these reasons, a highly curved FUP across the entire width of the vehicle may not be optimal. Instead, a design combining a straight central cross-member with angled outer sections may provide a more balanced solution, by maintaining effective load distribution in moderate and high overlap collisions while still providing a geometric structure that allows glance-off interactions in small overlap scenarios.

Previous studies have investigated the geometric angles required to achieve vehicle deflection. Open literature on this topic remains limited; however, Lambert and Rechnitzer (2002) theoretically investigated the relationship between overlap, closing speed, and the frontal angle required to redirect a PC. Their results show that the required angle increases significantly with overlap and is also influenced by closing speed.

Figures 10 and 11 illustrate the theoretically required frontal angle for vehicle deflection for friction coefficients of $\mu = 0.2$ and $\mu = 0.4$, respectively, with overlaps ranging from 10% to 40%. The results indicate that significantly larger angles are required as the overlap increases. For example, at approximately 80 km/h closing speed, the required angle increases from roughly 22° for a 10% overlap to approximately 47° for a 40% overlap when the friction coefficient is $\mu = 0.2$ (Lambert & Rechnitzer, 2002).

The results also demonstrate that friction between the interacting surfaces has an impact on the required angle, but the main driver is the overlap level.

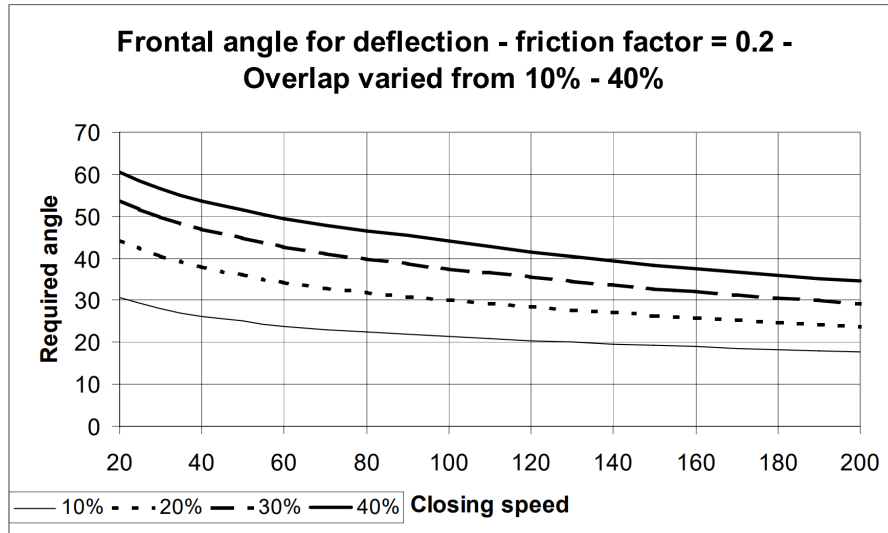


Figure 10: Required frontal angle for vehicle deflection as a function of closing speed and overlap for a friction coefficient of $\mu = 0.2$, adapted from Lambert and Rechnitzer (2002).

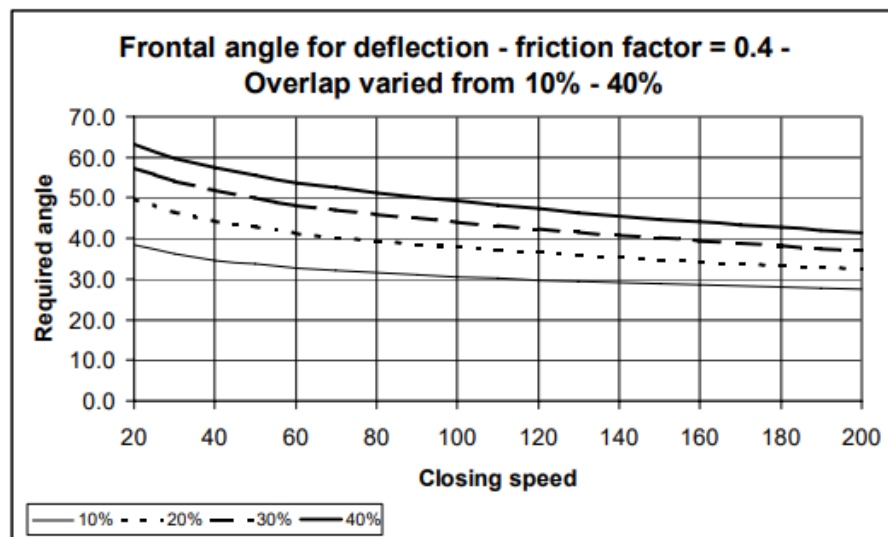


Figure 11: Required frontal angle for vehicle deflection as a function of closing speed and overlap for a friction coefficient of $\mu = 0.4$, adapted from Lambert and Rechnitzer (2002).

3.3.4 Long-Nose / Cab-Over-Engine

One challenge associated with conventional COE truck designs is the limited deformation space available ahead of rigid front-end components. As a result, unfavorable structural interaction may occur during frontal HGV-PC collisions. One approach to improving compatibility is the introduction of an additional deformable front structure capable of absorbing energy before the impact reaches the stiffer truck components.

Following the introduction of Directive (EU) 2015/719, manufacturers were allowed greater

flexibility in front-end design without reducing cargo capacity, creating opportunities for extended front structures and long-nose concepts. By increasing the available crush distance ahead of rigid components, additional frontal length may improve energy absorption and reduce the amount of energy transferred to the PC

Long-nose designs may also improve aerodynamic performance, although Robinson et al. (2010) showed that increasing front length alone does not necessarily reduce drag when a trailer is attached. Aerodynamic benefits therefore depend on the design of the entire vehicle combination.

However, increased front length introduces practical constraints related to manoeuvrability, approach angle, and ramp angle. As illustrated in Figure 12, increasing nose length while maintaining a constant approach angle requires increased ground clearance at the front-most edge of the vehicle.



Figure 12: Illustration of approach angle and ramp angle, reproduced from Robinson et al. (2010).

Robinson et al. (2010) demonstrated this relationship, showing that a longer nose requires substantially greater ground clearance than a conventional flat-front, see table 2. This may conflict with the objective of maintaining low front-end structures for compatibility with PC load paths.

Table 2: Required ground clearance for different nose lengths at a 10° approach angle, adopted from Robinson et al. (2010).

Nose length [mm]	Ground clearance [mm]
0	220
200	255
400	290
600	325
800	360
1000	395
2250	615

* Ground clearance measured at the front-most edge of the nose cone.

Consequently, extended front structures represent a trade-off between increased deformation space and practical design constraints. A moderate extension may therefore provide the most realistic balance between compatibility improvements and vehicle functionality.

3.3.5 Load

Load condition may influence truck ride height and thereby affect vertical compatibility with PCs. The magnitude of this effect depends on vehicle configuration, axle-load distribution, and suspension design. Many trucks use air suspension systems that compensate for load-induced ride-height variations by adjusting the air pressure in the suspension bellows to maintain a constant chassis height (Volvo Trucks, 2023).

The effect of loading also differs between truck categories. In rigid trucks, the payload acts directly on the chassis, whereas in tractor–semitrailer combinations a large share of the load is carried by the trailer axles. The load transferred to the tractor depends on vehicle geometry and fifth-wheel position (Volvo Trucks North America, 2017). Since the fifth-wheel position can in some cases be adjusted, the resulting axle loading and ride height may also vary (SAF-Holland, n.d.).

As an indication of the influence of loading, Volvo VNR Electric body-builder data show approximately 10 mm difference between loaded and unloaded ride height for rear suspension configurations without ECS compensation, while vehicles equipped with electronically controlled suspension maintain the same ride height under both conditions (Volvo Trucks North America, 2024).

Ride heights

Rear suspension height (A)	Frame height (B)	Ride height (C)	
		Loaded	Unloaded
RSH-STD	FRAMEHIG (FRAME300)	193 ±5 mm (7.6 ±0.2 inches)	203 ±5 mm (8 ±0.2 inches)
	FRAMELOW (FRAME266)	210 ±5 mm (8.2 ±0.2 inches)	220 ±5 mm (8.6 ±0.2 inches)
RSH-HIG	FRAMEHIG (FRAME300)	213 ±5 mm (8.3 ±0.2 inches)	223 ±5 mm (8.7 ±0.2 inches)
	FRAMELOW (FRAME266)	230 ±5 mm (9 ±0.2 inches)	240 ±5 mm (9.4 ±0.2 inches)

Note: Vehicles that have a 6x2 pusher axle configuration are equipped with ECS (Electronically Controlled Suspension). So the unloaded ride height is the same as the loaded ride height (use the loaded values for a vehicle that is unloaded) because the ECS automatically adjusts the rear ride height.

Figure 13: Loaded and unloaded rear ride height comparison for Volvo VNR Electric rear suspension configurations, used as an indicative example based on publicly available data from Volvo Trucks North America (2024).

Volvo manufacturer guidance for North American trucks also states, as a general chassis guideline, that if an unloaded chassis is measured, the loaded frame rail height may be assumed to be approximately two inches lower than the bare chassis, corresponding to approximately 51 mm (Volvo Trucks North America, 2024). This value should also be interpreted as an indicative example rather than a direct correction factor for European vehicles.

Measurements obtained from individual truck data indicated that load-dependent ride-height variations may be substantially larger. For the MAN TGX 18.480 (2025), the documented ride-height difference between unladen and maximum-load conditions was 66 mm, while the corresponding value for the MAN eTGX (2026) was 82 mm. Both vehicles were equipped with leaf spring suspension. Although these examples are not representative of all N3 vehicles, they demonstrate that load-induced ride-height variation can influence the vertical alignment between the FUP and PC load paths. Due to differences in suspension systems, vehicle geometry, and load distribution, a common ride-height correction value applicable to all trucks is difficult to justify.

3.3.6 FUP Stiffness and Energy Absorption

Preventing underride alone does not guarantee favorable crash performance. Even when the FUP activates the PC load paths, the high mass and stiffness of conventional COE trucks may still result in high structural loading. Due to the limited deformation space ahead of rigid truck components, a large share of the crash energy may be absorbed by the PC, resulting in high OLC values and occupant compartment intrusion.

Previous studies have shown that most production FUPs approved according to UNECE

R93 deform less than 50 mm, indicating limited energy absorption on the truck side (Anderson, 2003, as cited in Knight, 2016). Knight (2016) argues that this is partly because manufacturers often use common FUP designs across multiple vehicle configurations, while UNECE R93 contains no explicit energy absorption requirements.

However, increased energy absorption alone does not necessarily improve compatibility. If a deformable FUP is mounted close to highly rigid truck components, excessive deformation may allow these structures to directly engage the PC, potentially compromising structural interaction. Furthermore, if the deformation zone extends rearwards beyond the bonnet length of the PC, its vertical position becomes critical. Deformation concentrated in the lower part of the truck front may allow the PC to continue penetrating into the front extension, causing higher structures to interact with the windscreen or A-pillar region. Consequently, energy-absorbing front structures must be combined with sufficient deformation space and appropriate vertical alignment. Krusper (2008) states that approximately 360 mm of truck-side deformation space is required to avoid occupant compartment intrusion

Knight (2016) suggests that the FUP should not begin to deform immediately at first contact. Instead, it should initially behave in a rigid manner, providing a stable interface so that the PC crash structure is activated and starts to deform in a controlled way. Only after this initial interaction should the FUP enter an energy-absorbing phase at an appropriate trigger force. Knight suggests a trigger force of approximately 150–200 kN, with a stable force during the absorption phase of around 300 kN or lower in order to soften the impact. Krusper and Thomson (2014) state that a trigger force of 200 kN is optimal. The trigger force is important to set correctly, since it must both provide a stable initial interaction and allow effective energy absorption during the later phase of the crash.

Krusper and Thomson (2012) further identified FUP cross-member height as an important parameter. A taller cross-member may improve structural interaction, increase energy absorption, and accommodate greater vertical variation between HGVs and PCs. However, these benefits may be reduced when deformation is restricted by surrounding truck-front structures.

3.3.7 Problem of Varying Overlaps

The load distribution within the FUP system varies significantly depending on the impact overlap. This is relevant since frontal car-to-truck crashes are not concentrated to one specific overlap condition. As shown in Figure 14, Knight (2016) presents data showing

that the overlap distribution differs between fatal collisions in Great Britain and a sample of fatal and serious injury collisions in Germany. The largest difference is found in the 0–25% overlap range, where the German KSI sample contains a considerably higher share of crashes than the British fatal sample. However, when a straight average is taken between the two samples, the overlap distribution becomes relatively even across the investigated ranges. This indicates that the selection of only one individual overlap value would not be sufficient to represent the full range of relevant crash cases. Therefore, the FUP should be considered across multiple overlap levels.

Overlap (car)	Proportion of all crashes		
	GB Fatal	DE KSI	Mean
0%-25%	6%	40%	23%
25% - 50%	33%	22%	27%
50%-75%	31%	14%	22%
75%-100%	30%	25%	28%

Figure 14: Distribution of crash overlaps in car-to-truck head-on crashes, adapted from Knight (2016), based on data presented by Gwehenberger, Bende, Knight, and Klootwijk (2004).

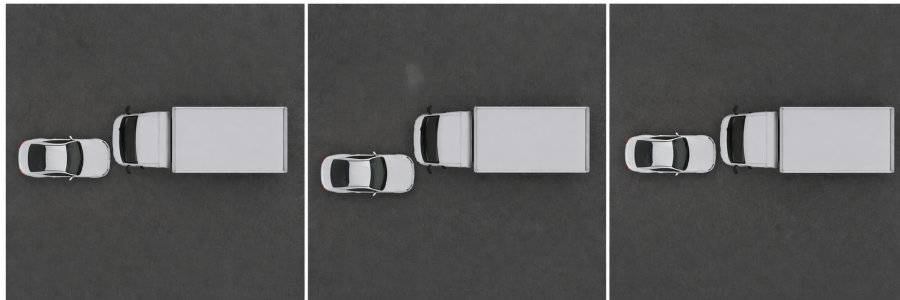


Figure 15: Schematic illustration of different PC overlap conditions in a frontal car-to-HGV impact. Generated with AI

In full overlap collisions, the load can be distributed across both vertical supports of the FUP and the car's crash boxes. This allows the crash forces to be transmitted into the designed load paths and can enable controlled energy absorption, provided that the load paths line up with the PC structures vertically. This is further evaluated in the compatibility comparison section.

At lower overlaps, load transfer becomes increasingly concentrated. Moderate overlaps may engage only one support, while very small overlaps may load the FUP cross-member

outside the supported region (Knight, 2016). Under such conditions, force transfer into the intended energy-absorbing structures becomes less effective and local FUP failure may occur. Similar load-path issues were observed in the physical HGV-PC crash test presented by Thomson et al. (2023).

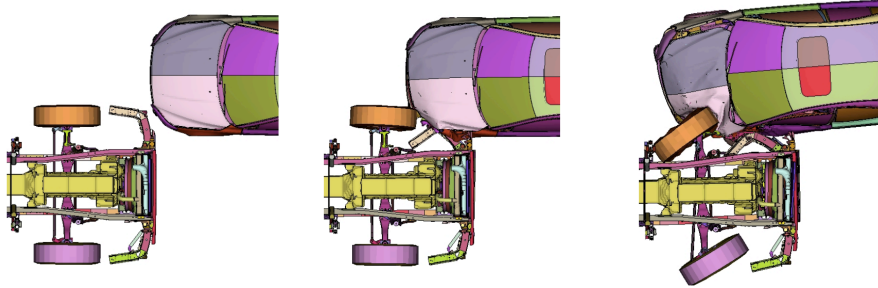


Figure 16: Example of overlap-dependent FUPD loading and structural interaction, adapted from Lefer and Reboloso (2012).

These overlap-dependent loading conditions illustrate the challenge of FUP design and evaluation. Different overlap scenarios produce fundamentally different load paths, which may be further complicated by variations in impact angle.

3.3.8 Frame rail intrusion

Krusper and Thomson (2008) observed that HGV frame rails overlap vertically with the engine region of PCs during frontal collisions. If the frontal load paths of the vehicles are misaligned, interaction may occur between the PC engine and the HGV frame rails. Since both components are highly rigid and located close to the passenger compartment, such interaction may contribute to occupant compartment intrusion.

To mitigate this issue, Krusper and Thomson (2008) proposed raising the foremost section of the frame rails to reduce the likelihood of interaction between the HGV frame rails and the PC engine, as illustrated in Figure 17.



Figure 17: Frame rails (grey) modification to prevent interaction with car engine (green), adapted from Krusper & Thomson (2008).

Whether a similar overlap exists in the current European vehicle fleet remains uncertain. The EU Cars Database_2026, used to represent the present PC fleet, does not contain engine-position measurements. Therefore, the VC-Compat database was used as a reference source, as it includes geometric data related to engine positioning and enables assessment of potential overlap between PC engines and HGV frame rails.

3.4 Data Sources and Collection

Gaining an understanding of the current compatibility of today’s HGVs and PCs, data from both trucks and cars are needed. Since different data bases and data collecting methods would be required to achieve a basis worthy of representing the fleet, the different data sources are explained in this section.

3.4.1 VC-Compat

The VC-Compat consortium provides a valuable reference for analyzing the development of frontal HGV–PC compatibility over time. The project established databases containing measurements from approximately 500 HGVs and 55 PCs, where relevant frontal structures were measured in the vertical, longitudinal, and transverse directions, as illustrated in Figure 18.

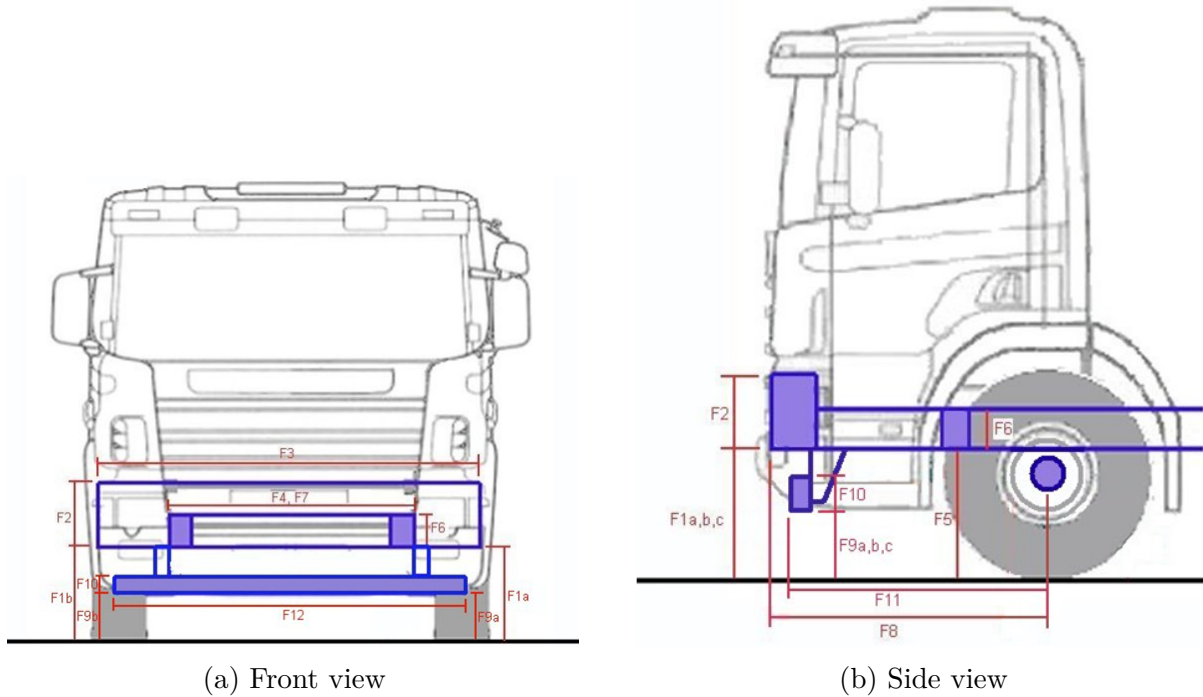


Figure 18: Structural placement and measurements in the study made by VC-Compat

Table 3 further explains the VC-Compat measurement definitions used in the comparison. The comparative diagrams between the VC-Compat and TFCM can be found in appendix B.

Table 3: Definitions of measurement parameters

Parameter	Definition
F1-u	Height from ground to lower surface of front bumper/front closure/bumper beam, unladen.
F1-l	Height from ground to lower surface of front bumper/front closure/bumper beam, laden.
F2	Vertical height/cross-section of bumper/front closure/bumper beam.
F3	Total width of front bumper/front closure/bumper beam.
F4	Width between outer faces of longitudinals/frame rails at the bumper connection.
F5-u	Height from ground to lowest surface of longitudinals/frame rails, unladen.
F5-l	Height from ground to lowest surface of longitudinals/frame rails, laden.
F6	Vertical height/cross-section of longitudinals/frame rails.
F7	Width between outer faces of longitudinals/frame rails.
F8	Longitudinal distance from front surface of bumper/front closure/bumper beam to centre of front axle.
F9-u	Height from ground to lower surface of FUP, unladen.
F9-l	Height from ground to lower surface of FUP, laden.
F10	Vertical height/cross-section of FUP front face.
F11	Longitudinal distance from front surface of FUP to centre of front axle.
F12	Total width of FUP front face.

Although the dataset covered a broad range of model years, 2003 was identified as the only common production year. The database can therefore be considered representative of the European vehicle fleet from approximately two decades ago and serves as a baseline for comparison with the present-day fleet.

Much of the collected structural data corresponds closely to the parameters investigated in the present study, including the FUPD, bumper beam, and frame rails. One uncertainty concerns how the consortium defines the bumper beam, which may influence the interpretation of comparative results.

Access to the raw data enables a detailed comparison of how the vehicle fleet and its structural compatibility characteristics have evolved over time. The measurements are also expected to be of high precision, as the data collection was conducted using a 3D measuring arm equipped with a rail, see figure 19(Martin, 2005, p. 4).



Figure 19: Measurement tool used in the VC-Compat data collection, Adapted from Martin (2005).

3.4.2 Internal Data and Assumptions

Compared to PCs, where structural and geometric data are relatively accessible through sources such as A2MAC1, publicly available information for HGVs is limited. Several data acquisition methods were therefore investigated, including CAD models, technical drawings, manual measurements, OEM-provided information, and publicly available data.

Access to CAD models and technical drawings proved challenging, as manufacturers are generally reluctant to share information that may provide competitive advantages. Consequently, the study initially relied on manual measurements of available vehicles using folding rulers and measuring tapes. To improve consistency and accessibility, a reference-point methodology was adopted, where dimensions were measured relative to a common

reference location.

One challenge during data collection was that many vehicles were equipped with protective skid plates beneath the front structure, restricting access to several components. This was observed across multiple manufacturers, including Scania and DAF.

Data collection was performed using a predefined measurement template inspired by the VC-Compat methodology. The template specified the components and dimensions to be recorded, ensuring consistent documentation and facilitating comparison between vehicles.

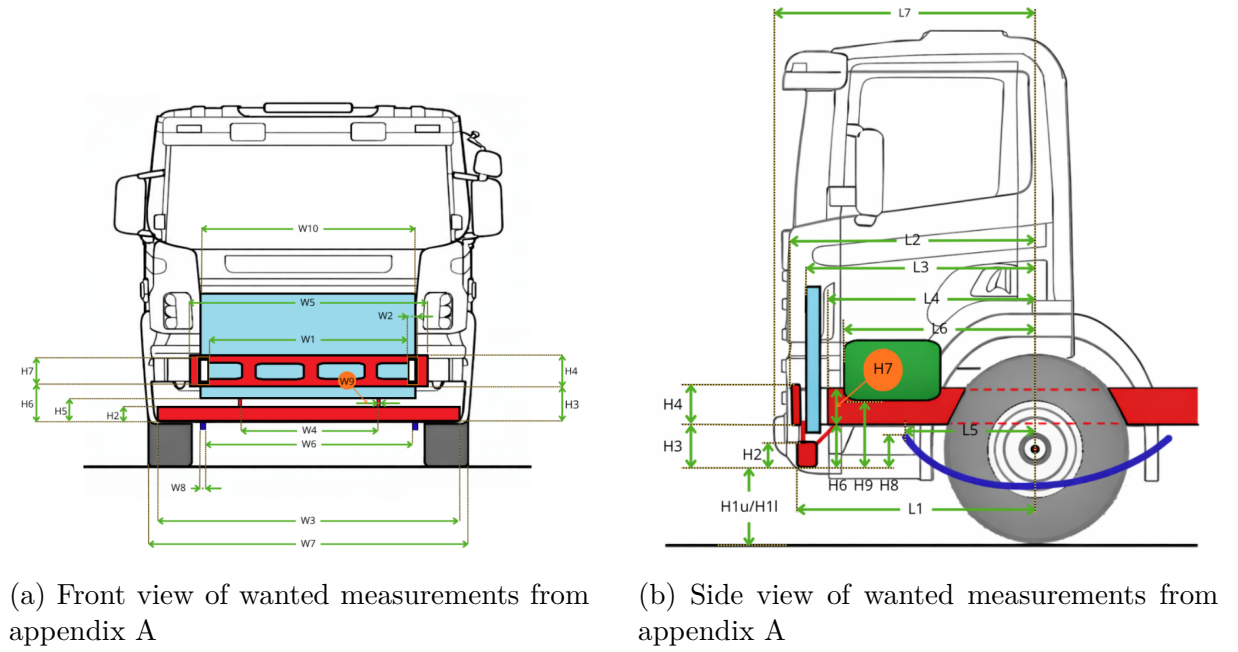


Figure 20: Relevant components and how to measure them according to appendix A

Due to the limited number of vehicles available for manual measurements and the restricted availability of CAD data, support from manufacturers was required to broaden the database. Relevant OEM representatives were therefore contacted and provided with a fillable measurement template and accompanying instructions. Figure 20 illustrates the visualization included in the distributed material, while descriptions of the individual parameters are provided in Appendix A.

In order to translate the measurements between the VC-Compat and TFCM database, table 4 explains the translations of the measurements.

Table 4: Conversion between VC-Compat measurements and the corresponding measurements in the Truck Front Components Measurements database.

VC-Compat measure	Meaning in VC-Compat	Converted from TFCM database
F1-u	Ground to lower bumper/front closure, unladen	H1u + H3
F1-l	Ground to lower bumper/front closure, laden	H1l + H3
F2	Bumper/front closure vertical height	H4
F3	Bumper/front closure width	W5
F4	Outer width of longitudinals/frame rails at bumper connection	W1 + 2×W2
F5-u	Ground to lower longitudinals/frame rails, unladen	H1u + H6
F5-l	Ground to lower longitudinals/frame rails, laden	H1l + H6
F6	Longitudinals/frame rails vertical height	H7
F8	Front bumper/front closure to front axle	L2
F9-u	Ground to lower FUP, unladen	H1u
F9-l	Ground to lower FUP, laden	H1l
F10	FUP vertical height	H2
F11	Front FUP to front axle	L1
F12	FUP width	W3

Note. For ground-related VC-Compat measures, the FUP ground clearance is added because several height measurements in the Truck Front Components Measurements database are measured relative to the lowest point of the FUP.

3.4.3 EU Cars Database_2026

PC structural and geometric data used in this study were obtained from the EU Cars Database_2026, which contains measurements from 45 PCs available through A2MAC1. As A2MAC1 is based on professionally documented vehicle teardowns and measured component geometries, the database provided accurate dimensional data representing the modern European PC fleet. This enabled comparison between PC load paths and contemporary HGV front structures within the compatibility analysis

3.5 Crash-relevant structure observations

During the project, several additional observations relevant to crash compatibility were identified. Beyond the initially selected components, most investigated trucks were equipped with a transverse tubular structure located at the foremost part of the vehicle front, slightly above the frame rails. In this report, the component is referred to as the torsion bar. Although positioned relatively high in the truck front, it may influence frontal structural interaction and was therefore included in the analysis.

Another recurring observation was the similar positioning of the steering gearbox and its connecting linkage across the investigated vehicles. In the investigated cases, it was

located far forward on the left-hand side of the truck.

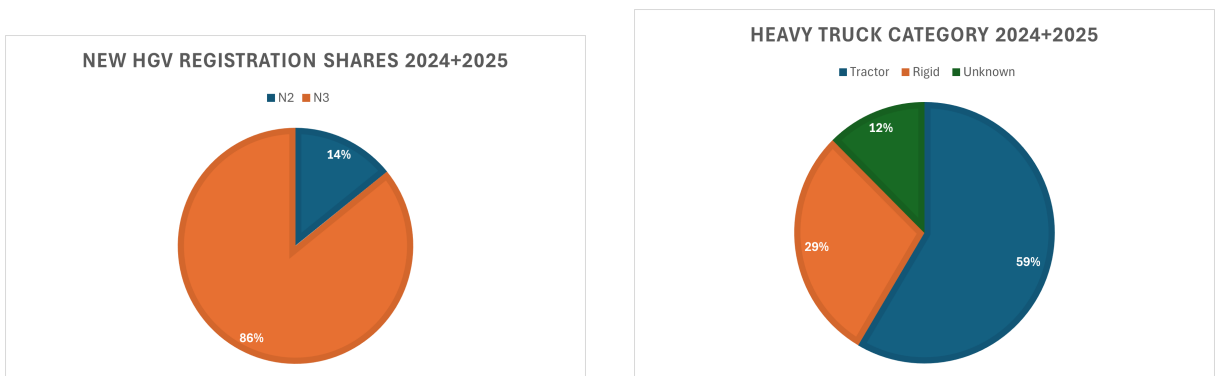
3.6 Organization and Filtering of Data

To improve comparison reliability, different inclusion criteria were applied depending on the measurement type. For ground-referenced measurements, only leaf-sprung vehicles from the TFCM database were included, as the true unladen ride height of air-suspended vehicles could not be verified during manual data collection. Measurements between structural components included all available vehicles, since these dimensions are independent of ride height. This approach was applied throughout the analysis, including the development of the average truck-front models.

To ensure that the TFCM database represented current road-going N3 truck designs, vehicles manufactured before 2020 and N2 vehicles were excluded. N2 vehicles were removed due to limited data availability, while vehicles before 2020 were removed to only represent the newer fleet.

Noticeable differences were also identified between the VC-Compat and TFCM datasets, indicating that the former may not fully represent the current truck fleet. Consequently, the compatibility analysis was based solely on the TFCM database and focused on vehicles produced between 2020 and 2026.

Registration statistics published by the ICCT (2025, 2026) show that the majority of newly registered HGVs belong to the N3 category, with tractor units representing the most common configuration. Figure 21 summarizes the distribution of vehicle categories and configurations.



(a) Registration shares between N2 and N3 trucks from 2024 and 2025.

(b) Type of HGV shares 2024 and 2025 according to vehicle registration data from ICCT

Figure 21: Shares of segment and type of new registered HGVs during 2024 and 2025 according to ICCT (2025,2026).

Eurostat (2026) data further indicate that the majority of driven distance is accumulated by N3 vehicles.

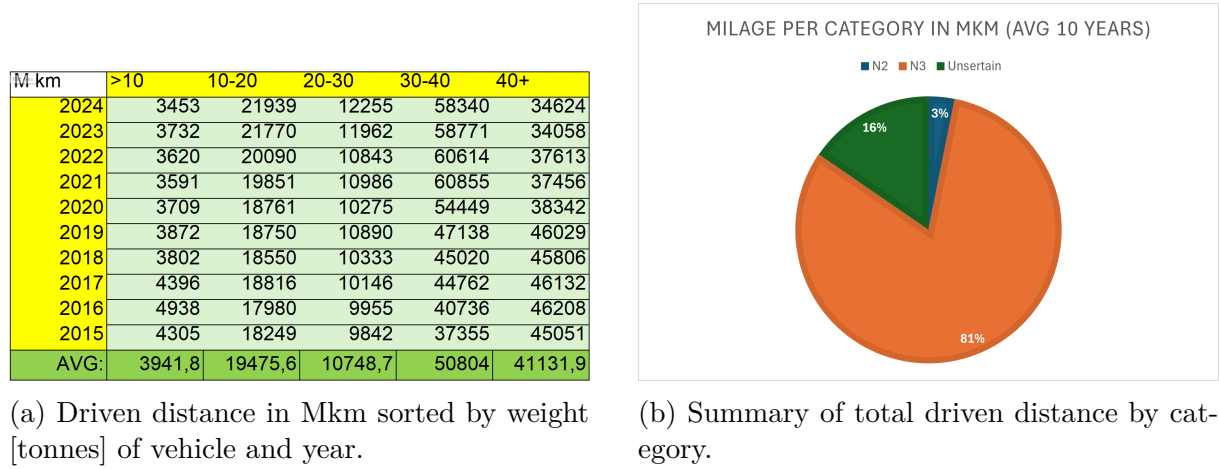


Figure 22: Overview of driven distance by the N2 and N3 truck category in the last 10 years. Based on data collected from Eurostat (2026).

Based on these sources, it can be concluded that approximately 9 out of 10 trucks encountered on European roads belong to the N3 category, and that roughly every second truck is an N3 tractor unit.

Together, these sources indicate that approximately 9 out of 10 HGVs operating on European roads belong to the N3 category and that roughly every second truck is an N3 tractor unit. Although N2 vehicles were excluded from the analysis, the remaining dataset therefore represents the majority of trucks encountered on European roads. However, substantial geometric differences may exist between N2 and N3 vehicles, meaning that the collected data should primarily be regarded as representative of the N3 segment.

3.7 Energy Absorption

This section presents simplified principles of crash energy absorption and their influence on frontal HGV–PC compatibility.

3.7.1 Energy Absorption Theory

For a frontal collision between a PC and a HGV, simplified equations for energy and impulse can be used. The initial kinetic energy of the system may be presented as the sum of the kinetic energies of the two vehicles. The PC mass is defined as m_p and its

initial velocity as v_p , while the truck mass is defined as m_t and its initial velocity as v_t . The total kinetic energy before impact is then

$$E_{k,i} = \frac{1}{2} (m_p v_p^2 + m_t v_t^2) \quad (2)$$

Due to the large mass ratio between HGVs and PCs, the truck typically undergoes only a small velocity change during impact, while the PC experiences a substantially larger velocity change. This is illustrated by Thomson et al. (2023), where a frontal collision between a Seat Leon and a Volvo FH resulted in a PC velocity change of approximately 100 km/h, compared to less than 6 km/h for the truck.

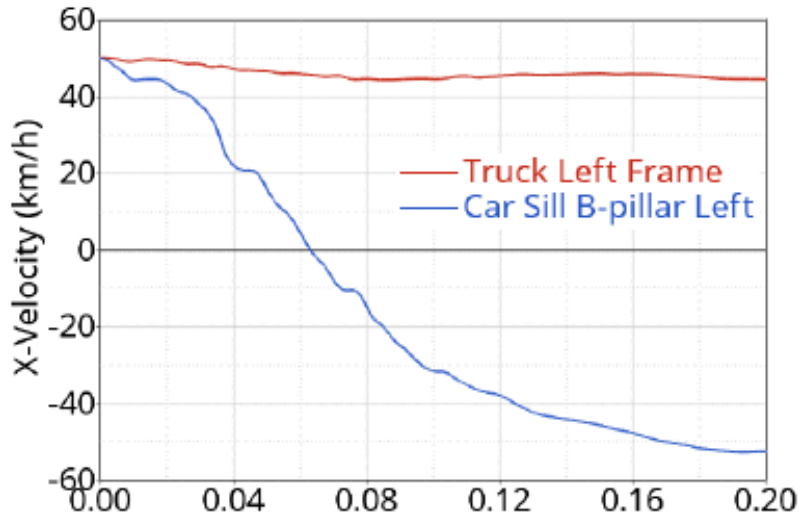


Figure 23: Car and truck velocity time profiles, adapted from Thomson et al. (2023).

This motivates that, in a simplified analysis, the crash can be treated as a perfectly inelastic collision. This means that the two vehicles are assumed to move together immediately after impact with a common velocity v_f . Since the two vehicles are travelling towards each other before impact, the velocities are treated with opposite signs. Applying conservation of linear momentum gives

$$m_p v_p - m_t v_t = (m_p + m_t) v_f \quad (3)$$

which can be rearranged to

$$v_f = \frac{m_p v_p - m_t v_t}{m_p + m_t} \quad (4)$$

After impact, the remaining kinetic energy of the combined system is therefore

$$E_{k,r} = \frac{1}{2}(m_p + m_t)v_f^2 \quad (5)$$

The difference between the initial and residual kinetic energy represents the energy dissipated during the collision, mainly through structural deformation. Thus, the total absorbed energy can be written as

$$E_d = E_{k,i} - E_{k,r} \quad (6)$$

By substituting the expression for v_f , the dissipated energy may be expressed as

$$E_d = \frac{1}{2} \frac{m_p m_t}{m_p + m_t} (v_p + v_t)^2 \quad (7)$$

This expression shows that the deformation energy depends on both vehicle masses and on the closing speed prior to impact. The total absorbed energy in the collision can then be considered as the sum of the energy absorbed by the PC and the truck structures, provided that other forms of energy dissipation are neglected:

$$E_d \approx E_p + E_t \quad (8)$$

Energy absorbed by the vehicle structures during a frontal crash can be estimated from the force–displacement curve at the deforming front end. For the PC, the absorbed energy can generally be calculated as the area under the force–displacement curve:

$$E_p = \int_0^{d_{\max}} F_p(d) \, dd \quad (9)$$

where $F_p(d)$ is the PC crush force as a function of deformation distance d , and $d_{\max}$ is the maximum deformation.

For a simplified approximation, the force–displacement relationship may be treated as linear. In that case, the absorbed energy of the PC front structure can be estimated as the area of a triangle:

$$E_p \approx \frac{1}{2} F_{\max} d_{\max} \quad (10)$$

where $F_{\max}$ is the peak crush force and $d_{\max}$ is the maximum deformation.

If the truck front is assumed to deform at an approximately constant force level, the absorbed energy in the truck structure may instead be approximated by

$$E_t \approx F_{\text{avg}} d_{\max} \quad (11)$$

where F_{avg} is the average crush force of the truck front structure and $d_{\max}$ is its deformation.

This energy model is a simplification of the crash event. It does not include all effects such as rotation, rebound, local fracture, friction, braking during the crash event, or other losses outside the main vehicle structures. However, it is useful for comparing simplified energy absorption levels between different vehicle structures.

4 Results and Analysis

This chapter presents the results obtained from the investigation of load-path compatibility between HGVs and PCs. The results include geometrical comparisons, structural interaction analyses, and other observations relevant to frontal crash compatibility between the investigated vehicle groups.

4.1 Collected Data

Ten trucks were considered suitable for a representative comparison between the current EU HGV and PC fleets. The included vehicle models are listed in Table 5. To avoid linking specific structural data to individual vehicle models, the order of the HGVs in Table 5 was randomized relative to the diagrams presented in Appendix C. The remaining vehicles were excluded due to incomplete data availability. This approach was adopted to maximize the representativeness of the dataset.

Table 5: Investigated truck models

Manufacturer	Model	Model year	Configuration	Suspension
Volvo	FH H2	2020	Tractor	Air
MAN	Etgx	2026	Tractor	Leaf
Mercedes	eActros	2023	Rigid	Air
Iveco	S-way 500	2024	Tractor	Leaf
Scania	25 L	2021	Rigid	Air/Leaf
MAN	TGX 18.480	2025	Tractor	Leaf
Mercedes	Actros L 1848	2024	Tractor	Leaf
Volvo	FMX 500	2024	Tractor	Air
Renault	T High 520	2026	Tractor	Leaf
MAN	TGX	2025	Tractor	Leaf

4.2 Observations and Patterns

During the project, several observations were made related to structural patterns, component placement, and compatibility aspects identified during the analysis of the collected data. These observations are presented in this section.

4.2.1 Structural Observations

During the development of the TFCM database, several observations were made regarding the position and design of truck-front components. Components located close to the vehicle front may influence crash interaction if the FUP does not provide sufficient load distribution or stopping capability.

One observation was the variation in frame-rail position. In some vehicles, the frame rails extended close to the front face of the truck, leaving limited deformation space ahead of these structures. If the FUP is unable to sufficiently distribute or absorb the impact load, the frame rails may become a rigid contact point for the PC.



Figure 24: Illustration of frame rail extended close to the front face of the truck generated by AI for visualization purposes.

The front ends of the leaf springs were also often positioned far forward and relatively low. Consequently, they may become involved in the crash interaction if the PC continues beneath or beyond the FUP. Since the front face is relatively small, the resulting load transfer may become more concentrated. All measured leaf springs had a front-face width of 90 mm, although their position relative to the truck front varied.

Another recurring observation was the forward positioning of steering components. The steering gearbox was typically located low on the left-hand side of the truck and, on average, approximately 376 mm behind the FUP. The pitman arm and drag link were also positioned relatively low, close to the leaf springs.

These components are unlikely to influence the initial PC interaction but may become

relevant in small-overlap collisions, where damage to the steering gearbox or linkage could affect post-impact steering capability.

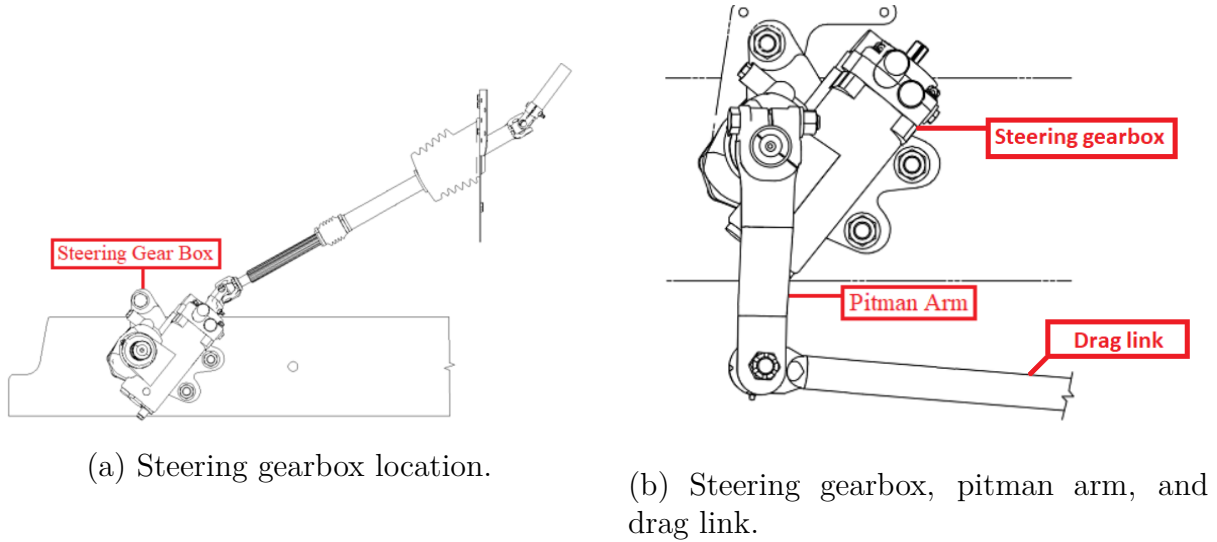


Figure 25: Overview of HGV steering components. The steering gearbox location is reproduced from Daimler Truck North America (n.d.), while the annotated steering linkage image is adapted from Daimler Truck North America (n.d.).

Several trucks also employed additional FUP support structures. The Iveco S-Way 500 (2024), Mercedes-Benz Actros L 1848 (2024), and eActros (2023) all used supplementary outer brackets positioned outside the primary supports. Such arrangements may improve resistance to local loading and reduce FUP cross-member bending during offset and small-overlap impacts.

4.2.2 Small-Overlap Load Path and Wheel Interaction

As presented in the literature study, the small-overlap scenario focuses on the structural interaction between the PCr longitudinal members and the truck FUP system in low-overlap crash configurations. Particular attention is given to the lateral position of the PC load paths relative to the FUP cross-member supports, since engagement outside these supports concentrates the impact load in an unsupported region of the cross-member. This may lead to local failure of the FUP, allowing the PC to bypass the protection system and continue underneath the truck, as demonstrated in the car-to-HGV crash test presented by Thomson et al. (2023).

A similar behaviour has also been observed in previous simulation work. Lefer and Reboloso (2012) showed that when the FUP lacks sufficient rigidity, the PC can continue through the FUP region and interact with the truck wheel instead of being stopped or

redirected by the FUP system similar to what is illustrated in figure 16.

If the PC continues underneath the truck, direct interaction with the front wheel may occur. This can lead to loading of the wheel and suspension components, potentially damaging the suspension and displacing the wheel rearwards. The wheel may also continue to climb onto the front structure of the PC and act as a secondary intruding component, causing additional damage to the PC chassis.



Figure 26: Sequence from the car-to-HGV crash test showing interaction between the PC and the truck front wheel, adapted from Thomson et al. (2023).

Rearward wheel displacement may in turn compromise or eliminate the truck's steering capability. Even if the suspension and steering linkage remain intact, loss of steering may still be likely, since in the investigated N3 truck configurations the steering box is commonly positioned at the far-left front side behind the FUP. Consequently, small-overlap crashes may not only present challenges for PC occupant protection, but could also affect vehicle controllability following the impact.

The car-to-HGV crash test presented by Thomson et al. (2023) also demonstrated that the FUPD can fracture when the load is not sufficiently distributed into the intended support structure. In this case, the left outboard part of the FUPD fractured close to the connection bracket, which reduced the possibility of maintaining controlled structural interaction between the truck and the PC.

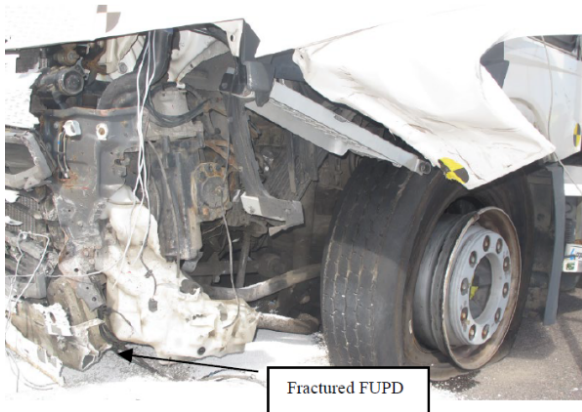


Figure 27: Fractured FUPD after the car-to-HGV crash test, adopted from Thomson et al. (2023).

To mitigate this failure mode, the FUP system must be able to maintain structural engagement even in low-overlap configurations where the PC load path is positioned outside the primary FUP support locations. A more load-capable FUP cross-member could reduce the risk of local collapse in the unsupported outer region and help distribute the impact load over a larger part of the truck front structure.

During the investigation of N3 trucks, it was observed that some manufacturers use an additional FUP support near the outer edge of the cross-member. One example is the Mercedes-Benz Actros, which uses an angled support extending towards the outer end of the FUP. Such a design may provide additional resistance against loading in an otherwise less supported area and reduce bending of the cross-member. This could improve the likelihood that the PC load paths remain engaged with the FUP system and reduce the risk of the vehicle continuing underneath the truck and subsequently interacting with the front wheel. A similar increase in resistance could also be achieved by using a stiffer FUP cross-member, however, adding local support in the critical outer region may provide a more weight-efficient solution.

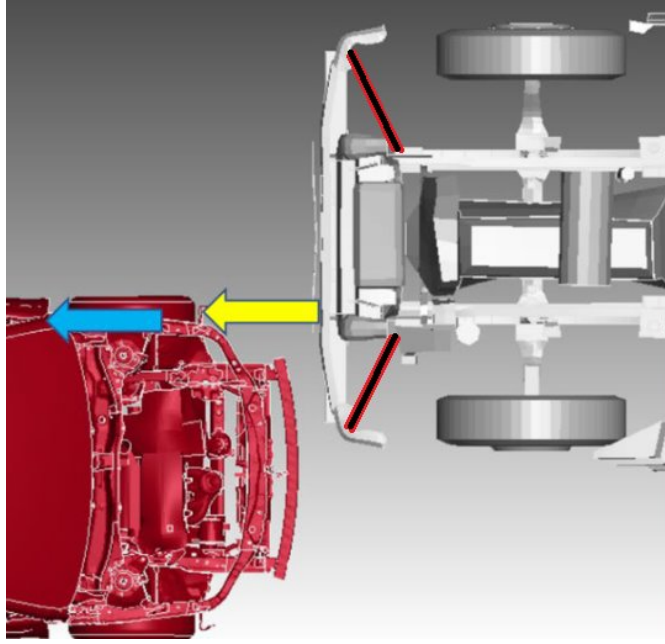


Figure 28: Illustration of an additional outer FUP support concept, similar to the support layout observed on the Mercedes-Benz Actros. Modified from Thomson et al. (2023).

This interpretation is consistent with the conclusions of Thomson et al. (2023), who identify insufficient structural interaction, together with both horizontal and vertical misalignment between the PC and the FUPD, as key contributors to the poor outcome in the car-to-HGV crash test. Thomson et al. further argue that an improved FUPD should be designed to better withstand the applied forces, improve geometric compatibility, and prevent underride. However, once underride prevention has been achieved, additional energy management is still required. In this context, an extended truck front could provide additional deformation space and thereby improve crash compatibility and occupant survivability in passenger-car-to-truck frontal impacts.

As described in the section on glance-off behaviour, an alternative strategy is to use an angled or curved FUP with the intention of promoting lateral deflection rather than full frontal engagement between the vehicles. This approach is used by some manufacturers; for example, one truck included in the present study, the MAN eTGX 2026, uses a curved FUP over the whole width. However, as discussed previously, such designs must be balanced against the need for sufficient cross-member stiffness and load distribution in higher-overlap impacts.

4.3 VC-Compat Comparison and Dataset Selection

To assess whether the truck-front measurements in the VC-Compat database, collected around the year 2000, are still representative of modern N3 trucks, a comparison was carried out between the VC-Compat dataset and the newly collected Truck Front Components Measurements database. Equivalent measurements were compared between the two datasets in order to identify potential geometric changes in truck-front design over time.

The comparison was visualized using boxplots for each parameter where comparable data were available in both datasets. The plots show the median, mean, 10th–90th percentile range, 25th–75th percentile range as blue box and individual vehicle measurements. The VC-Compat dataset contains data from 53 N3 trucks for which both laden and unladen heights are reported. In the Truck Front Components Measurements database, the number of vehicles varies between parameters depending on measurement availability, since the data were primarily obtained through manual measurements and only unladen measurements were recorded.

For several measurements, the components are located at similar positions in the two datasets. One example is the F6 measurement, which describes the longitudinal/frame rail height. As shown in Figure 29, the frame rail height is relatively similar between the VC-Compat dataset and the Truck Front Components Measurements database, indicating that some structural dimensions have remained comparable between the datasets.

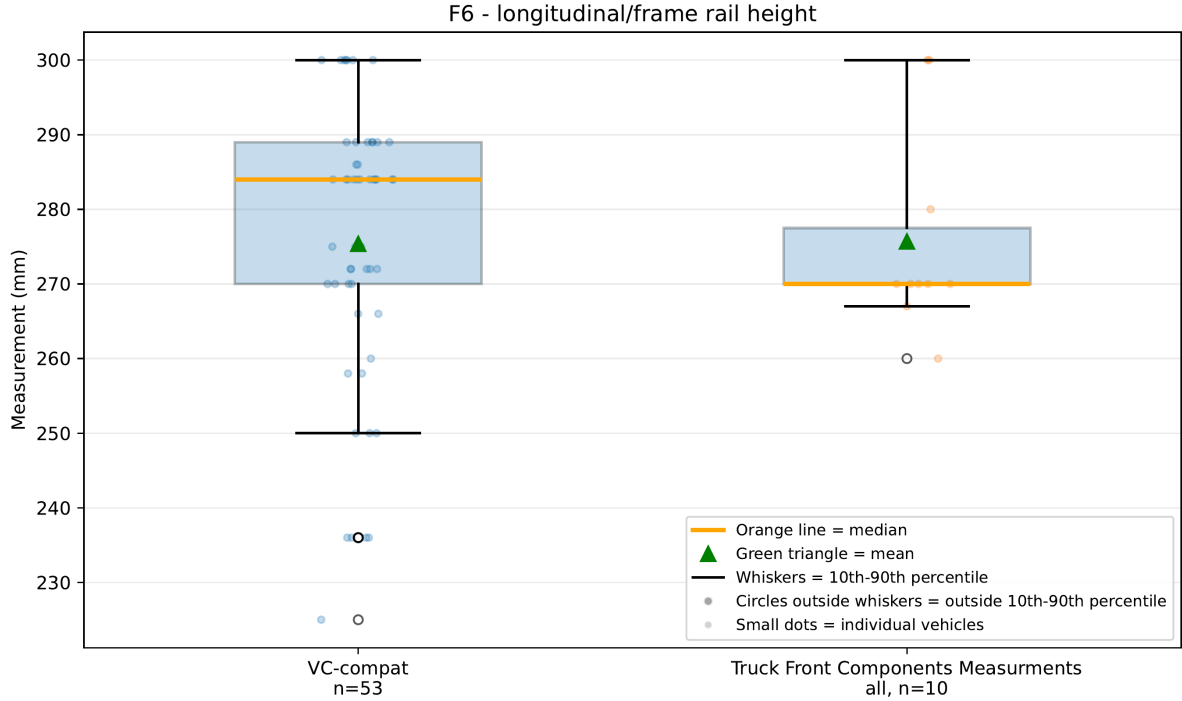


Figure 29: Comparison of F6 longitudinal/frame rail height between the VC-Compat dataset and the Truck Front Components Measurements database.

However, some larger deviations can also be observed. One example is the F3 measurement in VC-Compat, which describes bumper width and corresponds approximately to the bumper beam or front closure width in the Truck Front Components Measurements database. As shown in Figure 30, the VC-Compat data show a much larger spread and appear to contain two larger concentrations of data. This could indicate that the measurement may have been taken in two different ways, or that different components may have been measured. This is possible since the VC-Compat project included several contributors and measurement sources.

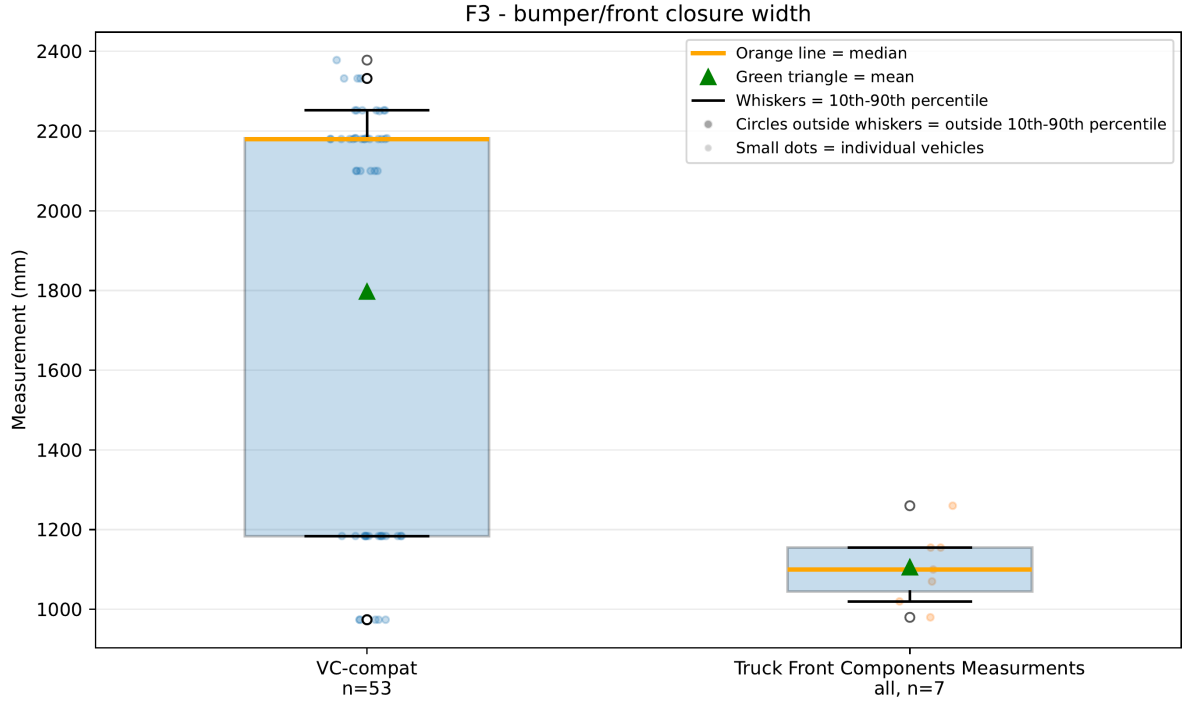


Figure 30: Comparison of F3 bumper/front closure width between the VC-Compat dataset and the Truck Front Components Measurements database.

A second important difference can be seen for the F9-u measurement, which describes the vertical position from the ground to the FUP. This parameter is particularly relevant for the present study, since the vertical position of the FUP directly affects the geometric compatibility with PC load paths. As shown in Figure 31, the average FUP height from the ground is lower in the Truck Front Components Measurements database than in the VC-Compat dataset. This indicates that the two datasets should not be used interchangeably for FUP vertical positioning in the compatibility analysis.

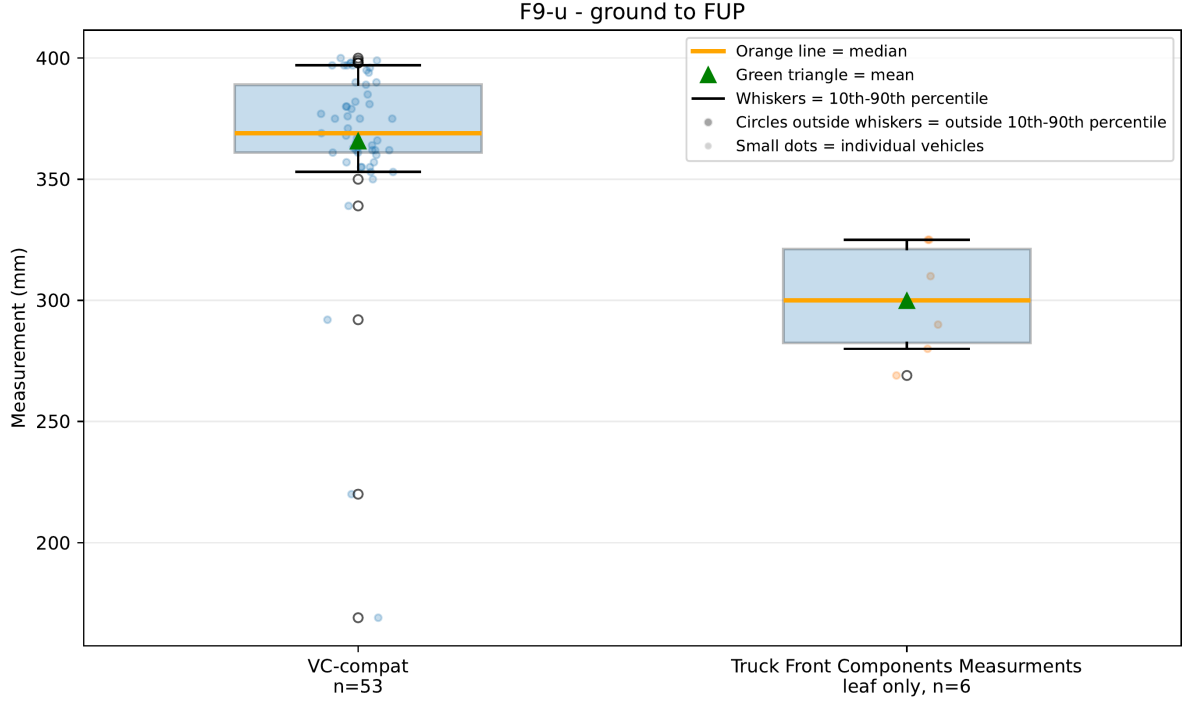


Figure 31: Comparison of F9-u ground-to-FUP height between the VC-Compat dataset and the Truck Front Components Measurements database.

To summarise the numerical differences between the datasets, table 6 presents the mean and median values for each comparable parameter. The differences are calculated as the Truck Front Components Measurements value minus the VC-Compat value.

Table 6: Mean and median differences between the VC-Compat database and the Truck Front Components Measurements database.

Parameter	VC-Compat mean [mm]	VC-Compat median [mm]	TFCM mean [mm]	TFCM median [mm]	Mean difference [mm]	Median difference [mm]
F1-u	647.8	732.4	730.0	735.0	+82.2	+2.6
F2	190.3	172.0	158.3	155.0	-32.0	-17.0
F3	1797.2	2180.0	1105.7	1100.0	-691.5	-1080.0
F4	955.8	903.0	1010.0	1055.0	+54.2	+152.0
F5-u	730.0	738.0	671.7	680.0	-58.3	-58.0
F6	275.4	284.0	275.7	270.0	+0.3	-14.0
F8	1412.5	1401.0	1348.6	1345.0	-63.9	-56.0
F9-u	365.7	369.0	299.8	300.0	-65.9	-69.0
F10	137.4	120.0	134.0	140.0	-3.4	+20.0
F11	1286.6	1290.0	1357.2	1300.0	+70.6	+10.0
F12	2084.2	2030.0	2226.0	2210.0	+141.8	+180.0

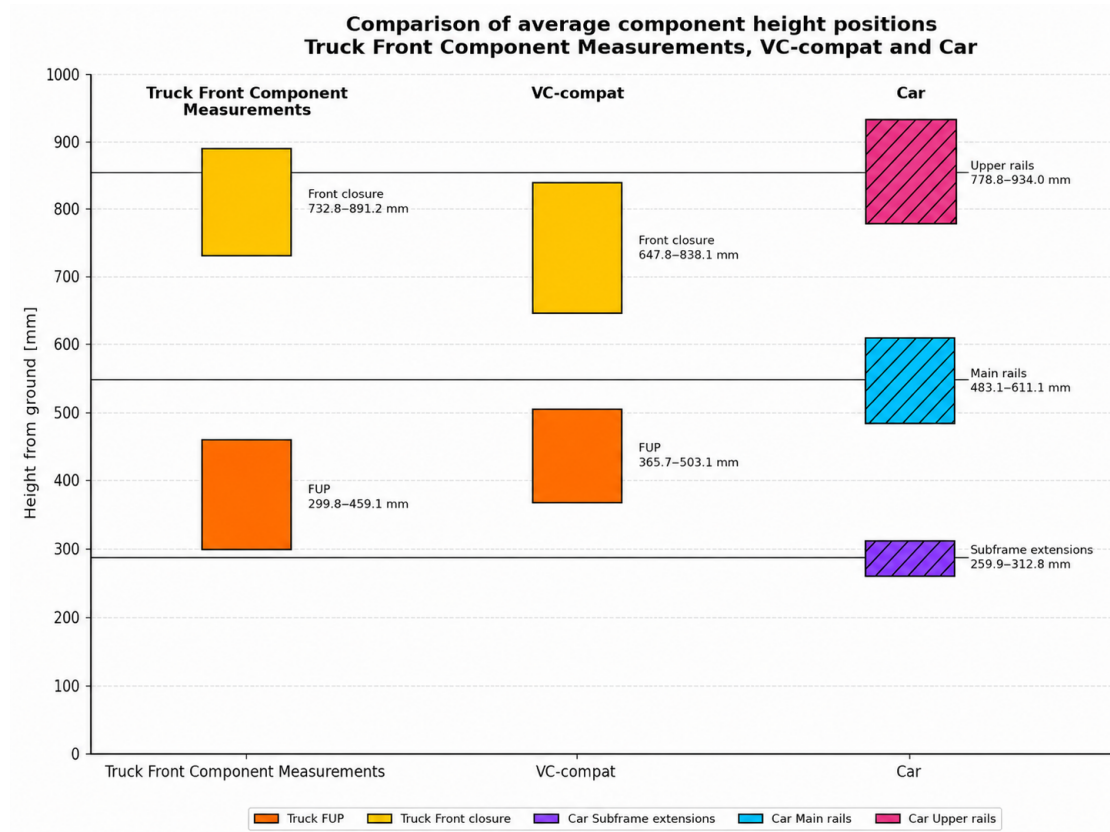


Figure 32: Comparison of average vertical component positions in the TFCM database, the VC-Compat database, and for PCs the EU Cars Database_2026.

Figure 32 summarises the main reason why VC-Compat was not used as the primary truck dataset in the average model compatibility analysis. Although the general vertical layout is similar, the figure shows noticeable differences in components that are central to the compatibility assessment, such as the FUP and the front closure.

The FUP position is particularly important, since it directly affects the comparison with the PC lower and main rails. A shift in FUP height can change whether the PC structures appear to engage the FUP or pass above or below it. The front closure position is also relevant for the interaction with higher PC structures, such as the upper rails, shown in the comparative overview.

Overall, the two datasets show similar trends for several parameters, but the differences in key vertical positions mean that they should not be used interchangeably for the compatibility analysis. Therefore, the VC-Compat database was used mainly as a reference for comparison, while the Truck Front Components Measurements database was used as the primary truck dataset for the average model compatibility analysis.

4.4 Comparative overview

In this section, an overview of the frontal structures representing the modern truck fleet is presented. The data from the VC-Compat project is therefore not included in this section.

4.4.1 Front View: Truck

Figure 33 presents the simplified front-view representation including the investigated components, all of them with the average measurements from the data collection. The figure shows the selected truck components included in the geometric compatibility assessment, including the FUP, FUP brackets, frame rails, front leaf springs, radiator, front closure, and torsion bar. The purpose of the figure is to provide an overview of the vertical and lateral positions of the main truck-front structures before comparing them with the PC load paths. The heights of the FUP brackets and the front leaf spring front face are approximated to 100 mm and 70 mm, respectively.

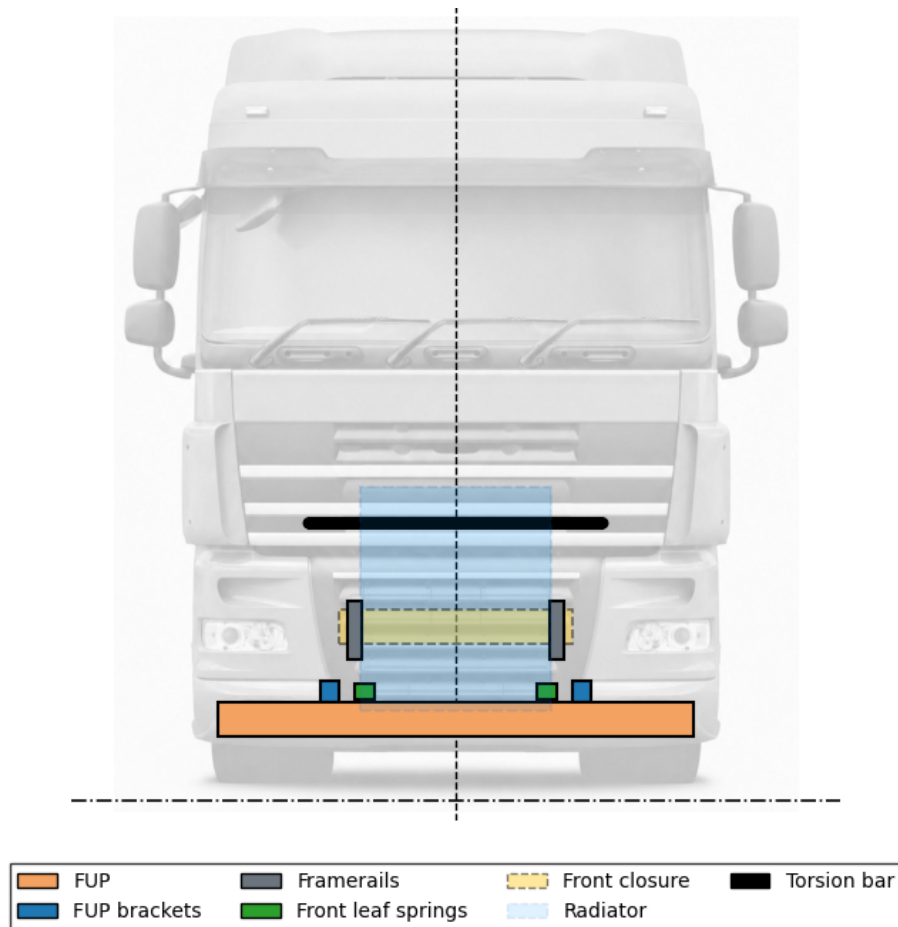


Figure 33: Simplified front-view representation of the average truck-front structure, showing selected components included in the geometric compatibility assessment.

4.4.2 Front View: PC

Figure 34 presents the simplified front-view representation of the average PC front structure. The figure shows the selected load-carrying components included in the geometric compatibility assessment, including the subframe extensions, main rails, lower crossbeam, bumper crossbeam, crash cans, subframe, and upper rails. The average width of the PC is shown by the dotted lines. The purpose of the figure is to provide an overview of the vertical and lateral positions of the main PC load paths before comparing them with the truck front structures.

The subframe extensions, crash cans, and lower crossbeam are positioned in the lower front region of the vehicle, while the main rails and bumper crossbeam are located higher up in the front structure. The upper rails represent an additional upper load path and are positioned at the highest level among the included components.

Since the lateral position of the upper rails was not available in the EU Cars Database_2026, their lateral placement was estimated using a Dacia Sandero as a reference vehicle. The Dacia Sandero was selected because it was the most sold PC in Europe during 2025 (EU Cars Database_2026). The lateral upper rail position was measured from A2MAC1 CAD geometry. However, since the upper rail lateral position is based on a single reference vehicle, it should be interpreted as an approximate visual indication rather than as a dataset-based average value.

The subframe is shown simplified as a transverse structure across the front of the vehicle, although its vertical position and height are based on the EU Cars Database_2026.

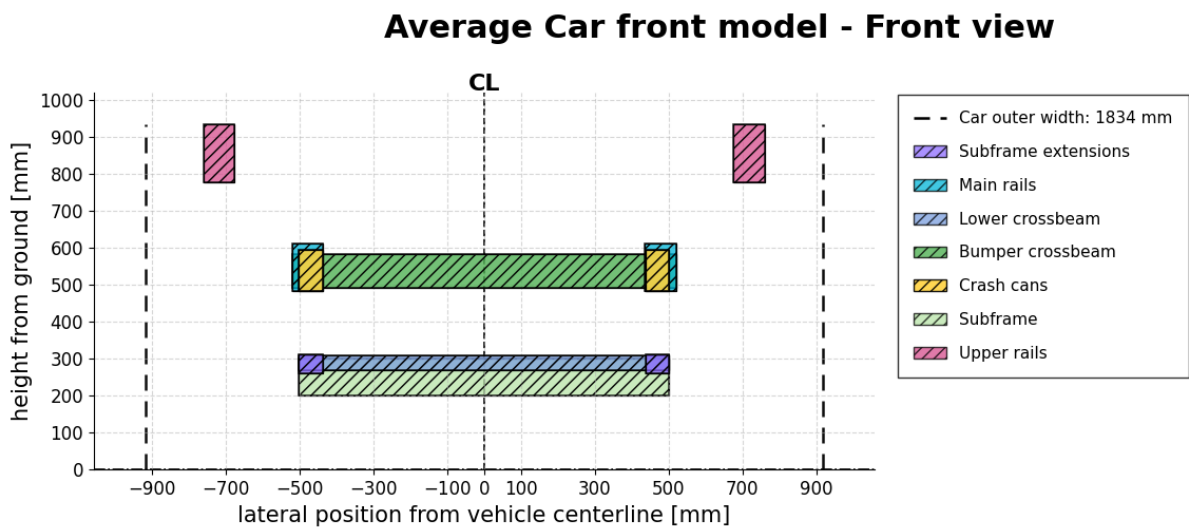


Figure 34: Simplified front-view representation of the average PC front structure, showing selected load-carrying components included in the geometric compatibility assessment.

4.4.3 Front-View Overlay Between Truck and PC Structures

The overlay should be interpreted as a simplified representation of the average structural layout rather than a complete front-end model. It is based on the EU Cars Database_2026 and the TFCM database. Only the most relevant load-carrying components are included in order to avoid visual clutter. The PC crossbeams are therefore excluded from figure 35, although they are located at similar vertical positions as the corresponding rail structures and may contribute to the same general load-transfer paths. In addition, the upper rail width is estimated, meaning that its lateral representation should be interpreted with some caution.

Figure 35 presents a combined front-view overlay of the average truck front structure and the average PC front structure. The purpose of the figure is to visualize the vertical and lateral alignment between the main load-carrying structures of the two vehicles.

The average truck FUP is positioned at approximately 300 mm above ground, while the average PC subframe extensions are positioned slightly lower, at approximately 260 mm. This indicates that the subframe extensions are located below the main FUP level, resulting in only limited vertical interaction between these structures in the average configuration.

Higher up in the structure, the PC upper rails show a closer vertical alignment with the truck front closure and are also positioned near the lower region of the truck frame rails. This suggests a potential interaction zone between the PC upper rails and the higher structural components of the truck front.

In contrast, the average PC main rails are positioned around 480 mm above ground and do not show a clear alignment with one of the main truck load-carrying structures in this front-view overlay. Overall, the overlay indicates that the average truck and PC structures do not show a strong vertical alignment across all primary load paths, but rather a partial alignment at certain levels. This may contribute to a load transfer that is inefficient compared to a configuration with more direct structural matching.

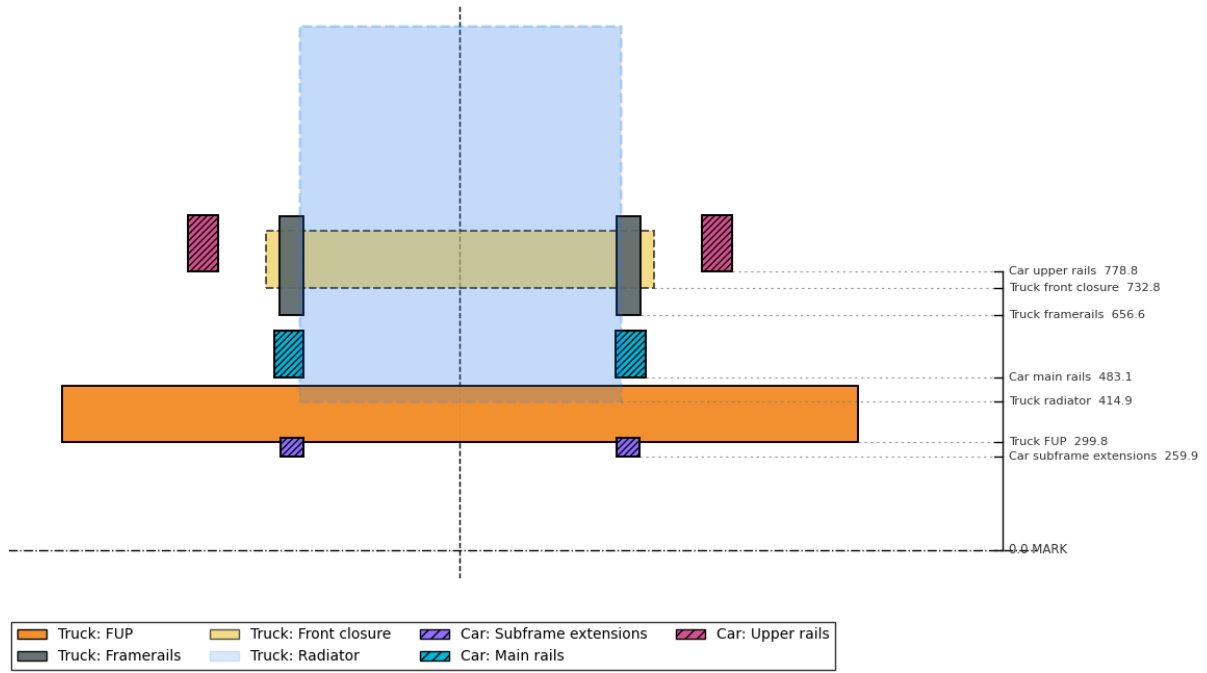


Figure 35: Combined front-view overlay of the average truck front structure and the average PC front structure, illustrating the vertical and lateral alignment between selected load-carrying components.

4.4.4 Overlap

Figures 36–45 present five offset overlap scenarios between the average truck front structure and the average PC front structure. The main scenarios correspond to 75%, 50%, and 25% overlap of the PC width, with additional detailed cases at approximately 34% and 29%. The purpose of these figures is to visualize how the lateral position of the PC load paths influences their geometric alignment with the selected truck front structures. The solid outline represents the truck outer width, while the dashed outline represents the PC outer width. For each overlap case, both a front-view overlay and a simplified top-view schematic are included.

75% overlap

At 75% overlap, the PC structures show the clearest lateral alignment with the FUP among the studied partial-overlap cases. The subframe extension, main rail, and their corresponding crossbeams are largely positioned within the FUP width, which indicates a greater possibility for load-path engagement. However, one side of the PC rails still misses the FUP laterally, and the vertical alignment remains unchanged compared with the front-view comparison shown in Figure 35.

Due to this remaining vertical misalignment, one main rail is positioned in the radiator

region rather than directly aligning with the FUP. The upper rail shows good alignment with the front closure. Compared with the 50% and 25% cases, this scenario therefore indicates the greatest potential for structural interaction between the PC load paths and the selected truck-front components.

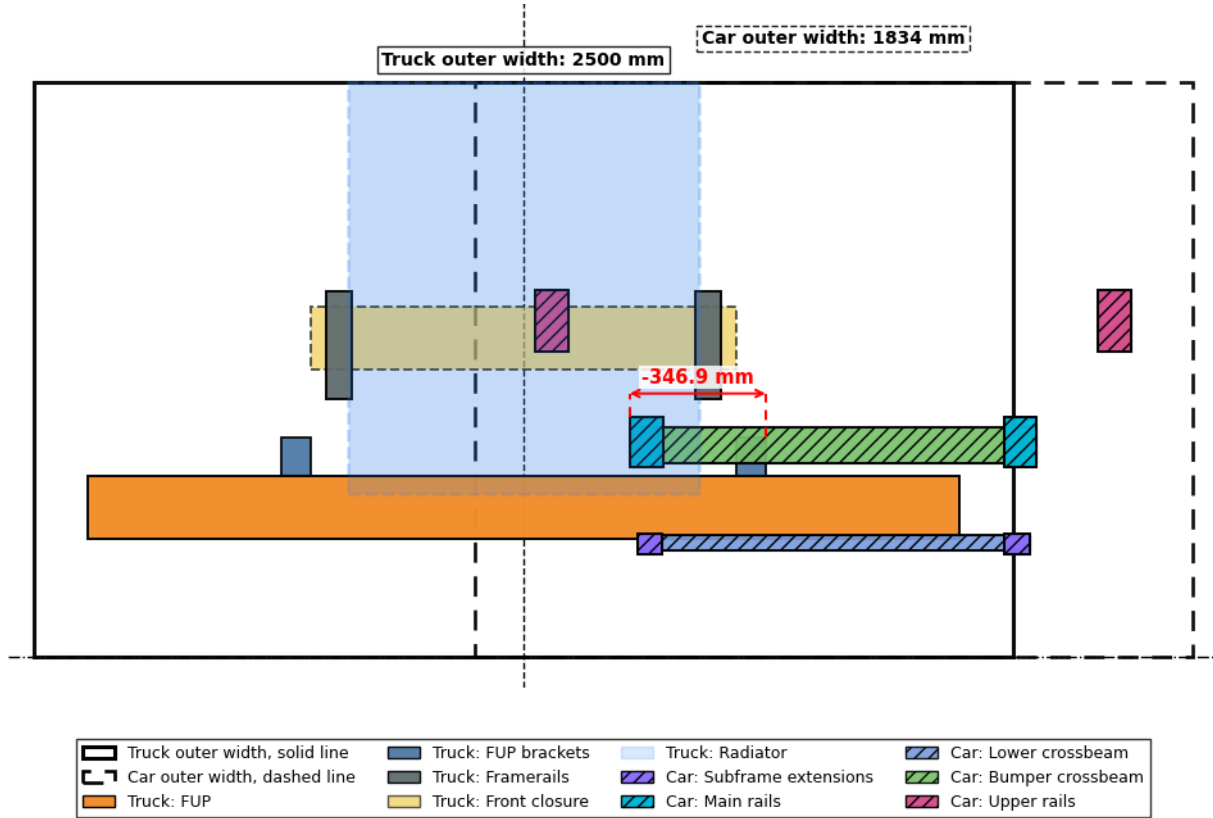
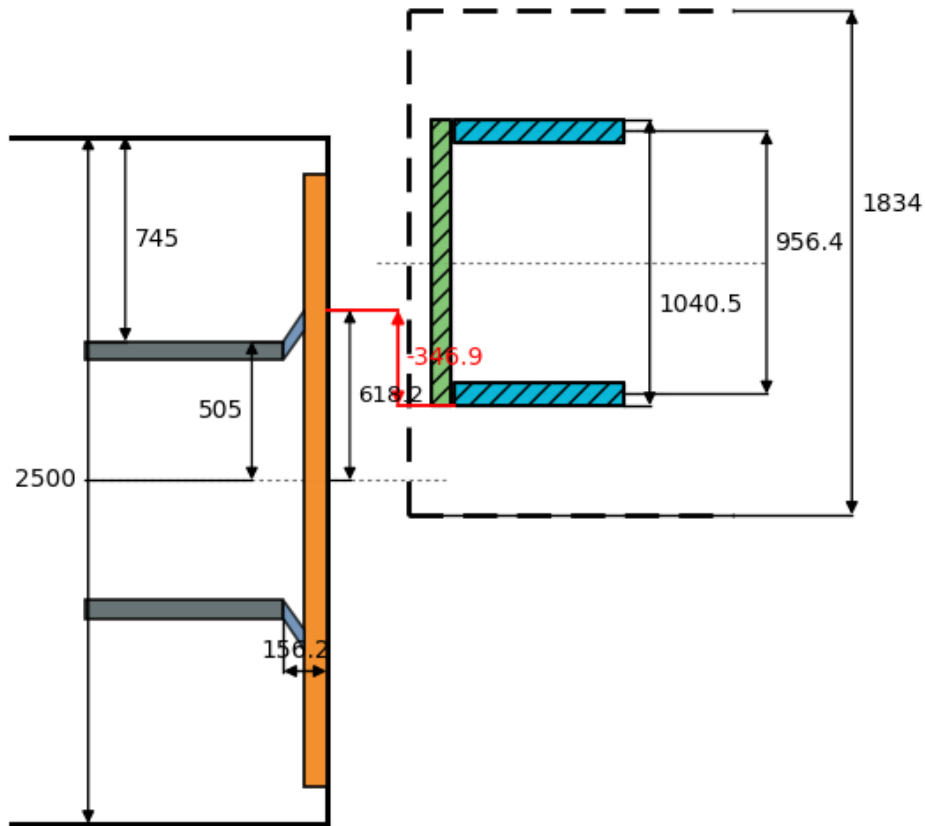


Figure 36: Average front-view overlay at 75% overlap between the truck front structure and the PC front structure.

Offset: 75% (1375.5 mm overlap)



— HGV outer width	— FUP	— Truck frame rails	— Car bumper crossbeam
— Car outer width	— FUP brackets	— Car main rails	

Figure 37: Simplified top-view schematic at 75% overlap, showing the lateral relationship between the FUP, FUP brackets, truck frame rails, and PC main rails.

50% overlap.

At 50% overlap, the PC structures show partial lateral alignment with the truck FUP and front structures. The PC rails and crossbeams are partly positioned within the FUP width, which indicates some possibility for load-path engagement, although the alignment is less complete than in the 75% case. The upper rail shows partial alignment with the truck frame rail and front closure region.

Compared with the 25% case, the PC load paths show greater lateral alignment with the FUP. This makes the 50% overlap case an intermediate configuration, where partial engagement may occur but full load-path compatibility remains limited.

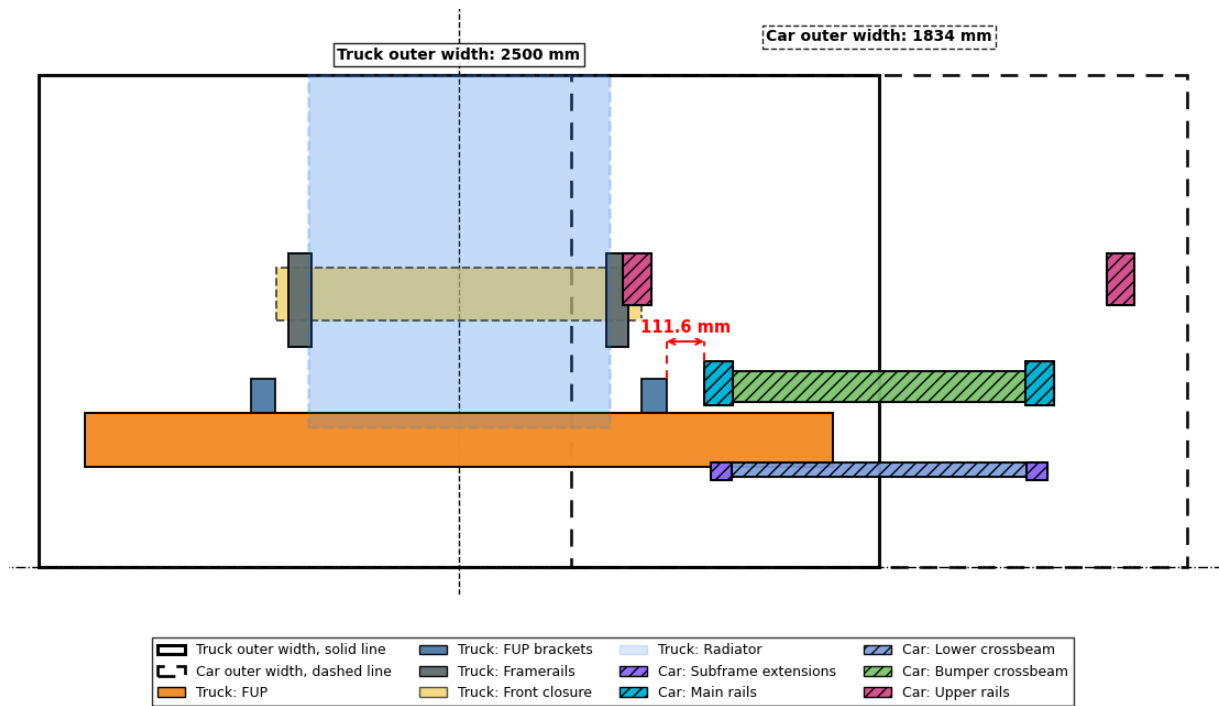
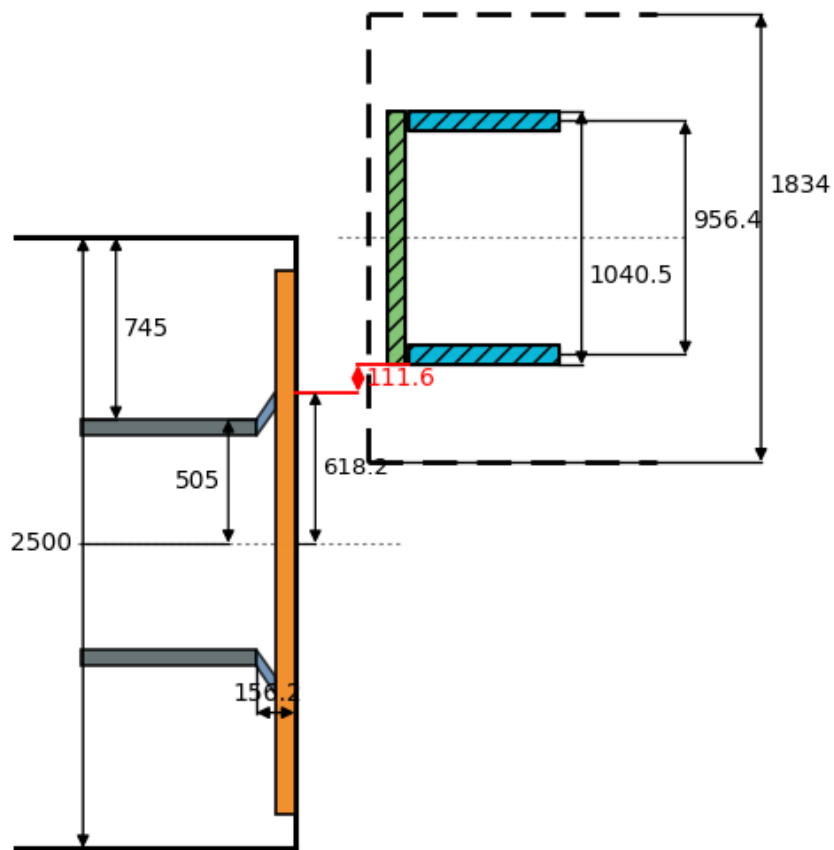


Figure 38: Average front-view overlay at 50% overlap between the truck front structure and the PC front structure.

Offset: 50% (917 mm overlap)



— HGV outer width	— FUP	— Truck frame rails	— Car bumper crossbeam
— Car outer width	— FUP brackets	— Car main rails	

Figure 39: Simplified top-view schematic at 50% overlap, showing the lateral relationship between the FUP, FUP brackets, truck frame rails, and PC main rails.

25% overlap

At 25% overlap, the PC load paths are mainly positioned outside the central truck front structures. The PC rails and crossbeams show no lateral alignment with the FUP, while the upper rail is largely offset from the truck frame rail region. Compared with the 50% and 75% cases, this represents a more critical small-overlap configuration. The outer PC structure still overlaps with the FUP region, but the primary PC load paths may not engage efficiently with the main truck front structures.

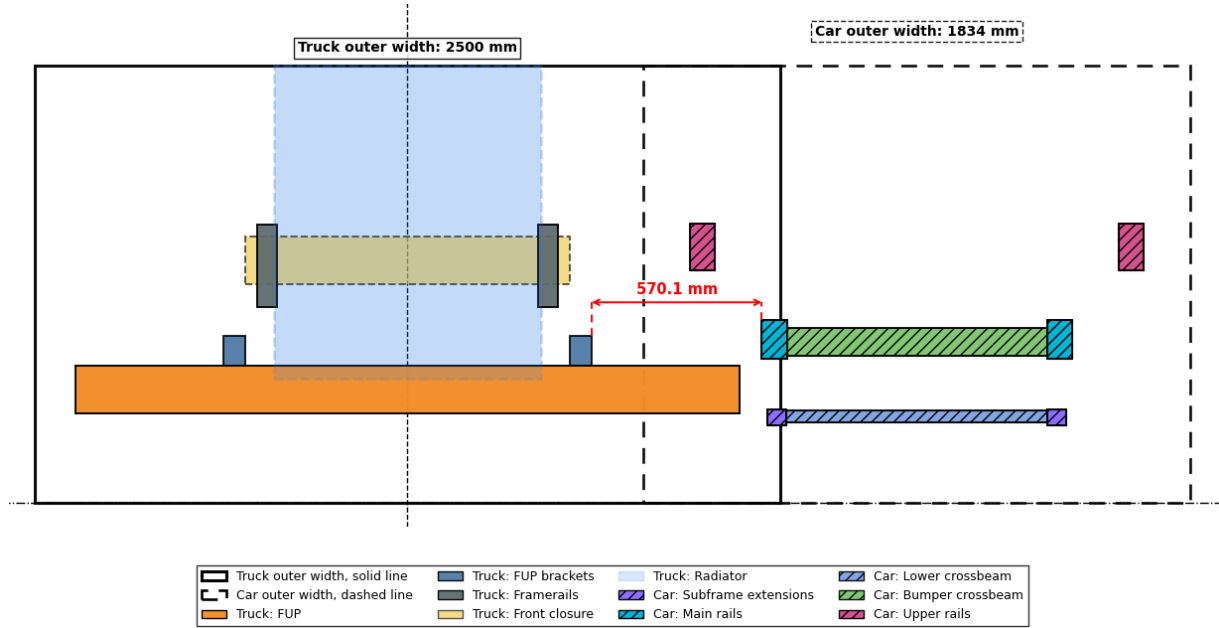
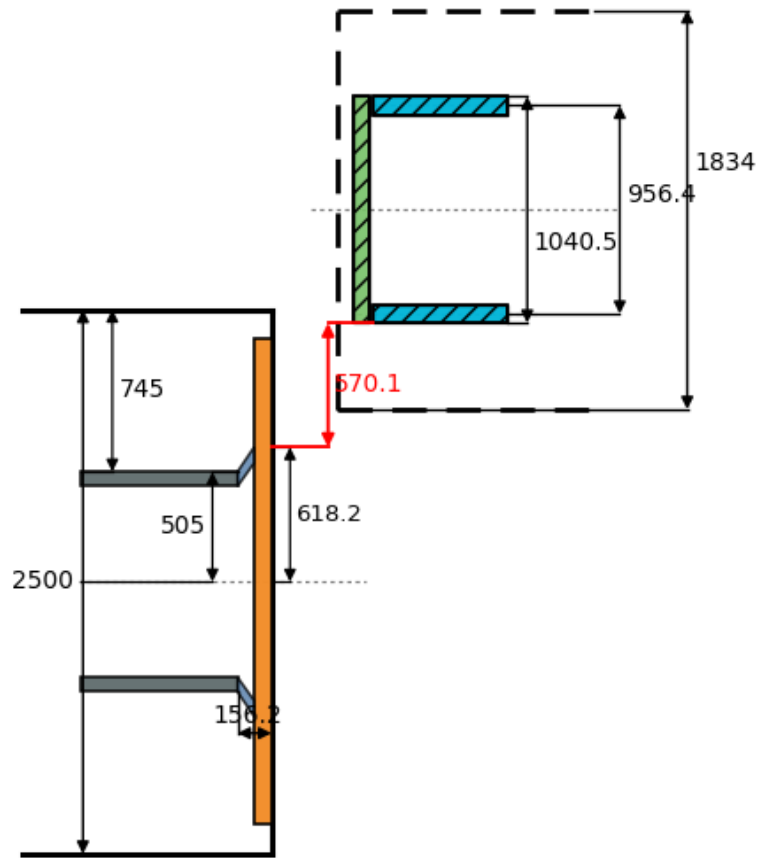


Figure 40: Average front-view overlay at 25% overlap between the truck front structure and the PC front structure.

Offset: 25% (458.5 mm overlap)



— HGV outer width	— FUP	— Truck frame rails	— Car bumper crossbeam
— Car outer width	— FUP brackets	— Car main rails	

Figure 41: Simplified top-view schematic at 25% overlap, showing the lateral relationship between the FUP, FUP brackets, truck frame rails, and PC main rails.

The 75%, 50%, and 25% cases show a clear trend: as the overlap decreases, the PC load paths move progressively away from the central truck front structures. This reduces the likelihood of direct load transfer into the main FUP and frame rail region. Since the smaller-overlap cases represent a more critical geometric configuration, two additional overlap positions were examined in more detail.

High loading on the FUP

A critical loading case was identified at approximately 34% overlap. In this configuration, the PC main rail reaches the outer FUP region, but the contact location is far from the

nearest FUP bracket. This creates a large moment arm between the impact location and the FUP support, indicating a possible risk of localised bending or rotational loading of the FUP cross-member. Even if the FUP is contacted by a main PC load path, the load may therefore be difficult to transfer efficiently into the FUP support structure.

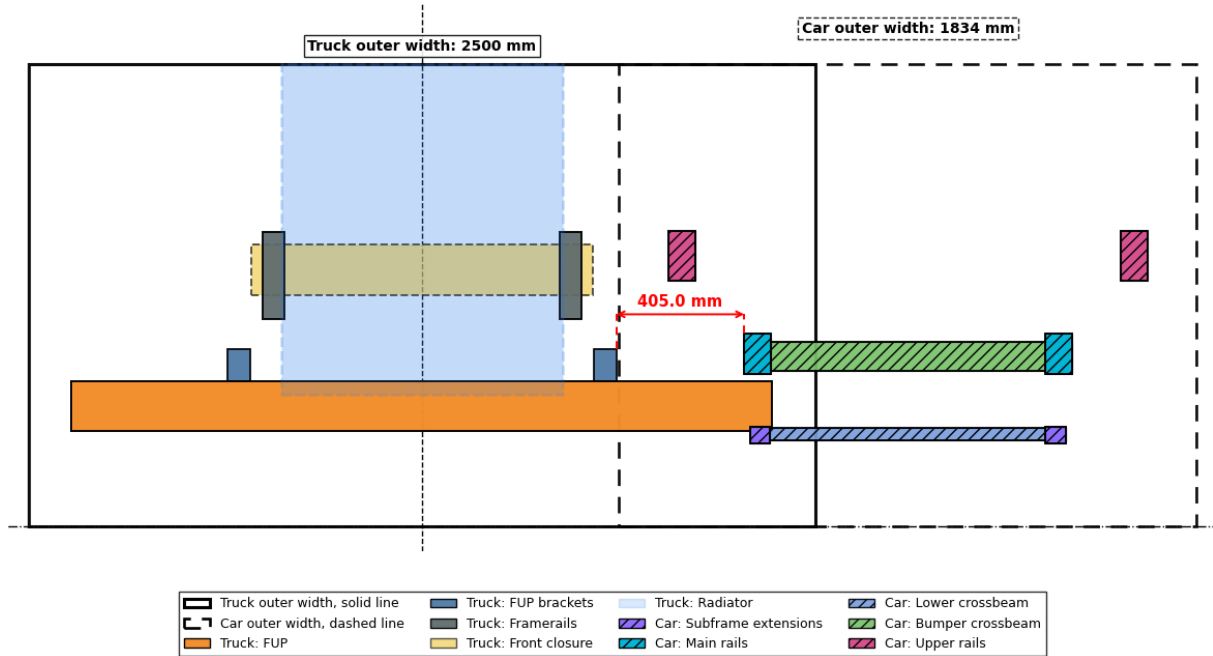
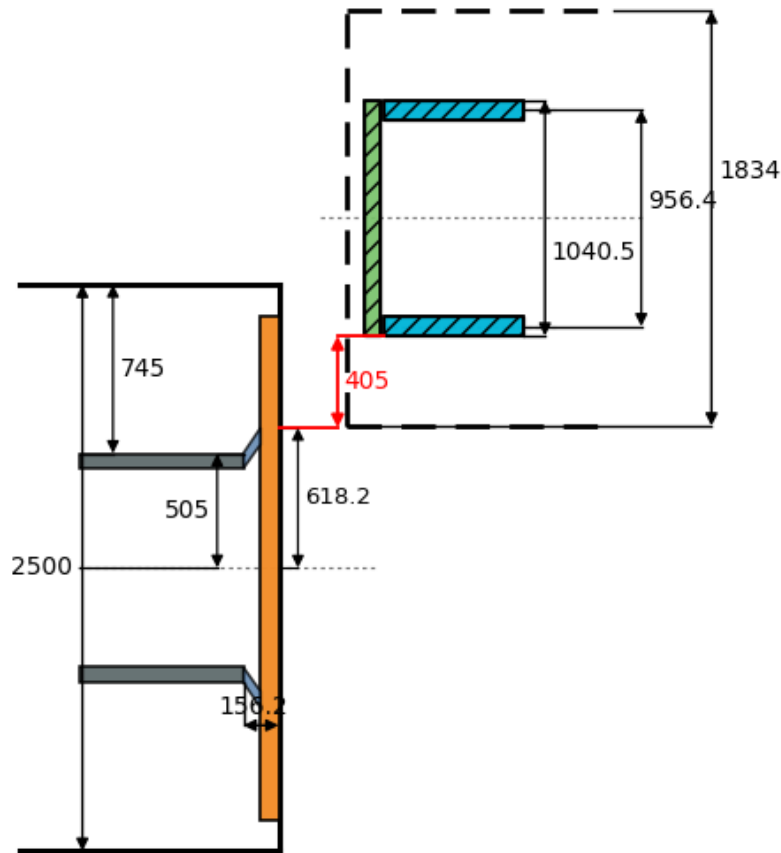


Figure 42: Average front-view overlay at approximately 34% overlap, showing a case where the PC main rail contacts the FUP with a large distance to the nearest FUP bracket.

Offset: 34% (623.6 mm overlap)



— HGV outer width	— FUP	— Truck frame rails	— Car bumper crossbeam
— Car outer width	— FUP brackets	— Car main rails	

Figure 43: Simplified top-view schematic at approximately 34% overlap, showing the moment arm between the main rail impact location and the nearest FUP bracket.

Potentially critical 29% overlap.

Another potentially critical case was identified at approximately 29% overlap. In this configuration, the main PC load paths may miss the FUP, despite a substantial overlap between the outer vehicle structures. The upper and lower PC rails are positioned close to, but not clearly engaged with, the outer FUP region. This indicates a possible transition zone between missed load-path engagement and localised outer-FUP loading.

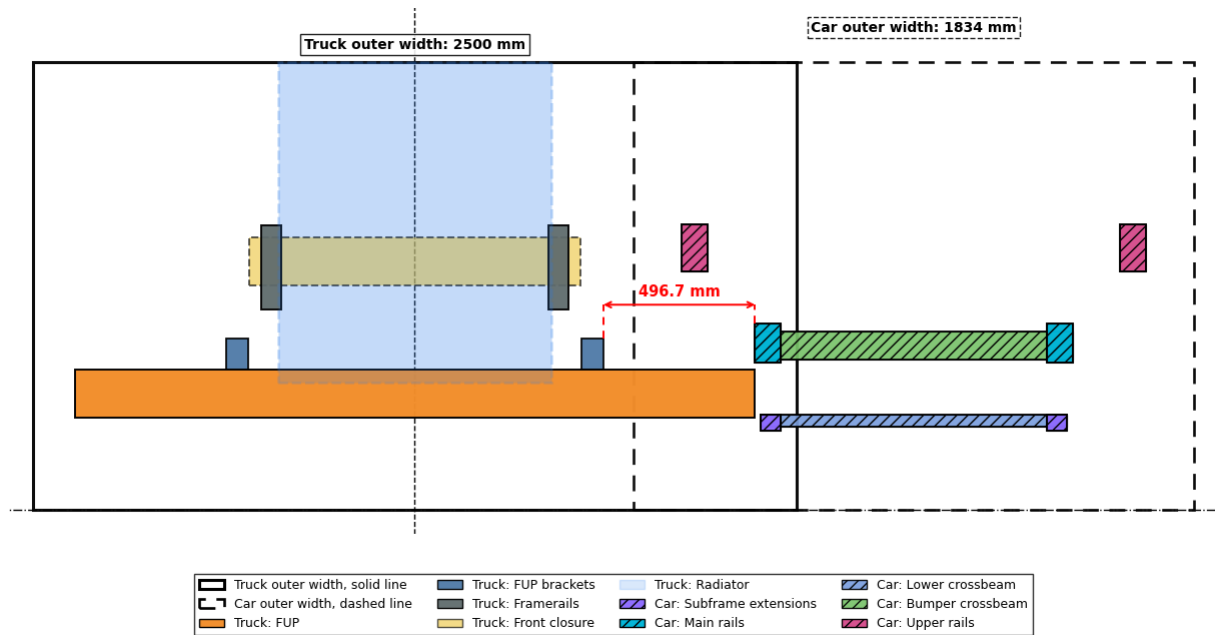


Figure 44: Average front-view overlay at approximately 29% overlap, showing a case where the main PC load paths may miss the FUP despite substantial vehicle overlap.

Offset: 29% (531.9 mm overlap)

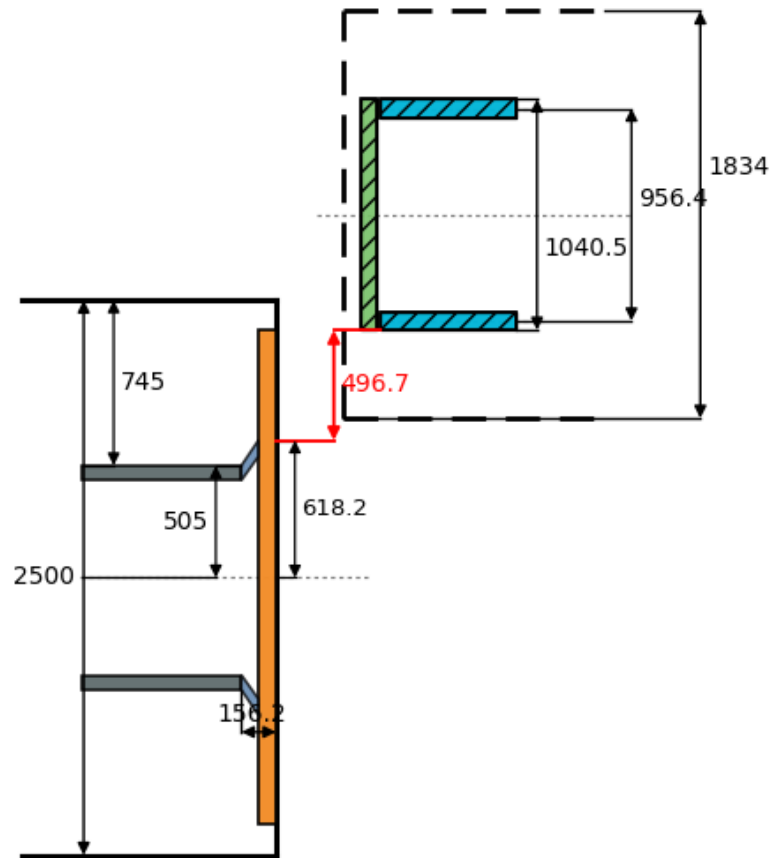


Figure 45: Simplified top-view schematic at approximately 29% overlap, showing the lateral position of the PC main rails relative to the FUP and FUP brackets.

Together, the 29% and 34% cases show that relatively small changes in overlap can shift the interaction from missed engagement of the main PC load paths to highly localised loading of the outer FUP region. This makes the outer FUP geometry and bracket support layout particularly important in small-overlap configurations.

4.4.5 Side View: HGV and PC

Figure 46 presents a side-view overlay of the average truck front structure and the average PC front structure. The figure is intended to visualize the geometric relationship between the main truck structures and the PC load paths when the vehicles are positioned facing

each other. The purpose of the side-view overlay is to provide a more intuitive understanding of the vertical and longitudinal alignment between the components than can be obtained from tabulated measurement values alone.

The PC components are represented using average vertical positions and average longitudinal dimensions, except for the firewall, where the longitudinal thickness is approximated for visualization.

For the truck, the database provides vertical component positions, whereas longitudinal component depths are generally not available, except for the FUP. Consequently, the truck-side components should primarily be interpreted with respect to their height above the ground and their longitudinal placement, rather than as exact representations of component depth. The vehicles in the background are not drawn to an exact scale and should only be considered an approximate visual reference.

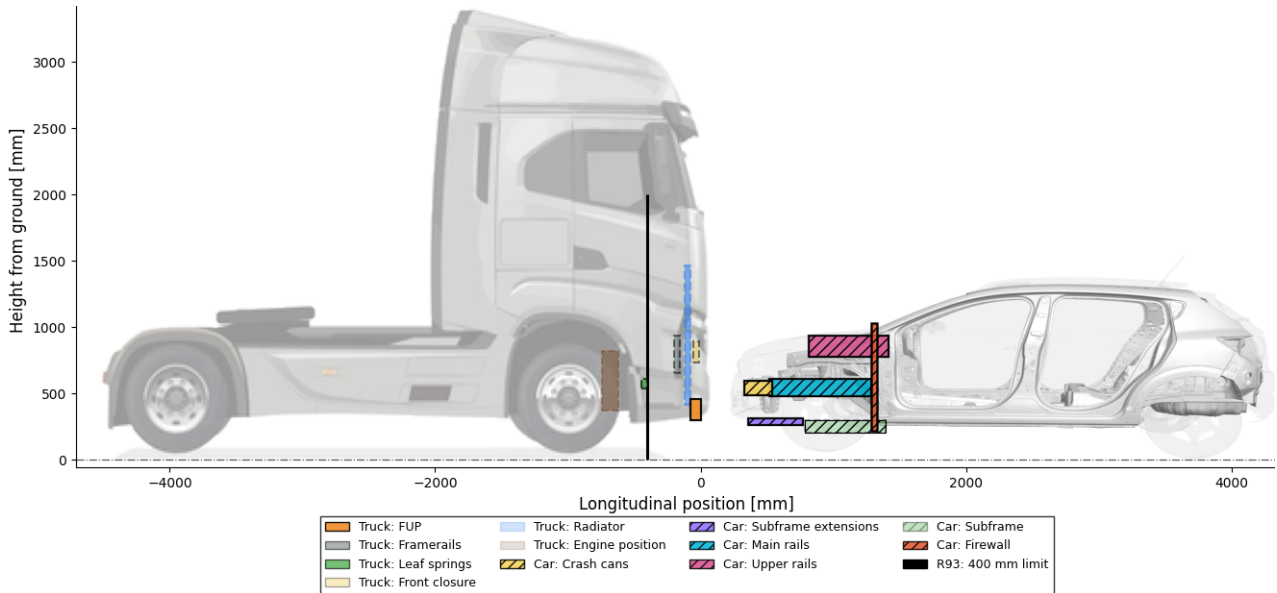


Figure 46: Side-view overlay of the average truck front structure and the average PC front structure, illustrating the approximate vertical and longitudinal relationship between selected components.

4.4.6 PCs engine VS. HGVs framerrail vertical overlap

The analysis investigating whether the current HGV fleet overlaps PC engines in a vertical comparison showed, as illustrated in figure 47, that only one out of the 55 investigated PCs had an engine placement that did not interact with any of the investigated HGV frame rails. The remaining 54 PCs showed vertical overlap with all investigated HGV frame rails.

This indicates that if the truck frame rails penetrate through the frontal load paths of

the PC during a collision, interaction between the PC engine and the truck frame rails is likely to occur.

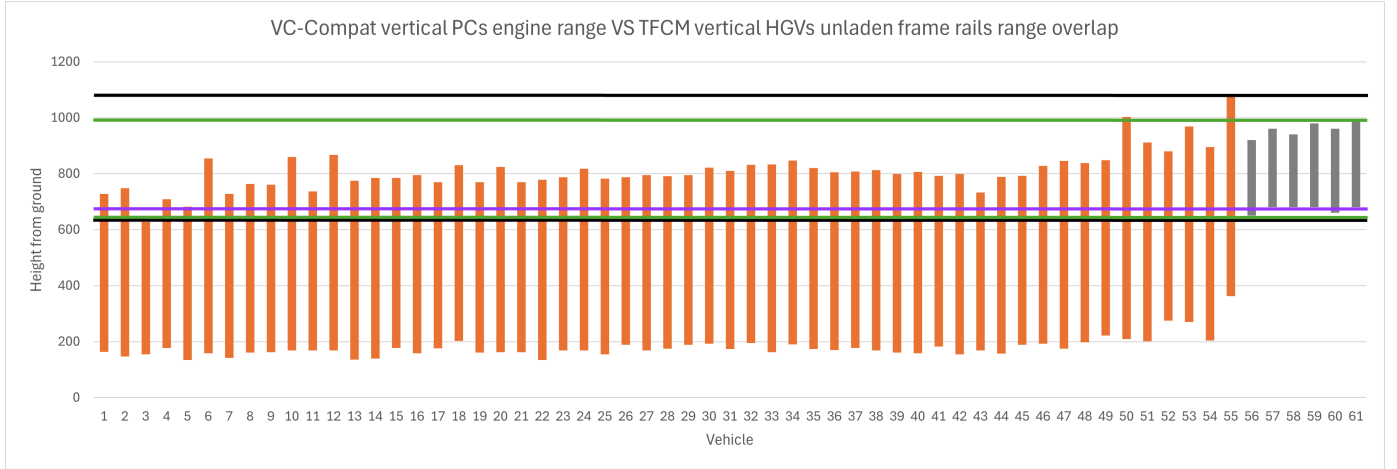


Figure 47: The VC-Compat PCs engine VS. TFCM HGVs Frame rails vertical range overlap.

4.5 Validation of Car Energy Absorption

The crash energy absorbed by the PC can be approximated by Equation 10, assuming a linear force–deflection relationship. In reality, the force–deflection curve during a crash is usually nonlinear and may contain several peaks and variations over the crush distance. Therefore, a more accurate estimate of the absorbed energy requires integration of the actual force–deflection curve, as shown in Equation 9.

To evaluate whether the simplified triangular approximation is feasible for this thesis, force–deflection curves from U.S. NCAP frontal crash tests were used. Wiacek et al. (2015) presented average force–deflection curves for different vehicle classes based on model year 2002–2014 NCAP test data. In this thesis, the curves for 2011–2014 PCs and Multi-Purpose Vehicles (MPVs) were used as reference curves.

The absorbed energy obtained from the triangular approximation (Equation 10) was compared with values estimated by trapezoidal integration of the published force–deflection curves. Since the force and deformation values were manually extracted from published graphs, the results should be regarded as approximate.

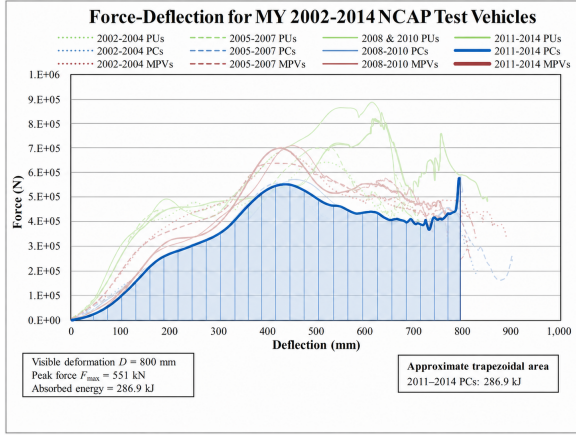


Figure 48: Trapezoidal integration of the average force–deflection curve for 2011–2014 PCs, adapted from Wiacek et al. (2015).

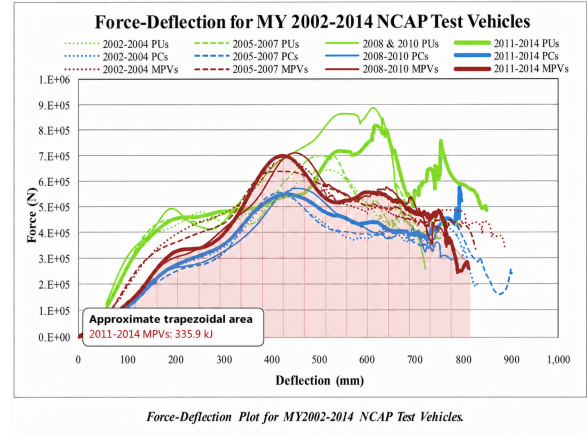


Figure 49: Trapezoidal integration of the average force–deflection curve for 2011–2014 MPVs, adapted from Wiacek et al. (2015).

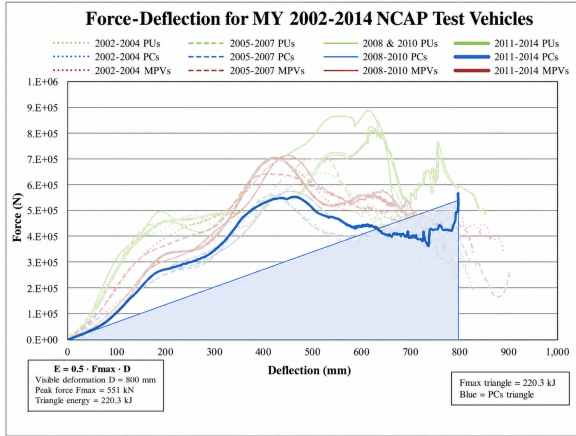


Figure 50: Simplified $F_{\max}$ triangular approximation of absorbed crash energy for 2011–2014 PCs, adapted from Wiacek et al. (2015).

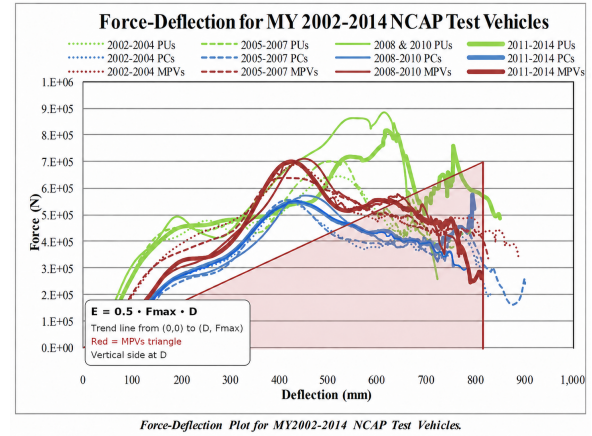


Figure 51: Simplified $F_{\max}$ triangular approximation of absorbed crash energy for 2011–2014 MPVs, adapted from Wiacek et al. (2015).

The triangular approximation underestimated absorbed energy by approximately 23.2% for PCs and 14.7% for MPVs. Although simplified, the method provides a reasonable estimate for preliminary crash-energy calculations and comparative analyses.

Table 7: Comparison between trapezoidally integrated crash energy and the simplified $F_{\max}$ triangular approximation based on Wiacek et al. (2015).

Category	D [mm]	$F_{\max}$ [kN]	Trapezoidal area [kJ]	$F_{\max}$ triangle [kJ]	$F_{\max}$ vs trapezoidal [%]
2011–2014 PCs	800	551	286.9	220.3	-23.2
2011–2014 MPVs	815	703	335.9	286.5	-14.7

5 Discussion

This section discusses the previously presented results and their implications for frontal crash compatibility between HGVs and PCs.

5.1 Discussion of Results

Krusper and Thomson (2012) showed that a taller FUP cross-member can improve structural interaction and accommodate vertical variation between HGVs and PCs. This supports the average-model comparison, which indicates that a relatively tall FUP front face may be required to engage multiple PC load paths simultaneously.

However, increasing FUP height introduces challenges related to stiffness, load-carrying capacity, and weight. Existing FUP designs demonstrate that increased structural coverage does not necessarily require a solid cross-member. For example, a Volvo Truck Corporation patent describes a FUPD beam consisting of a hollow housing structure with internal channels and reinforcement members (Volvo Truck Corporation, 2021). This supports the use of profile-based or multi-member concepts to achieve increased coverage while maintaining structural efficiency.

However, the spacing between the beams would be critical. If the gaps are too large, the structure may not provide a continuous contact surface for the PC load paths and could instead create local intrusion points. If the spacing is too small, the structure may become unnecessarily heavy and difficult to package.

Based on this principle, one possible design direction is a grid-like FUP consisting of several hollow horizontal members stacked vertically. Such a concept could increase the effective contact area while remaining relatively weight-efficient. However, beam spacing, support layout, and structural stiffness would require optimization to ensure effective load transfer without creating local intrusion points. The concept should therefore be regarded as a potential design direction requiring further simulation and physical validation.

An additional aspect not investigated in detail in this study is the potential influence of front axle displacement during frontal impacts. Herbst et al. (2023) show, based on heavy-truck crash cases and testing, that rearward displacement of the front axle or wheel can damage nearby fuel tanks and contribute to post-crash fires. The authors also demonstrate that FUP systems can reduce axle and wheel displacement, thereby decreasing the likelihood of fuel tank interaction.

Although the fuel-tank configurations investigated by Herbst et al. (2023) may differ from those of the European N3 vehicles considered in this study, the findings highlight that the benefits of a FUP may extend beyond underrun prevention and load-path compatibility. In vehicle layouts where fuel tanks or other critical components are positioned close behind the front axle, limiting axle displacement may provide an additional safety benefit.



Figure 52: Example of fuel tank placement behind the front axle region, illustrating a possible post-crash vulnerability if the front wheel or axle is displaced rearwards, reproduced from *Volvo FH gas-powered*, by Volvo Trucks, n.d.

This aspect was outside the scope of the present work but could be considered in future evaluations of FUP concepts.

The overlap analysis indicates that the outer region of the FUP is critical in small-overlap configurations. The cases around 29% and 34% represent the transition between missed load-path engagement and localised loading of the outer FUP region. Therefore, FUP performance should not only be evaluated based on total width, but also on how the outer sections are shaped, supported, and integrated with the surrounding truck-front structure. Since the exact angle and transition position cannot be determined from the geometric analysis alone, the following discussion should be interpreted as design reasoning rather than a final geometry recommendation.

At approximately 29% overlap, the results indicate that increased lateral FUP coverage could reduce the risk of missed engagement. However, simply widening a straight FUP may not be optimal, since early engagement of a straight outer edge could reduce the possibility of glance-off at very small overlaps. The overlap analysis indicates that the outer region of the FUP is critical in small-overlap configurations. The cases around 29% and 34% are especially relevant, since they represent the transition from missed load-path engagement to highly localised loading of the outer FUP region.

At approximately 34% overlap, the results show a highly localised FUP contact region.

The PC load paths begin to interact with the outer FUP, but the contact occurs far from the nearest FUP bracket. In the average model, this corresponds to a bending moment arm of approximately 405 mm, which may increase bending loads in the cross-member. At the same time, the outer width of the PC remains outside the truck frame rail region. This means that the impact configuration combines localised FUP loading with limited alignment to the main truck support structure. Additional outer support could therefore be beneficial, since it may help distribute the load into the truck structure and reduce local bending of the FUP cross-member.

From a design perspective, this suggests that the region around this type of overlap may be more suitable for controlled redirection than for direct structural engagement. If the FUP remains straight in this region, the interaction may create concentrated loading in the outer cross-member rather than efficient load transfer into the truck structure. A curved or angled outer FUP section could therefore be considered around the region where the PC outer width begins to move outside the main frame rail support region. Such a transition could help guide the PC away from the truck front instead of forcing a highly localised structural engagement.

However, this should not be interpreted as defining an exact starting point for the curvature. The optimal transition would depend on the individual truck packaging and on the glance-off requirements discussed earlier. The result should therefore be seen as an indication that the outer FUP region is important for small-overlap design, rather than as a fixed design rule.

The longitudinal position of the shaped outer section is also important. If the outer FUP geometry is angled or curved too far rearwards, the PC may contact other truck-front components before the intended FUP engagement occurs. In that case, the benefit of the shaped FUP could be reduced, since the crash interaction may already have been influenced by less favorable structures. Therefore, the outer FUP geometry should be considered together with the longitudinal position of surrounding truck-front components, not only with the lateral overlap.

The transition between the straight central FUP section and the angled outer section is also important. If the PC load paths first contact the FUP close to a sharp transition, local force concentration may occur. Therefore, if the same logic as discussed by Knight regarding protruding stiff edges is applied, a gradual curved transition may be preferable to a sharp corner. A curved transition could reduce local load concentration and lower the risk that a protruding edge creates local intrusion into the PC.

A more advanced concept could be to combine an angled FUP with an extended and shaped front structure. The main benefit of such a concept is the increased available

deformation distance on the truck side. Instead of relying only on displacement behind the FUP, an extended front could provide controlled deformation before the impact reaches the conventional truck-front region. This could allow the truck to absorb a larger share of the crash energy and reduce the energy that must be absorbed by the PC. The length of this extension should be balanced with the maneuverability requirements.

Using the constant-force energy approximation from Equation 11, the absorbed truck-side energy can be estimated for the extended-front concept. If the extended front with FUP provides approximately 400 mm of controlled deformation and the FUP system then allows a further 360 mm of controlled displacement under the truck, the total available deformation distance becomes approximately 760 mm. Assuming a simplified average energy-absorbing force of 300 kN, based on Knight (2016), the absorbed energy becomes

$$E_t = 300 \text{ kN} \cdot 760 \text{ mm} = 228 \text{ kJ} \quad (12)$$

For comparison, a more conventional FUP concept with only 360 mm of controlled displacement at the same force level would absorb

$$E_t = 300 \text{ kN} \cdot 360 \text{ mm} = 108 \text{ kJ} \quad (13)$$

The assumed 300 kN should be interpreted as a simplified average longitudinal force level for the engaged energy-absorbing FUP system, not as a point load or surface pressure. In a real crash, the load distribution would depend on overlap, contact area, and the number of engaged PC load paths. Therefore, this force may be distributed over a larger part of the FUP in high-overlap impacts, while in small-overlap impacts it may be introduced more locally into the FUP cross-member.

To relate the estimated FUP energy absorption to the collision energy available for structural deformation, Equation 7 was used with a passenger car mass of $m_p = 1600 \text{ kg}$ and a truck mass of $m_t = 12000 \text{ kg}$. Two simplified head-on cases were evaluated, where both vehicles travel towards each other at either 60 km/h or 80 km/h. This gives closing speeds of 120 km/h and 160 km/h, respectively.

For the 60 + 60 km/h case, the closing speed becomes $120 \text{ km/h} = 33.33 \text{ m/s}$:

$$E_{d,60} = \frac{1}{2} \frac{1600 \cdot 12000}{1600 + 12000} (33.33)^2 \approx 784 \text{ kJ} \quad (14)$$

For the 80 + 80 km/h case, the closing speed becomes $160 \text{ km/h} = 44.44 \text{ m/s}$:

$$E_{d,80} = \frac{1}{2} \frac{1600 \cdot 12000}{1600 + 12000} (44.44)^2 \approx 1394 \text{ kJ} \quad (15)$$

The PC-side absorption was represented by the trapezoidal integration of the Wiacek et al. (2015) average force–deflection curves, giving 286.9 kJ for 2011–2014 PCs and 335.9 kJ for 2011–2014 MPVs. These values were combined with the estimated truck-side absorption of 108 kJ for the 360 mm FUP concept and 228 kJ for the 760 mm extended-front concept. The resulting combined structural absorption is compared with the simplified dissipated collision energy in Table 8.

Table 8: Combined PC and truck-side structural absorption compared with dissipated collision energy.

	PC + 360 mm FUP concept	MPV + 360 mm FUP concept	PC + 760 mm extended-front	MPV + 760 mm extended-front
E_p [kJ]	286.9	335.9	286.9	335.9
E_t [kJ]	108	108	228	228
$E_p + E_t$ [kJ]	394.9	443.9	514.9	563.9
Share of $E_{d,60}$ [%]	50.3	56.6	65.6	71.9
Share of $E_{d,80}$ [%]	28.3	31.8	36.9	40.4

The comparison shows that the combined PC and truck-side structural absorption would correspond to approximately 50–57% of the simplified dissipated collision energy in the 60 + 60 km/h case for the 360 mm FUP concept, and approximately 66–72% for the 760 mm extended-front concept. For the 80 + 80 km/h case, the corresponding values decrease to approximately 28–32% and 37–40%, respectively. This illustrates that the share of the dissipated collision energy covered by the combined PC and HGV structural absorption depends strongly on impact speed. The values should be interpreted as a simplified energy comparison rather than a prediction of the actual energy distribution in a specific crash configuration.



Figure 53: Conceptual extended-front FUP layout, adapted from Danish Transport Authority (2011). Truck shown in the Danish Transport Authority proposal adapted with AI. The original Scania concept used a 600 mm extended front end, while this modified visualization illustrates a 400 mm extended front and the proposed FUP layout used in this thesis. The dimensions are conceptual and not fully scale-accurate.

An important question is how such a concept could be combined with the current UNECE R93 quasi-static test requirements. One possible interpretation is that the FUP could be designed to remain sufficiently stiff under the R93 approval loads, but begin controlled deformation at higher real-world crash loads. This is possible in principle because the highest R93 test load is 160 kN, while Krusper and Thomson (2012) suggested an optimal trigger force of approximately 200 kN. This means that the FUP could theoretically pass the quasi-static test with limited deformation, but still be designed to trigger and deform under higher crash forces.

This interpretation is also supported by the difference between regulatory test loads and crash loads discussed in the literature. Krusper and Thomson (2012) reported substantially higher force levels in crash simulations, for example a Ford Taurus generating approximately 650 kN. This indicates that the quasi-static R93 load levels may be considerably lower than the forces acting on the FUP in a real crash. Therefore, a FUP could potentially be tuned with a trigger force above the R93 test level, allowing it to remain compliant during approval testing while still providing controlled energy absorption in a real-world impact.

Krusper and Thomson (2012) also suggested that approximately 360 mm of deformation

under the truck is optimal, because then the displacement is stopped before the PC underruns into the truck cabin region. This means that the concept would need a controlled displacement stop. The FUP should not simply deform freely, but should absorb energy over a defined distance and then prevent further underrun.

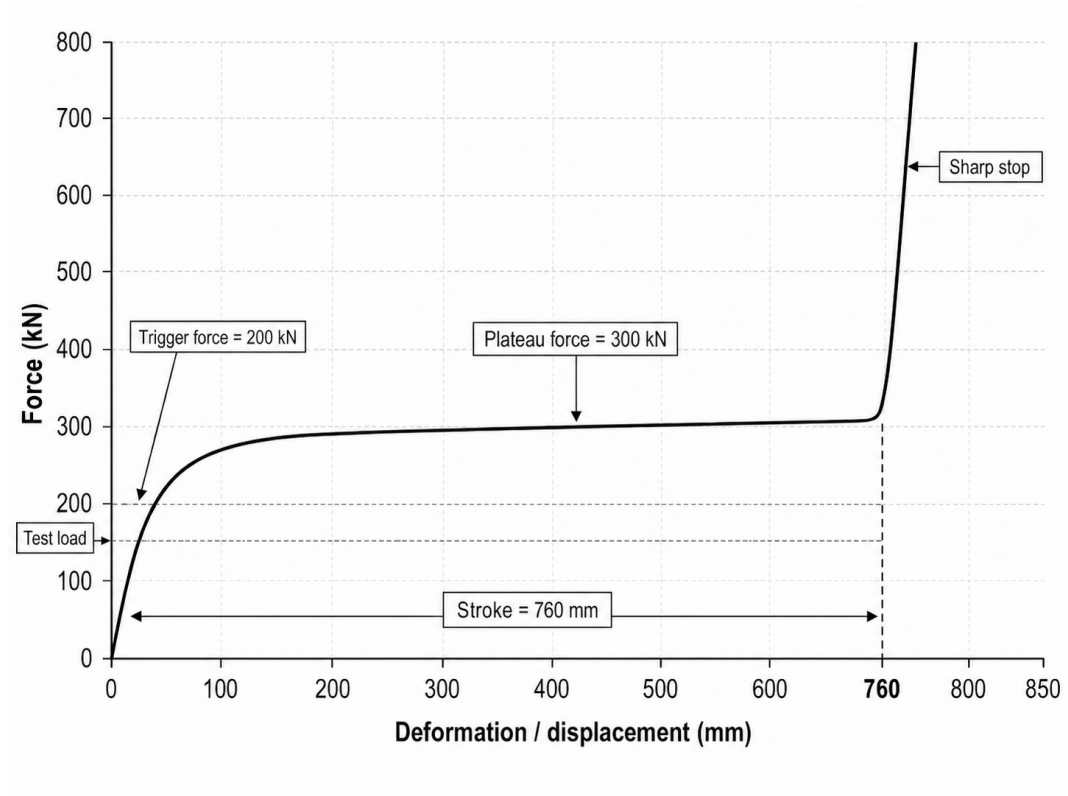


Figure 54: Conceptual force–displacement curve for an energy-absorbing FUP system, illustrating the trigger force, plateau force, controlled deformation distance, and final displacement stop.

By combining this principle with an extended front, a larger total deformation distance than 400 mm could theoretically still be compatible with the regulation, if the FUP position after the quasi-static test remains within the allowed limit. In this concept, the extended front with the FUP provides additional controlled crush distance before the FUP reaches the conventional truck-front region, while the FUP itself is tuned to pass the quasi-static test and then deform under higher crash loads. This would need careful design of the trigger force, plateau force, support stiffness, and displacement stop, and would have to be verified through simulation and physical testing.

5.2 Reliability and Validity

The TFCM database is based on manually collected measurements. Consequently, some uncertainty is associated with component identification and measurement accuracy. The results should therefore be interpreted as approximate geometric measurements intended for compatibility assessment rather than exact component definitions.

An additional limitation concerns the truck-width definition used in the overlap analysis. While PC width was based on measured body width, truck width was approximated from the measured tire-to-tire distance. The average measured tire-to-tire width was 2416 mm, while the truck body is typically somewhat wider than the outer tire width. Since the full body width was not measured, an average truck width of 2500 mm was adopted for the overlap calculations. This approximation affects the calculated overlap percentages and the lateral positioning of the PC relative to the truck front, and should therefore be interpreted as an approximate rather than vehicle-specific width definition.

The limited database size represents another source of uncertainty. Although several major manufacturers were included, the TFCM database represents only a subset of modern N3 vehicles. Variations in suspension configuration, frame layout, ride height, axle configuration, and customer-specific options mean that the results should be interpreted as indicative geometric trends rather than population-representative values.

To improve comparability, vehicles manufactured before 2020, N2 vehicles, and most construction trucks were excluded. While this improved dataset consistency, it also limits the generalizability of the results. Consequently, the conclusions should primarily be regarded as representative of modern N3 vehicles.

The simplified average model was further compared with the full-scale HGV–PC crash test presented by Thomson et al. (2023). A simplified top-view representation was created to assess the relative positioning of the FUP, frame rails, FUP brackets, and PC load paths. Although the comparison does not constitute a formal validation, the geometric relationships observed in the test vehicle corresponded reasonably well with the average model, providing a qualitative plausibility check of the compatibility analysis.

Figure 39 shows the 50% overlap scenario based on the average truck and PC models in top view, while Figure 55 shows the corresponding configuration from the full-scale HGV–PC crash test. The Volvo FH used in the test corresponds reasonably well with the average model in terms of overall width, FUP position, frame rail position, and the lateral relationship between the PC and truck front. This indicates that the average model provides a realistic representation of the general geometric compatibility problem.

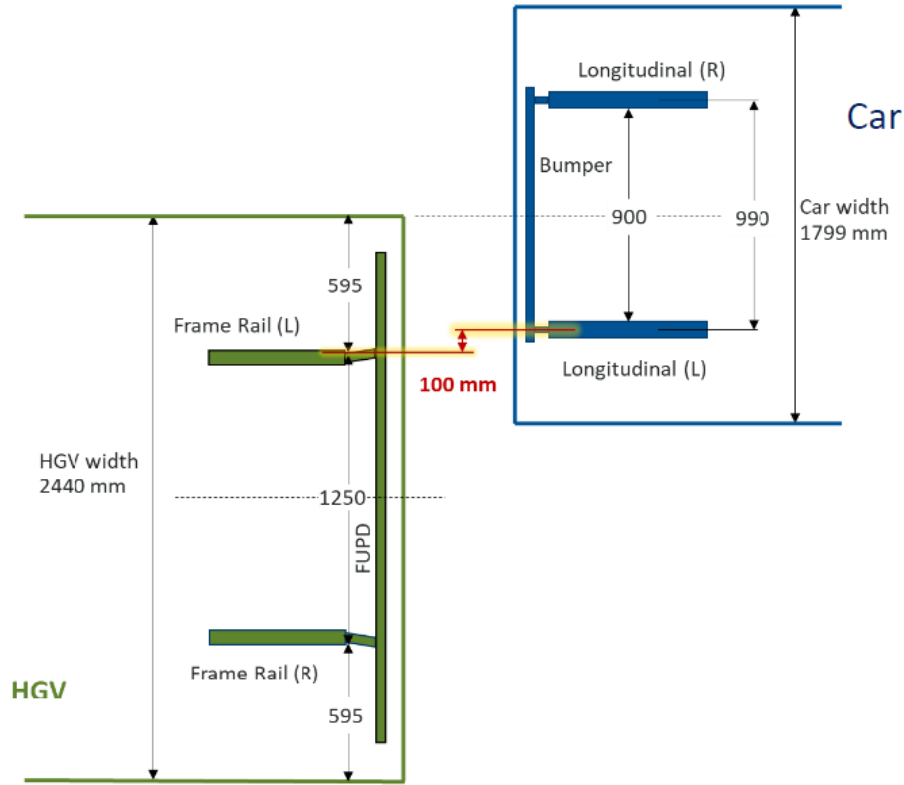


Figure 55: Comparison case showing the overlap geometry from the car-to-HGV crash test, adapted from Thomson et al. (2023).

One limitation of the average models is that they cannot fully represent the variation between individual vehicle models and segments. Furthermore, the PC model is based on an unweighted dataset, meaning that all vehicles influence the average equally regardless of their prevalence in the European vehicle fleet.

The average component positions may also be influenced by outliers and segment-specific differences in structural layout. Consequently, the average models should not be interpreted as representations of specific vehicles, but rather as simplified reference models intended to identify general compatibility trends.

The same limitation applies to the truck model. Due to the limited size of the TFCM database, individual vehicles may have a noticeable influence on the calculated average positions. The average truck and PC models should therefore be regarded as visual and analytical tools for evaluating general alignment trends between PC load paths and truck-front structures rather than exact representations of real vehicle configurations.

5.3 Project Limitations

During the project, several additional limitations were identified. The availability of suitable HGVs was limited due to restricted public documentation and practical difficulties in accessing certain components for measurement.

The use of simplified manual measurement methods introduces uncertainties related to measurement precision and component identification. Furthermore, the analysis primarily considered outer geometric boundaries rather than complete internal structural layouts, which may influence the accuracy of the compatibility assessment.

Unlike PCs, HGVs are highly modular and frequently adapted to specific applications. As a result, vehicles sharing the same model designation may still differ substantially in structural layout, ride height, and component positioning. This variability complicates both fleet representation and the development of standardized FUP concepts. As discussed in Section 3.3.1, braking-induced ride-height changes may further influence vertical load-path alignment.

Finally, validated simulations and physical crash testing were outside the scope of the project. The presented conclusions should therefore be interpreted as analytical observations and geometrical compatibility indications derived from the available dataset.

6 Conclusion

The compatibility analysis identified several factors considered important for improving frontal crash compatibility between HGVs and PCs. Based on the collected geometric data, literature review, and analytical observations, a set of conceptual FUP design guidelines was developed. The following recommendations should be interpreted as design directions derived from the findings of this study rather than as complete design requirements.

6.1 Conceptual design recommendations

The following guidelines should be interpreted as conceptual design directions based on the geometric compatibility analysis, rather than as complete design requirements.

Based on the compatibility analysis, a straight FUP cross-member appears beneficial in higher-overlap impacts, as it provides a larger and more uniform contact surface for force transfer. However, in small-overlap impacts, a purely straight FUP may be less effective in promoting lateral vehicle deflection (*glance-off*). A balanced design approach could therefore consist of a straight central section combined with angled or curved outer sections. Such a configuration may provide stable load transfer in higher-overlap collisions while supporting PC redirection in lower-overlap scenarios.

A low FUP ground clearance combined with sufficient vertical height may improve engagement with multiple PC load paths, including the subframe extensions, main rails, and potentially the subframe. Based on the average values used in this study, the highest average main rail position is approximately 611 mm above ground and the lowest average subframe extension position approximately 260 mm above ground, indicating that a FUP front-face height of approximately 351 mm would be required to cover this range. If the outlier values are considered, the required vertical coverage increases to approximately 568 mm, based on the lowest subframe extension position of 204 mm and the highest main rail position of 772 mm. However, increasing FUP height introduces additional design challenges, as the structure must maintain sufficient stiffness and load-carrying capacity without becoming excessively heavy. A FUP front-face height of approximately 351 mm is therefore recommended based on the average-model analysis, see figure 56.

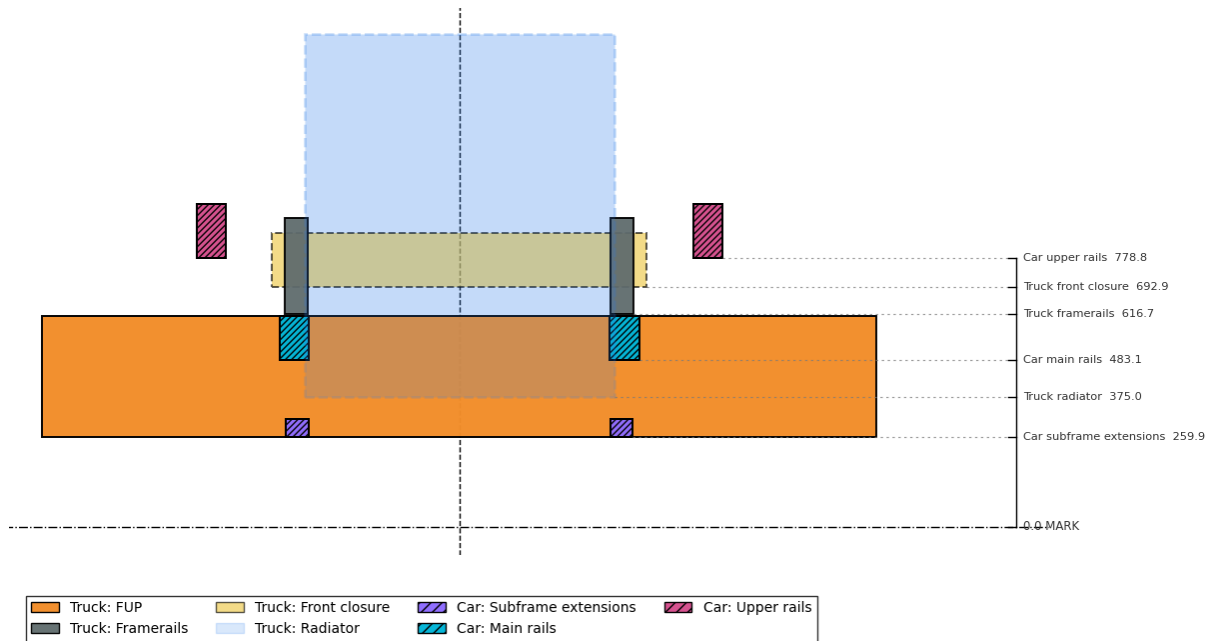


Figure 56: Simplified average-model visualization of the proposed FUP vertical coverage in relation to PC load paths and selected truck-front structures. The figure illustrates how a taller FUP front face could engage the subframe extensions and main rails, while also showing the higher upper rail load path and truck-front structures.

For an extended-front concept, greater vertical coverage may be required to engage the upper rail load path, as the highest average upper rail position was approximately 934 mm above ground. This value may therefore serve as an indicative guideline for the height required to include the major passenger-car load paths. However, increasing the structure height also introduces a potential compatibility risk, since structures extending above the upper rail load-path region may interact with the PC A-pillar region. This is particularly relevant since the lowest maximum upper rail position observed in the dataset was approximately 815 mm above ground. Any extension above the upper rail load-path region would therefore require careful control of deformation and displacement to avoid creating new intrusion mechanisms.

A key design objective is to maintain a robust contact surface between the PC load paths and the truck front structure. A stiff FUP cross-member may therefore be advantageous, as it is less susceptible to local deformation or failure at critical impact locations. Controlled energy absorption could instead be achieved through deformable supports or dedicated crash elements positioned behind the cross-member. Such a configuration may preserve structural engagement while allowing controlled deformation of the support system. A conceptual visualization of this design approach is shown in Figure 53.

For glance-off behaviour, the FUP should provide sufficient stiffness for vehicle redirection, suitable angled or curved geometry, compatible vertical alignment with PC load paths,

and a smooth contact surface that promotes sliding rather than structural locking.

The final conceptual FUP design is visualized in Figures 57–60. The CAD model should be interpreted as a visualization of the proposed design direction, not as a validated final design.

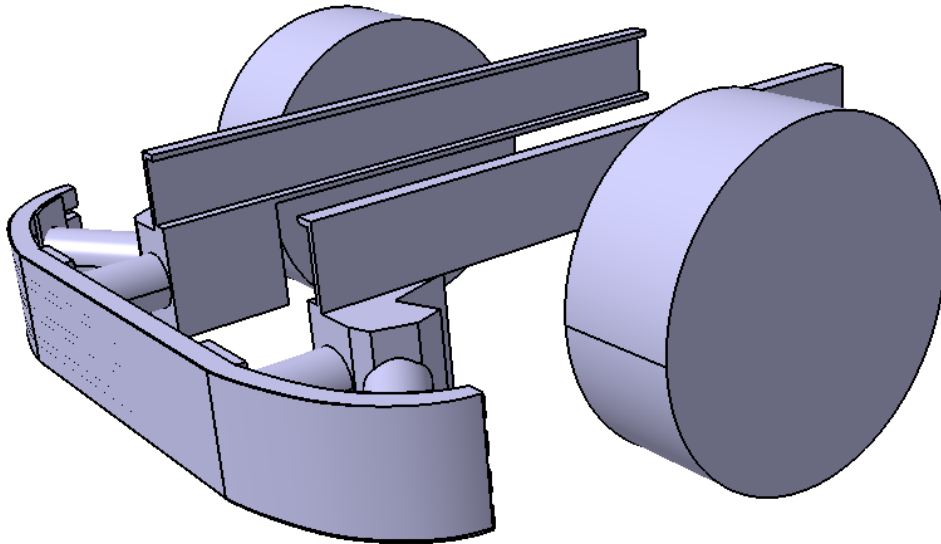


Figure 57: 3D overview of the conceptual FUP design, showing the increased vertical coverage, straight central cross-member, and angled outer sections.

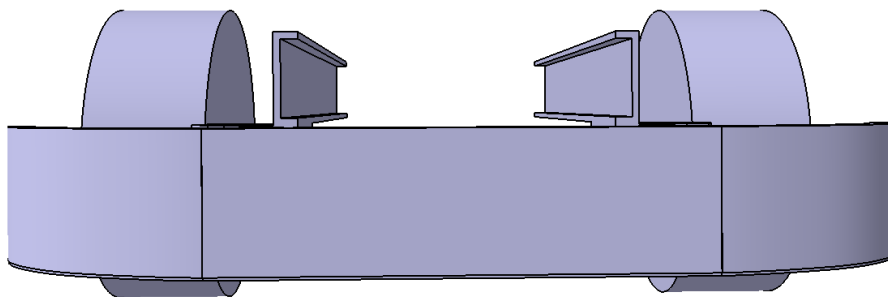


Figure 58: Front view of the conceptual FUP design, illustrating the vertical coverage and beam layout.

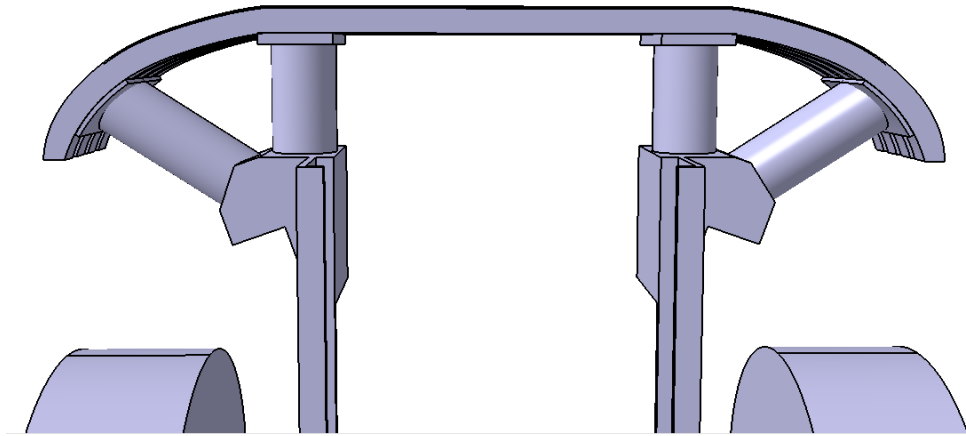


Figure 59: Top view of the conceptual FUP design, illustrating the straight central section and angled outer sections.

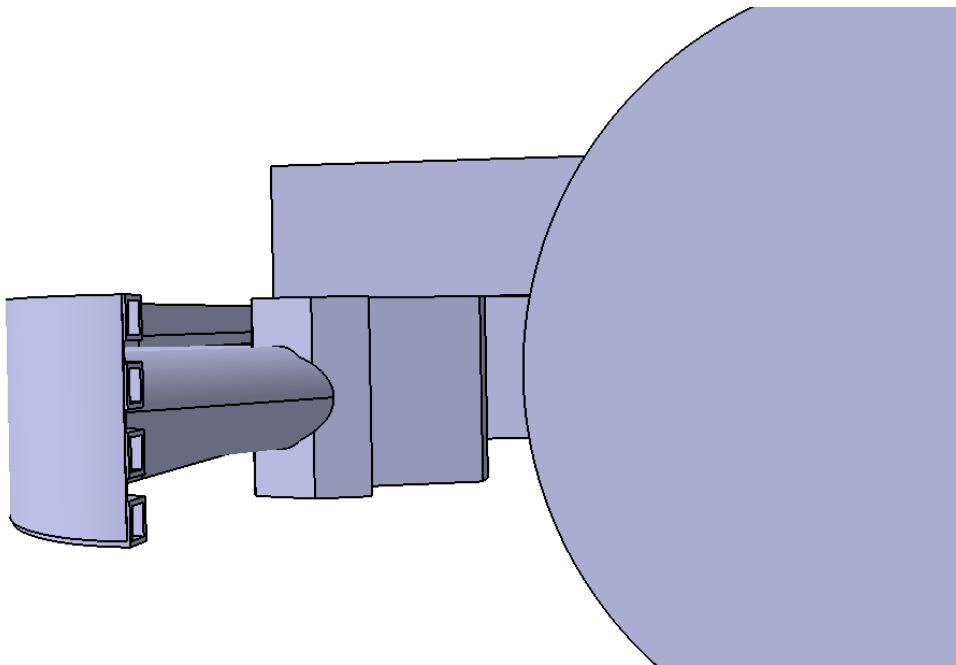


Figure 60: Side view of the conceptual FUP design, illustrating the longitudinal depth of the structure.

An extended-front version of the concept was also developed to visualize how the FUP could be integrated with additional frontal deformation space. This concept should be interpreted as a possible future design direction, where increased deformation length is combined with controlled FUP engagement.

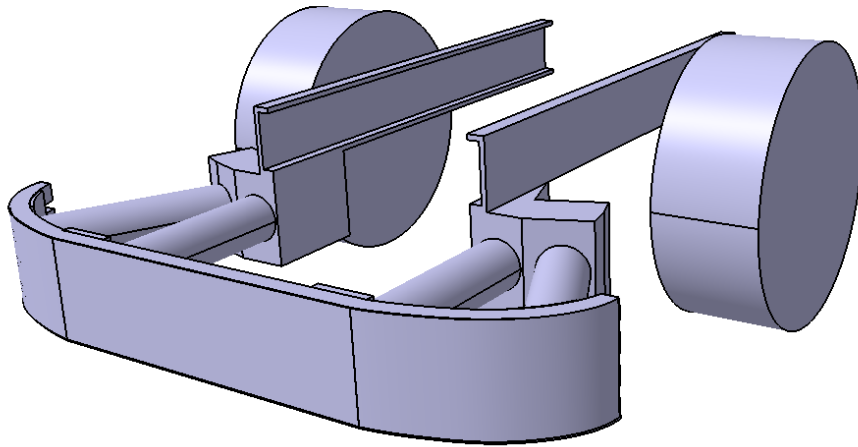


Figure 61: 3D overview of the extended-front FUP concept with 760 mm available displacement, illustrating how the proposed FUP layout could be combined with additional frontal deformation space.

6.2 Future Work

A suitable continuation of this work would be to establish a larger and more representative HGV database in order to improve the understanding of the current vehicle fleet. Based on the identified geometries and collision configurations, a generic FUP concept could then be developed and evaluated through crash simulations covering different impact angles, overlap conditions, and ride-height configurations. Subsequent physical testing would provide an opportunity to validate the simulation results and assess the crash performance of the proposed design.

The current regulations and test procedures are designed in a way that manufacturers may fulfill the requirements without necessarily improving real-world crash compatibility. Future work should therefore also investigate more representative and realistic test procedures capable of better reflecting actual collision conditions between HGVs and PCs.

Since collisions between HGVs and PCs involve high energy levels, continued research regarding crash energy management remains highly relevant. Future studies could therefore further investigate how extended front structures or more energy-absorbing FUP concepts may contribute to improved energy absorption and compatibility performance during frontal impacts.

Additional investigation of the parameters influencing glance-off behavior could also contribute to improved frontal collision outcomes by increasing the likelihood of vehicle deflection rather than full structural engagement and direct energy absorption.

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A Truck measurements – Crash compatibility

Truck measurements – Crash compatibility

A study made for contribution in ISS Trucks Project

A study made by

Herman Jävert & John Müller

in Collaboration with

Chalmers University of Technology



Purpose of Vehicle Dimension Data Request

This study is conducted with the vision of working toward safer road transportation, with a focus on ensuring that no life is lost in accidents involving Heavy Goods Vehicles (HGVs) and passenger cars (PCs). The study will support the development and dimensioning of a car surrogate barrier for the Integrating Safety Systems (ISS) for Trucks project. Access to the requested data would support our ongoing work in advancing the understanding of vehicle compatibility and interaction in frontal crash scenarios. We believe that the resulting insights could contribute to the broader safety ambitions outlined in Euro NCAP's Vision for truck compatibility assessment 2031.

To better understand the compatibility between cars and trucks through the analysis of structural geometry, additional information on HGVs is required. The aim is to analyze truck front structures from trucks across different OEMs to build a general database. By comparing truck and car front structures, the study will highlight compatibility challenges between truck and car structures. This will provide a clearer view of where potential improvements can be made to enhance future safety between HGVs and PCs.

Requested measurements - Specification

Reference Definitions

- **Frame Rails (Longitudinals)** – Main longitudinal chassis beams.
- **Bumper Beam** – Cross beam connecting Frame Rails in the front.
- **Measurements are taken on the left side of the vehicle unless otherwise specified.**
- **If the measured truck uses Air suspension, W6, W8 and H8 can be disregarded.**

The lowest structural point of the FUP is used as a reference point for multiple measurements to get a relative position of the components, not affected by ride height variables.

Longitudinal measurements

L1 – Front axle to foremost point of FUP

Longitudinal distance from the centerline of the front axle to the foremost point of the FUP front face.

L2 – Front axle to foremost point of bumper beam

Longitudinal distance from the centerline of the front axle to the foremost structural surface of the bumper beam.

L3 – Front axle to front of radiator

Longitudinal distance from the centerline of the front axle to the foremost structural point of the radiator assembly.

L4 – Front axle to front of frame rails

Longitudinal distance from the centerline of the front axle to the front end of the chassis frame rails.

L5 – Front axle to front of leaf springs

Longitudinal distance from the centerline of the front axle to the forwardmost point of the front leaf spring assembly.

L6 – Front axle to front of engine

Longitudinal distance from the centerline of the front axle to the forwardmost structural point of the engine assembly.

L7 – Front axle to front of truck

Longitudinal distance from front axle to the most forward point of the truck (Not including mirrors).

Height measurements

H1u – Vertical clearance from ground to FUP

Vertical distance from the ground to the lowest edge of the Front Underrun Protection device (FUPD).
Unladen

H1l – Vertical clearance from ground to FUP – at maximum laden weight

Is needed to be laden, the rest of measurements should be taken unladen.

H2 – Vertical cross section of FUP

Vertical distance between the uppermost edge and the lowest edge of the front face of the FUP.

H3 – Vertical distance from FUP to bumper beam

Vertical distance from the lowest structural point of FUP to the lowest edge of the structural bumper beam.

H4 – Vertical cross section of bumper beam

Vertical distance between the uppermost structural edge and the lowest structural edge of the bumper beam.

H5 – Vertical distance from FUP to radiator

Vertical distance from the lowest structural point of FUP to the lowest structural point of the front radiator assembly.

H6 – Vertical distance from FUP to frame rails

Vertical distance from the lowest structural point of FUP to the lowest structural edge of the left chassis frame rail, measured at the foremost accessible section of the frame rail behind the bumper beam or FUP mounting structure.

H7 – Vertical cross section of frame rail

Vertical distance between the upper and lower structural surfaces of a chassis frame rail, measured at the foremost accessible section of the left chassis frame rail behind the bumper beam or FUP mounting structure.

H8 – Vertical distance between foremost point of leaf suspension to FUP

Vertical distance from the lowest structural point of the FUP to the foremost end of the front leaf spring assembly.

H9 – Vertical distance between FUP and engine

Vertical distance from the lowest structural point of the FUP to the lowest structural point of the engine assembly.

Width measurements

W1 – Inner width of frame rails

Transverse distance between the inner faces of the left and right chassis frame rails.

W2 – Cross section width of frame rail

Transverse distance between the outer face and the inner face of a single chassis frame rail.

W3 – Total width of FUP

Transverse distance between the outermost left and outermost right edges of the FUP.

W4 – Width between inner mounting points

Transverse distance between the inner structural edges of the left and right mounting brackets used to attach the FUP to the vehicle structure.

W5 – Total width of bumper beam

Transverse distance between the outermost left and outermost right edges of the structural bumper beam.

W6 – Distance between inner edges of front leaf springs

Transverse distance between the innermost structural edges of the left and right front leaf spring assemblies.

W7 – Width between outside of front tyres

Transverse distance between the outermost sidewalls of the left and right front tyres, measured at axle center height.

W8 – Width of front leaf spring

Transverse distance between the outer edges of a single front leaf spring, measured at the foremost section of the leaf spring assembly.

W9 – Width of FUP mounting bracket

Transverse distance between the outer and inner structural faces of the FUP mounting bracket/tower.

W10 – Width of radiator

Transverse distance between the outer and inner structural faces of the radiator.

Demonstrating Pictures

All measurements can be recorded directly in the provided Excel file, which contains fields for each H, L, and W parameter.

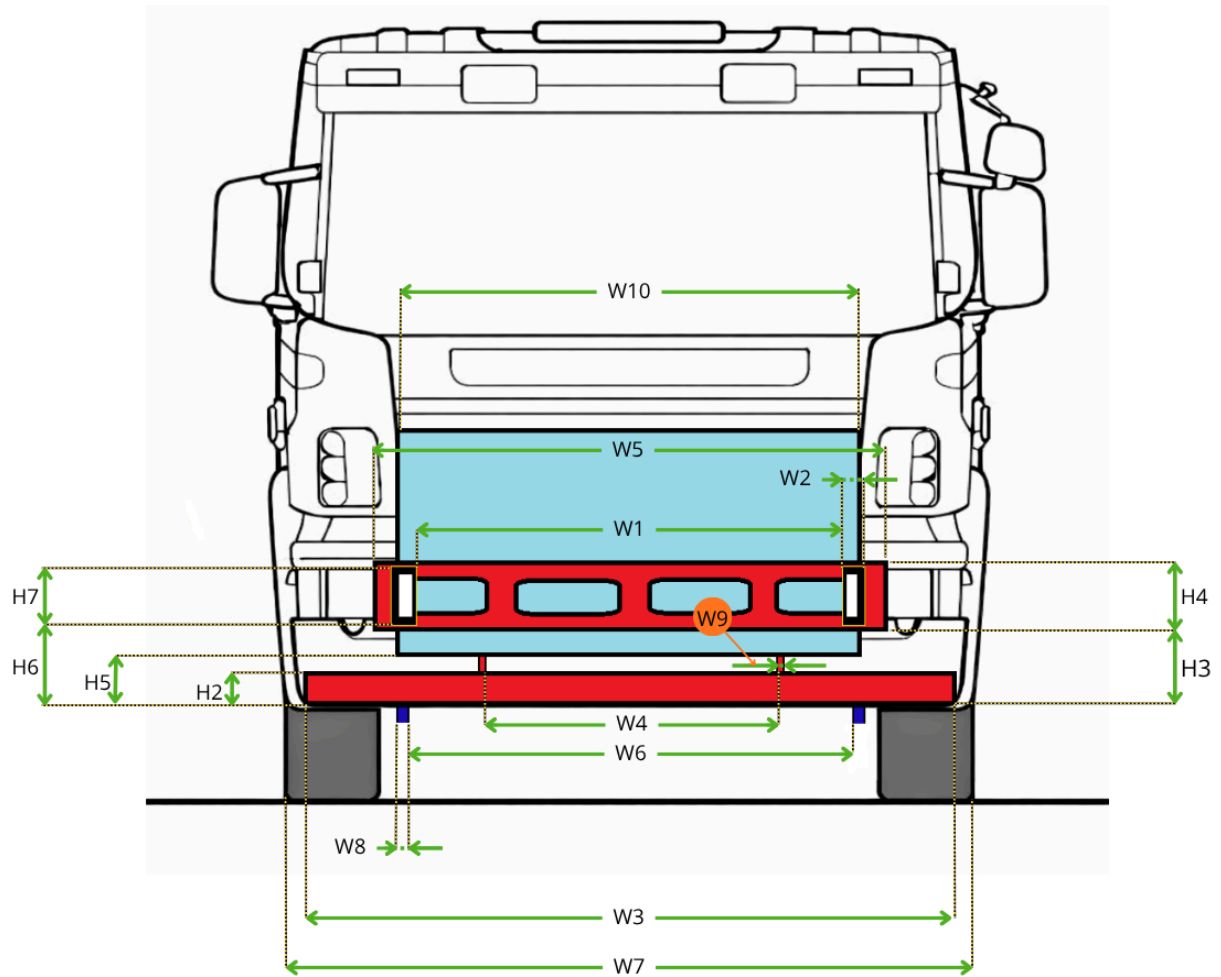


Figure 1: Front view demonstrating requested measurements.

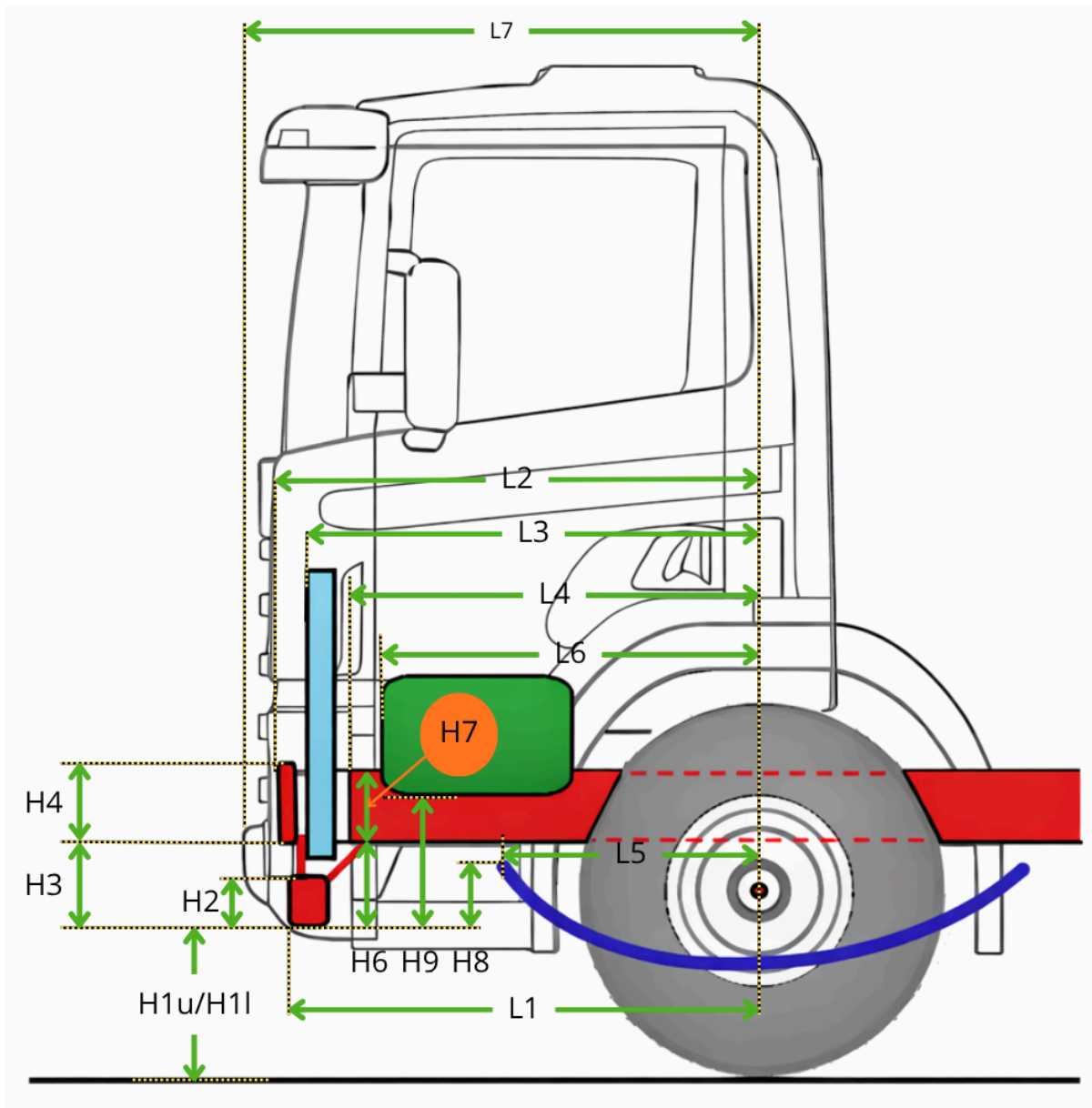


Figure 1: Side view demonstrating requested measurements.

B VC-Compat and Truck Front Components Measurements Comparison

The following figures show the comparison between the VC-Compat database and the Truck Front Components Measurements database. The boxplots show the median, mean, 10th–90th percentile range, and individual vehicle measurements for each comparable parameter.

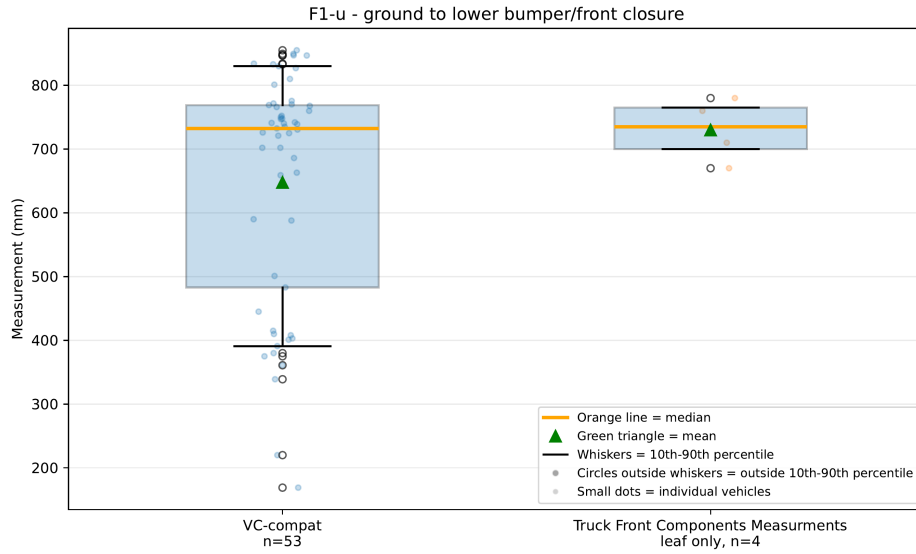


Figure 62: Comparison of F1-u, ground to lower bumper/front closure, between the VC-Compat database and the Truck Front Components Measurements database.

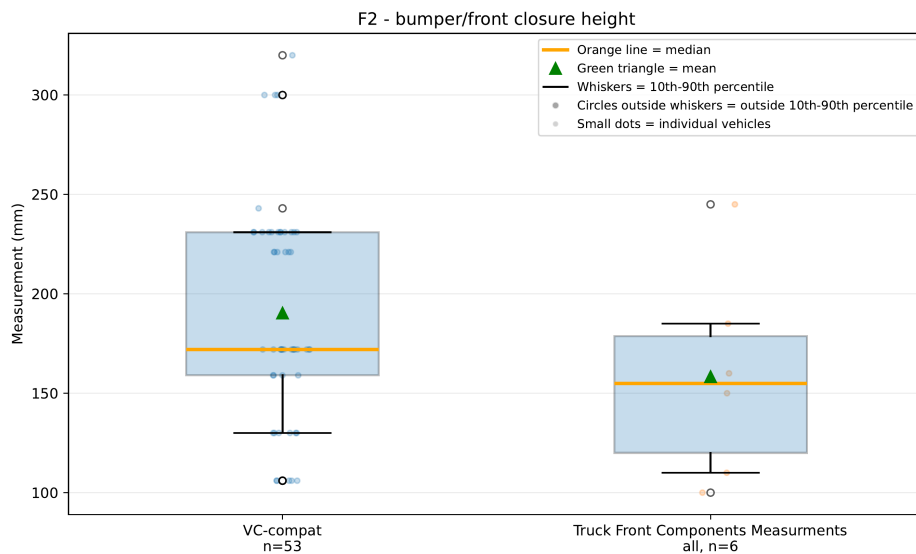


Figure 63: Comparison of F2, bumper/front closure height, between the VC-Compat database and the Truck Front Components Measurements database.

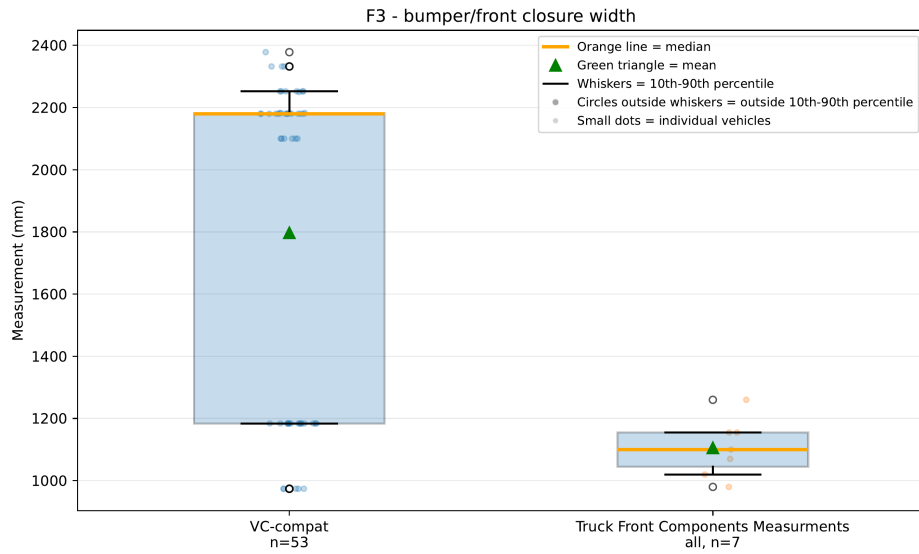


Figure 64: Comparison of F3, bumper/front closure width, between the VC-Compat database and the Truck Front Components Measurements database.

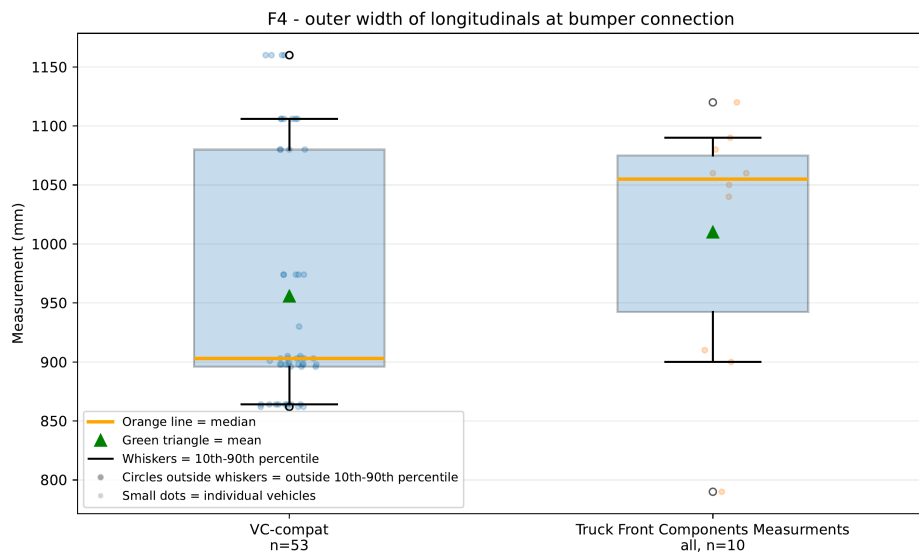


Figure 65: Comparison of F4, outer width of longitudinals at bumper connection, between the VC-Compat database and the Truck Front Components Measurements database.

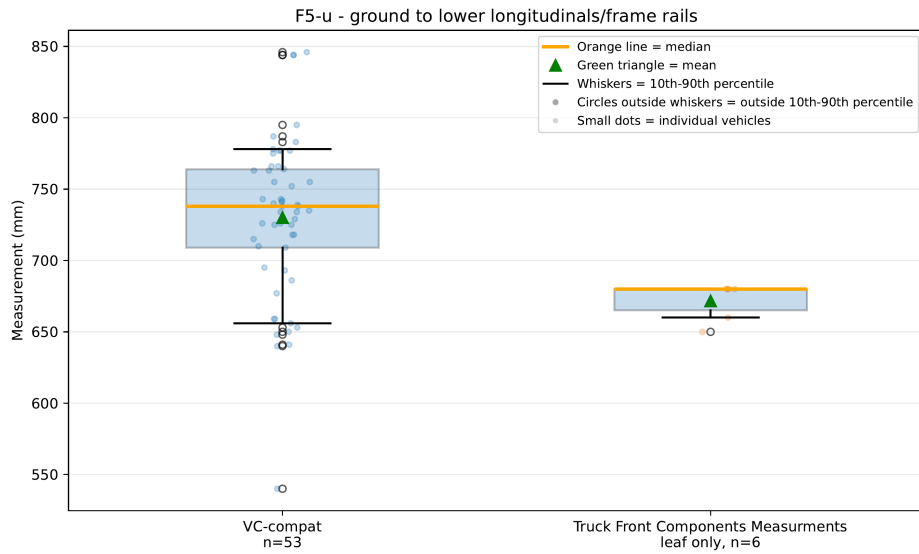


Figure 66: Comparison of F5-u, ground to lower longitudinals/frame rails, between the VC-Compat database and the Truck Front Components Measurements database.

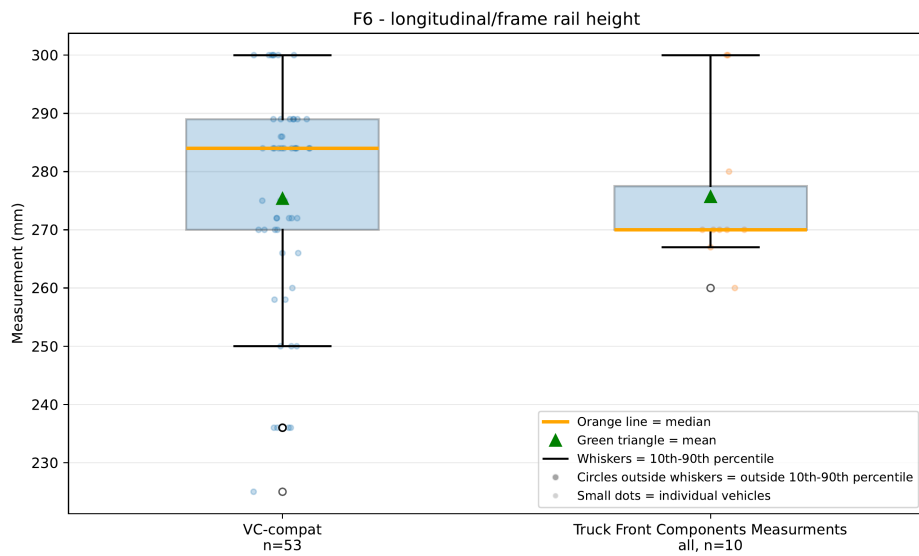


Figure 67: Comparison of F6, longitudinal/frame rail height, between the VC-Compat database and the Truck Front Components Measurements database.

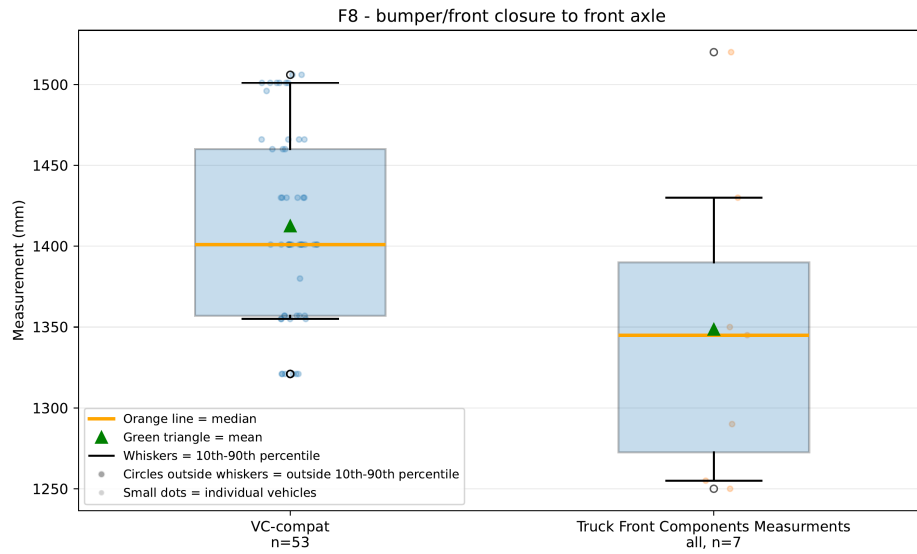


Figure 68: Comparison of F8, bumper/front closure to front axle, between the VC-Compat database and the Truck Front Components Measurements database.

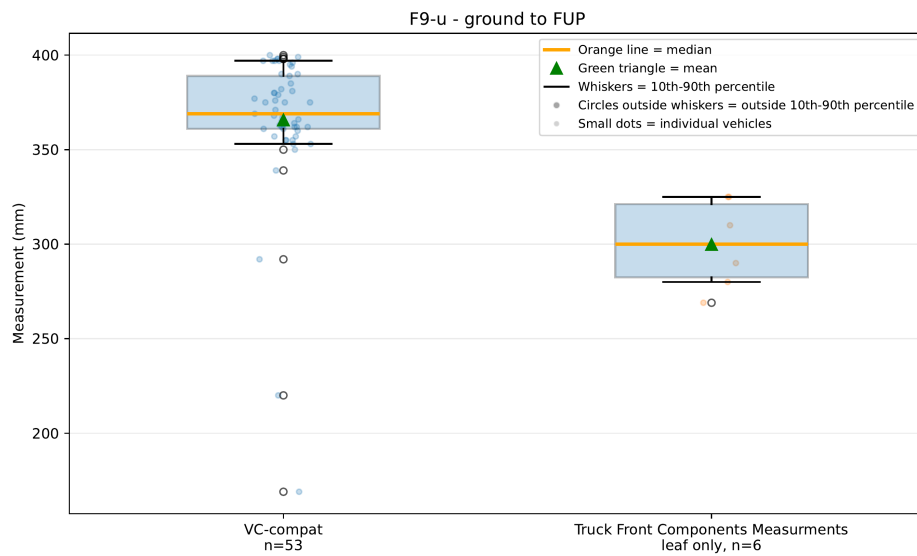


Figure 69: Comparison of F9-u, ground to FUP, between the VC-Compat database and the Truck Front Components Measurements database.

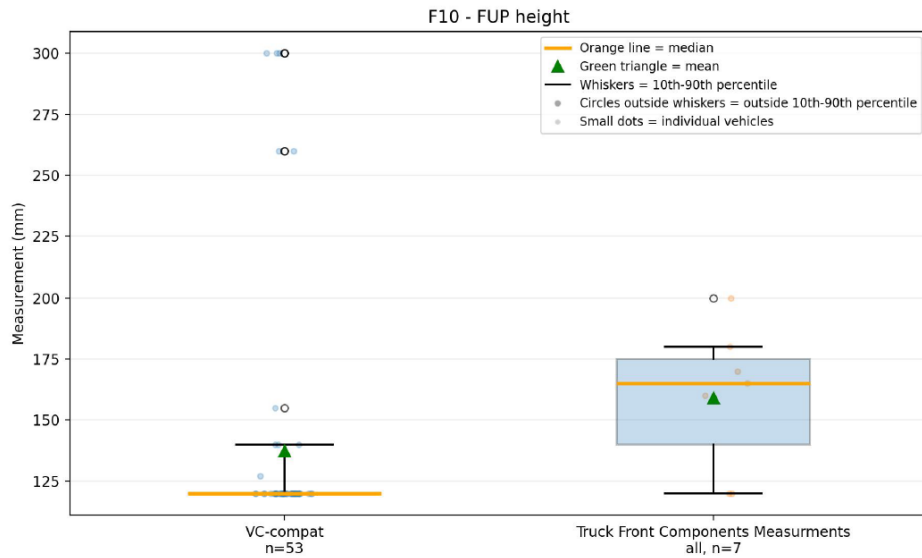


Figure 70: Comparison of F10, FUP height, between the VC-Compat database and the Truck Front Components Measurements database.

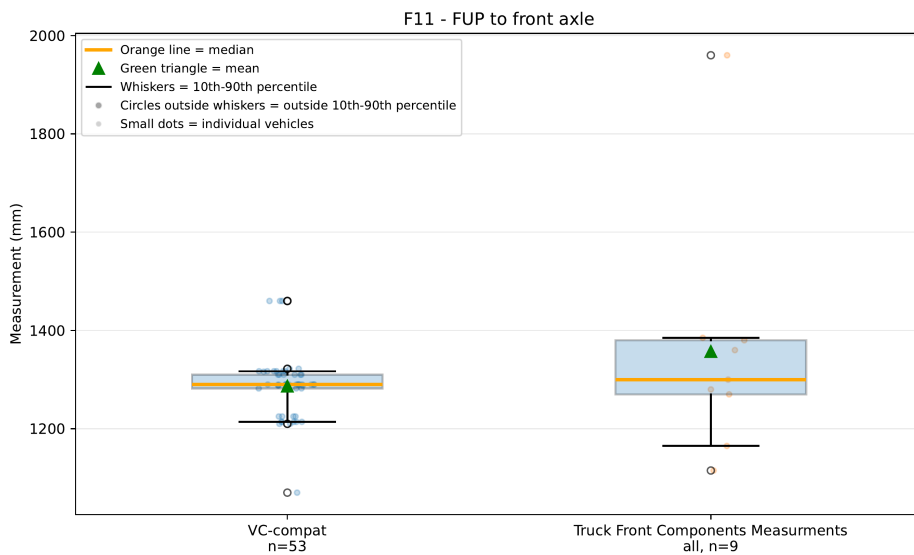


Figure 71: Comparison of F11, FUP to front axle, between the VC-Compat database and the Truck Front Components Measurements database.

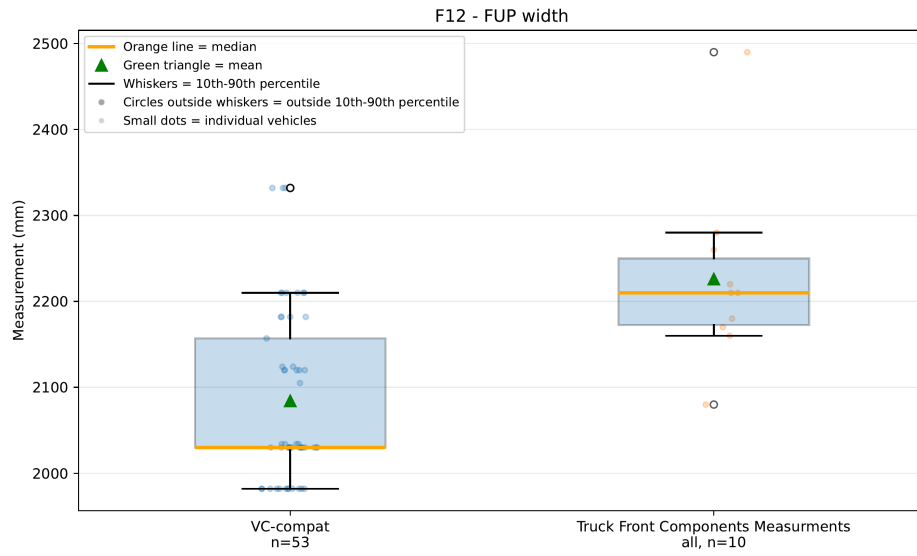


Figure 72: Comparison of F12, FUP width, between the VC-Compat database and the Truck Front Components Measurements database.

C Geometrical Measurement Distributions

The following figures present the geometrical measurement distributions collected from the investigated HGVs. The diagrams were generated based on the TFCM and VC-Compat database. The TFCM database contains a larger number of measurements, and as a result, some of the diagrams include data exclusively from the TFCM database. Since the collected measurements cannot be officially linked to a specific vehicle model, the following information should be considered when interpreting the diagrams:

- Diagrams containing only six vehicles represent vehicles from the TFCM database considered representative of ground-related measurements.
- Diagrams containing ten vehicles represent the TFCM database with construction vehicles included (e.g., Volvo FMX).
- Diagrams containing 59 and 63 trucks compare the measurements available in the VC-Compat database with those from the TFCM database.
- Empty fields indicate that the measurement was either not possible to obtain or that the component was not present on the respective vehicle.

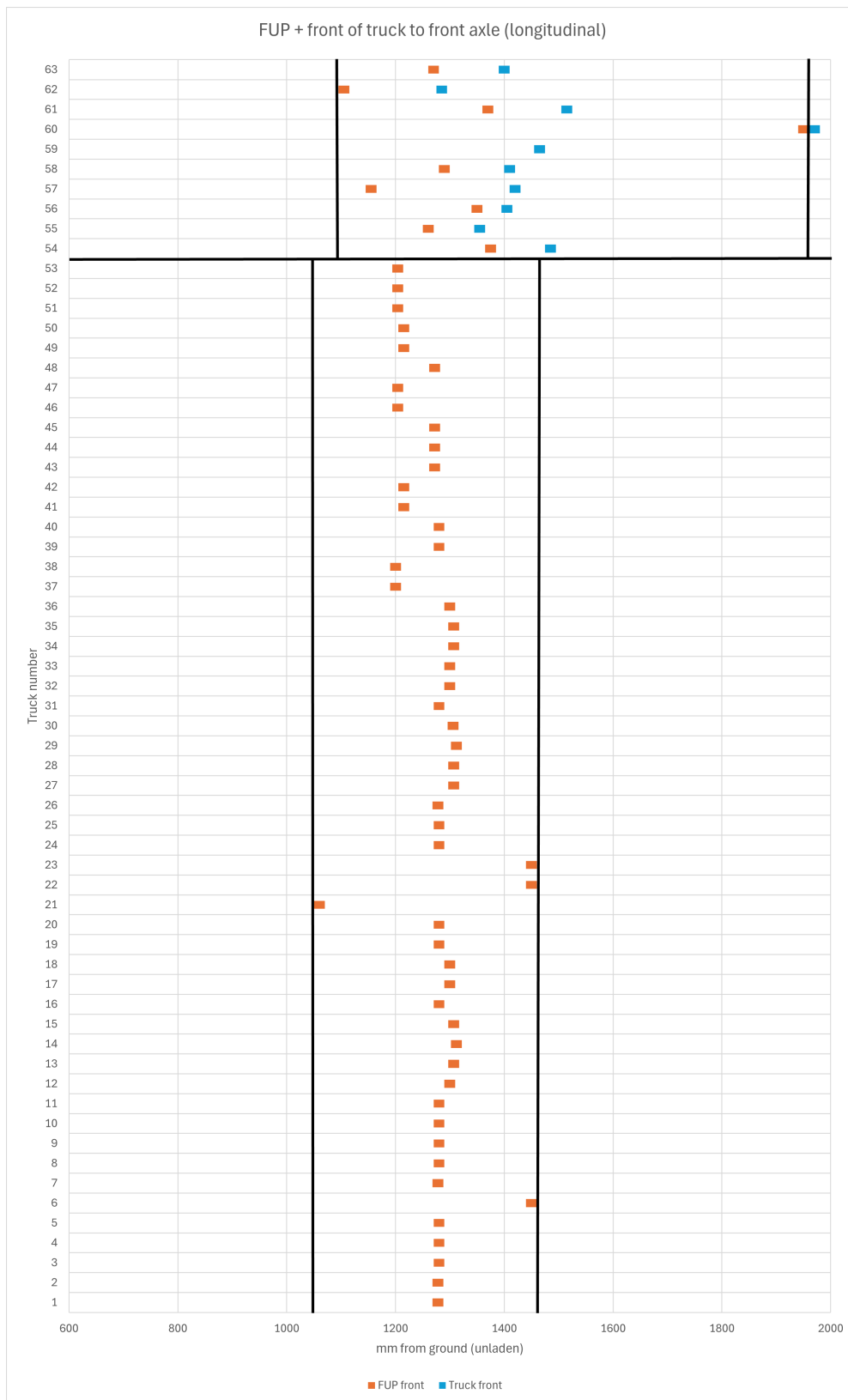


Figure 73: Longitudinal FUP distribution.

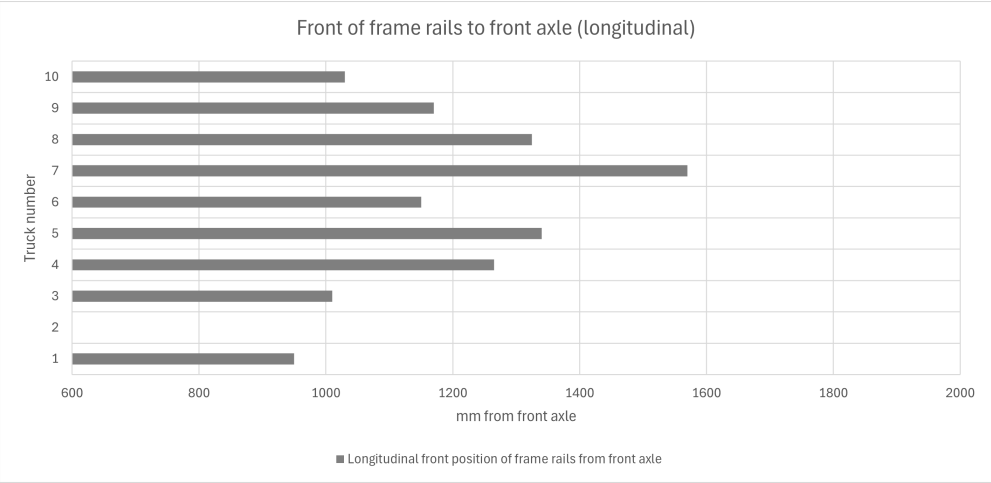


Figure 74: Longitudinal frame rails distribution.

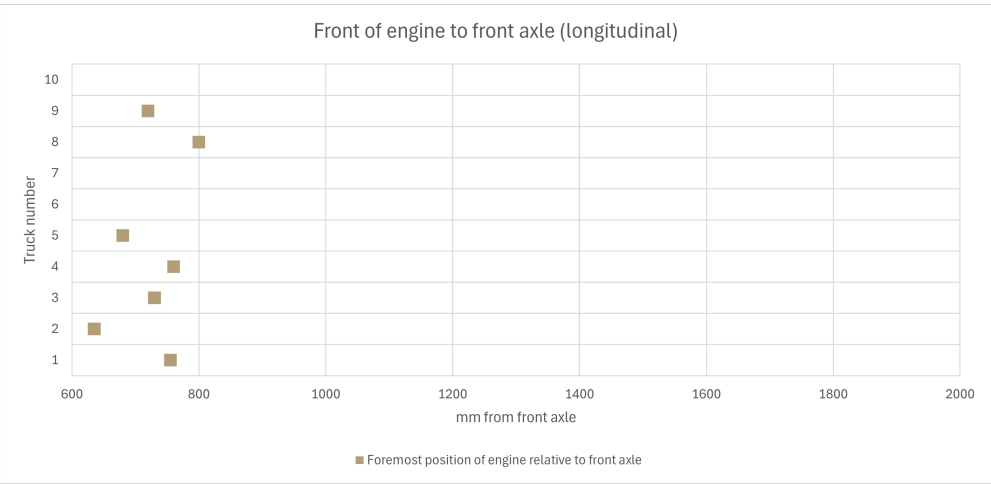


Figure 75: Longitudinal engine distribution.

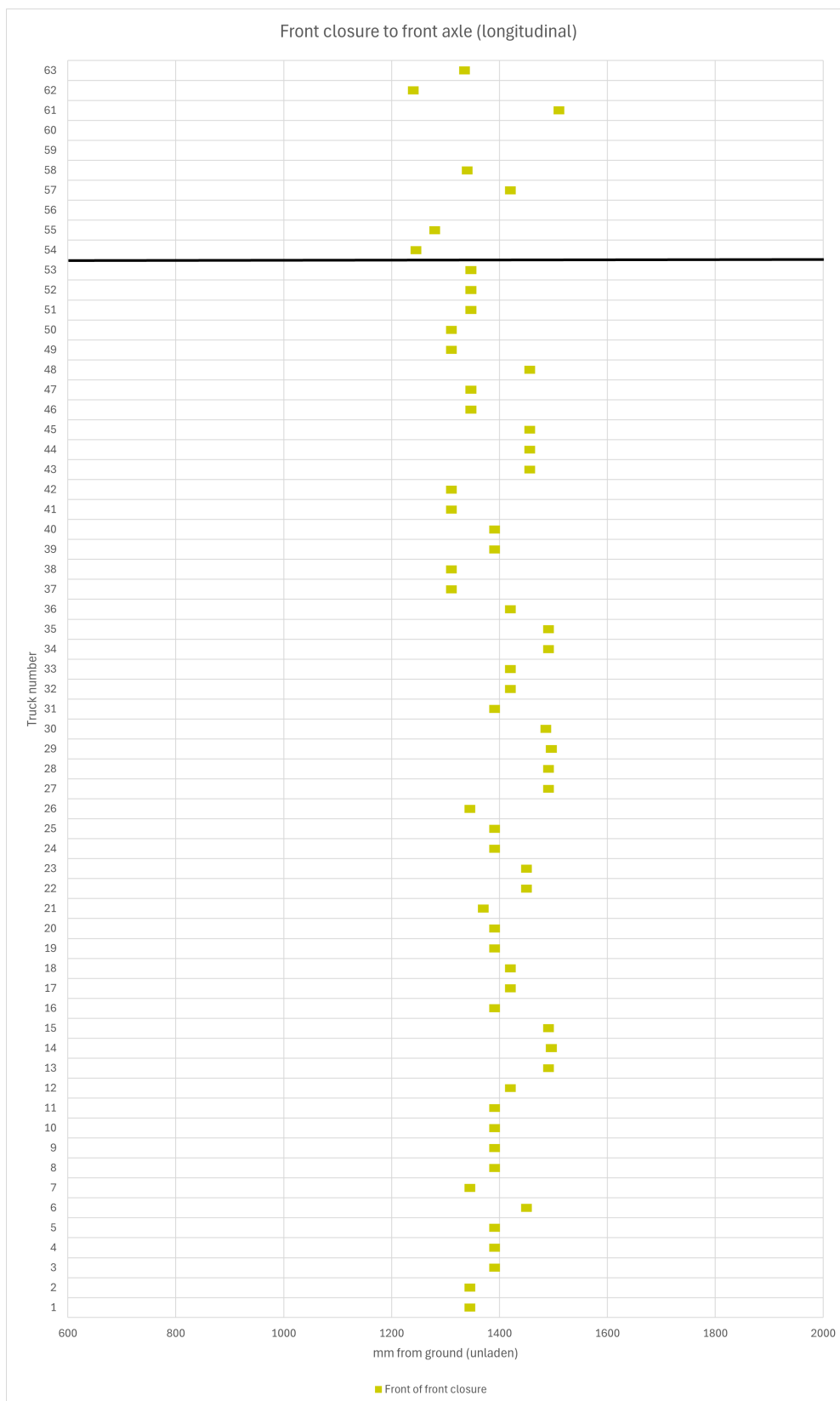


Figure 76: Longitudinal front closure distribution.



Figure 77: Longitudinal radiator distribution.

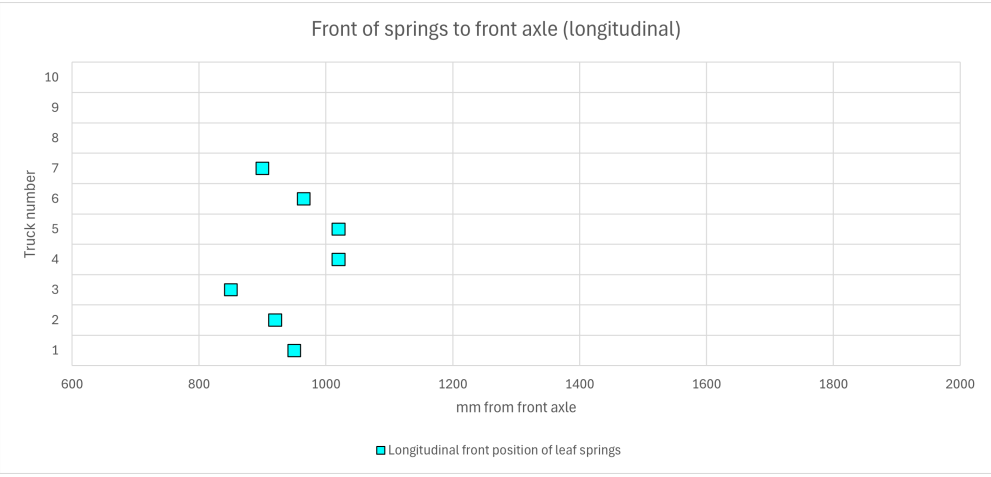


Figure 78: Longitudinal springs distribution.

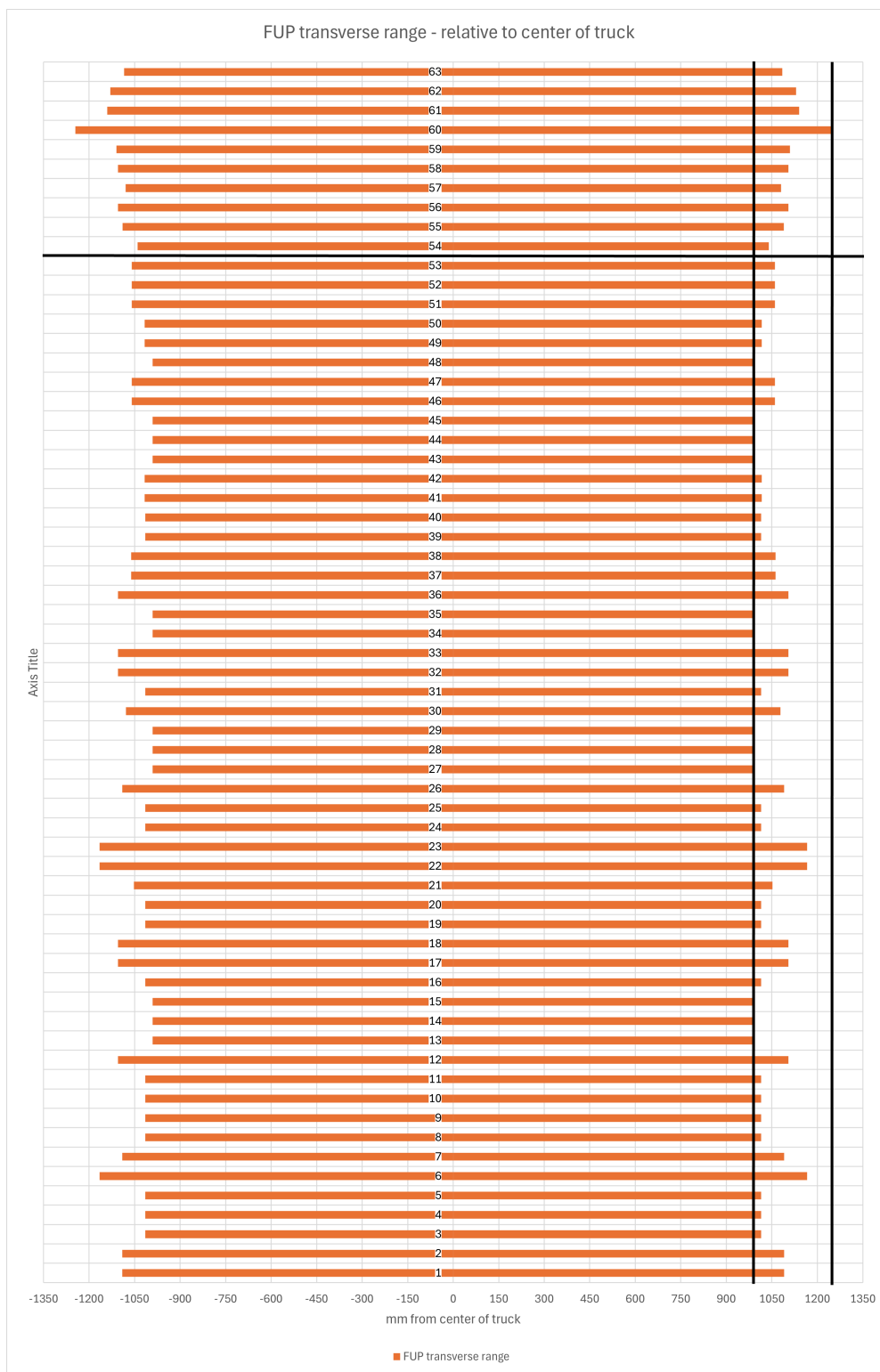


Figure 79: Transverse FUP distribution.

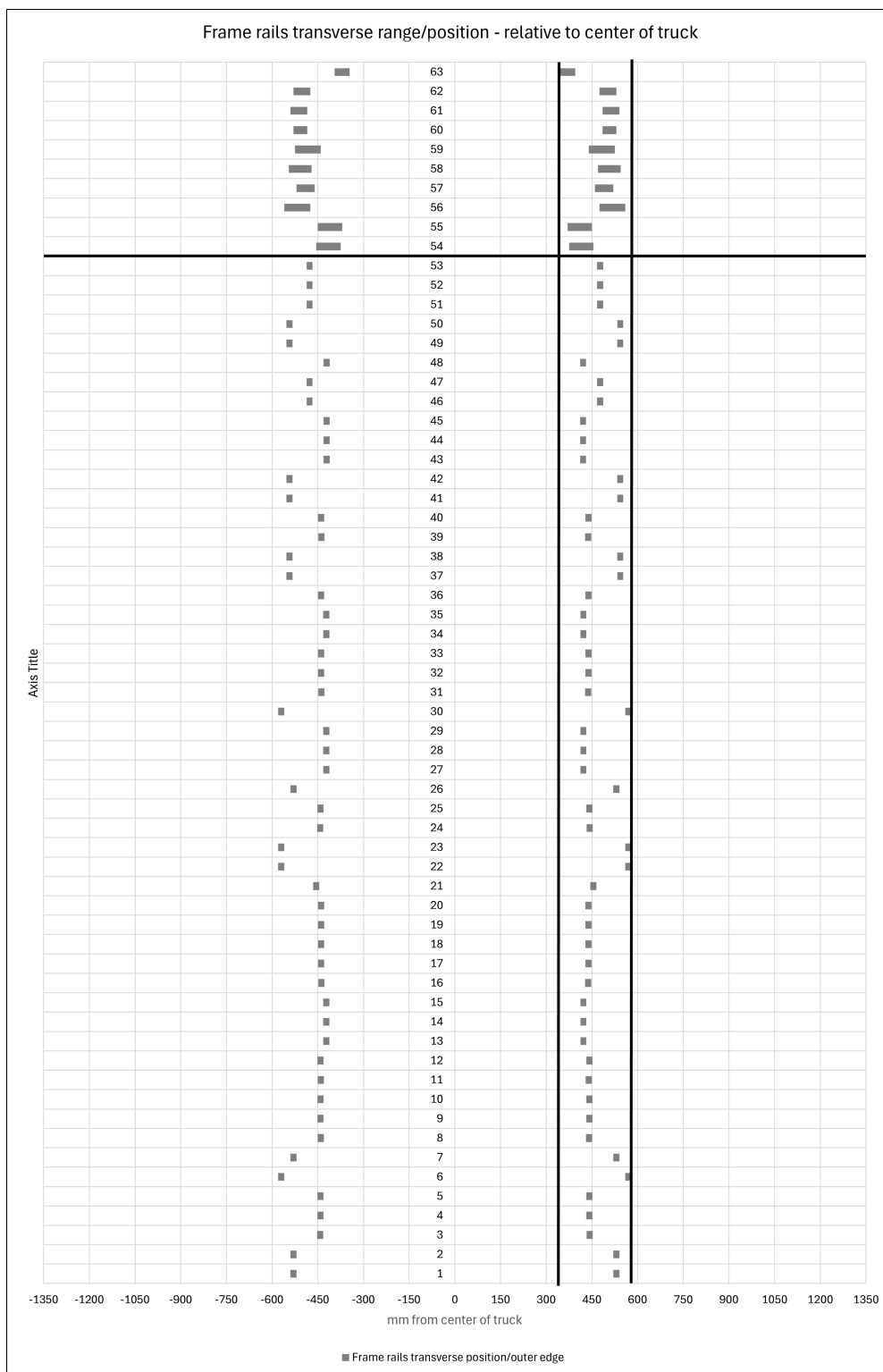


Figure 80: Transverse frame rails distribution.



Figure 81: Transverse front closure distribution.

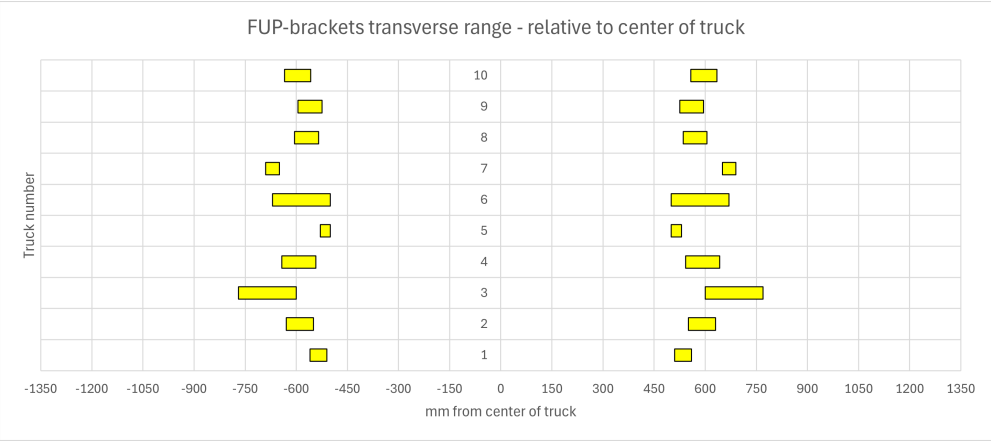


Figure 82: Transverse FUP brackets distribution.

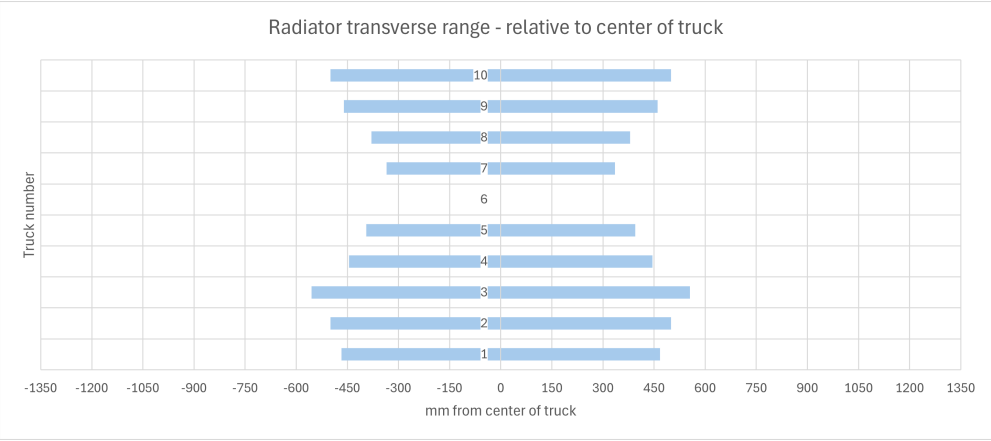


Figure 83: Transverse radiator distribution.

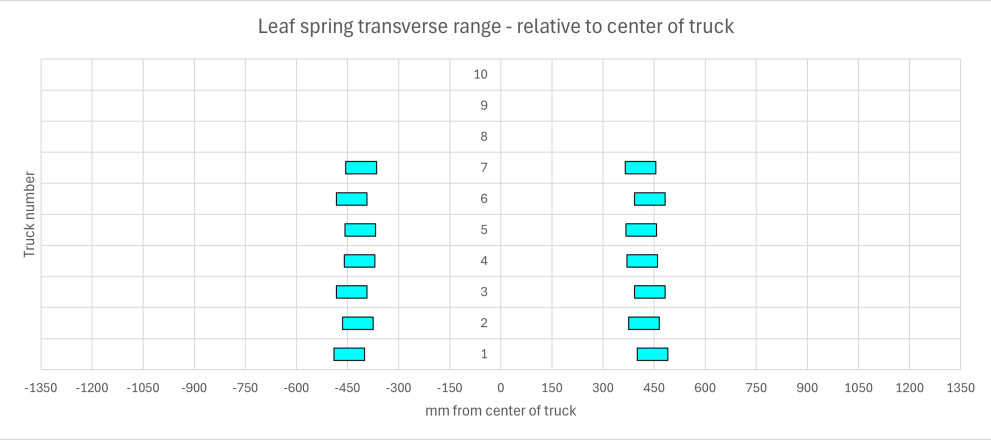


Figure 84: Transverse springs distribution.

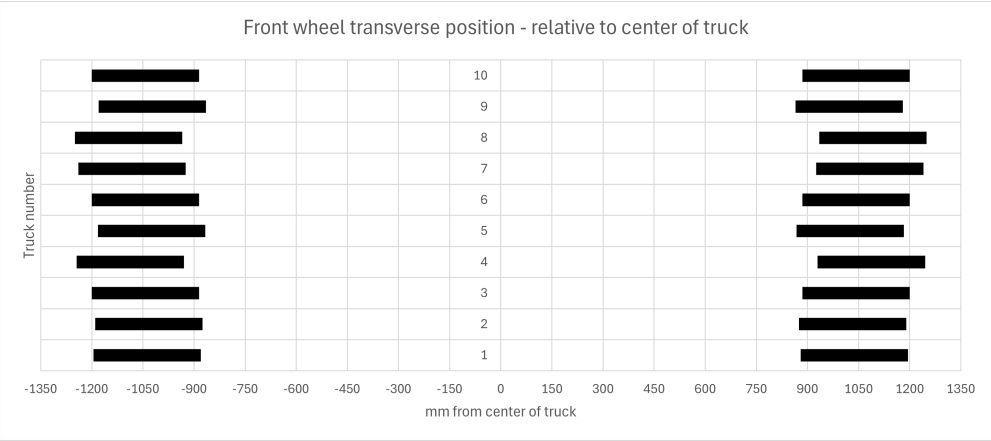


Figure 85: Transverse tyres distribution.

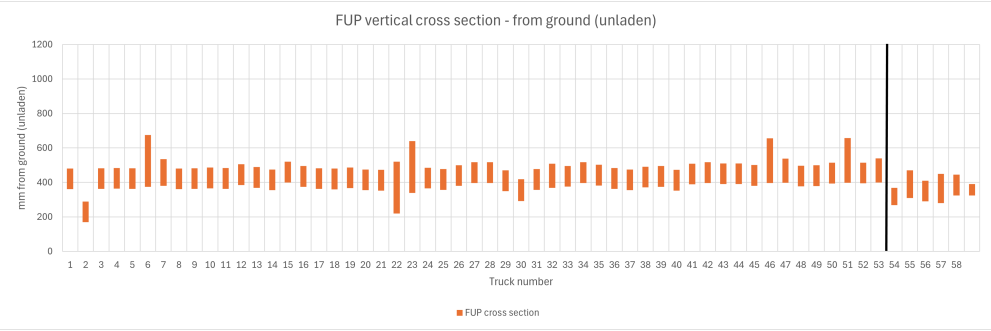


Figure 86: Vertical FUP distribution.

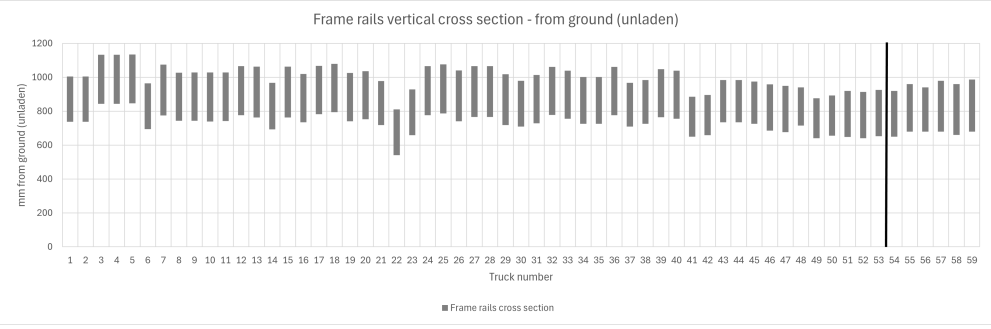


Figure 87: Vertical frame rails distribution.

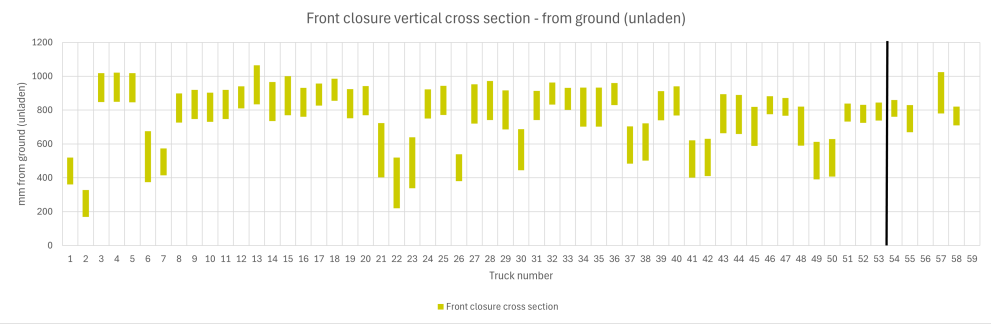


Figure 88: Vertical front closure distribution.

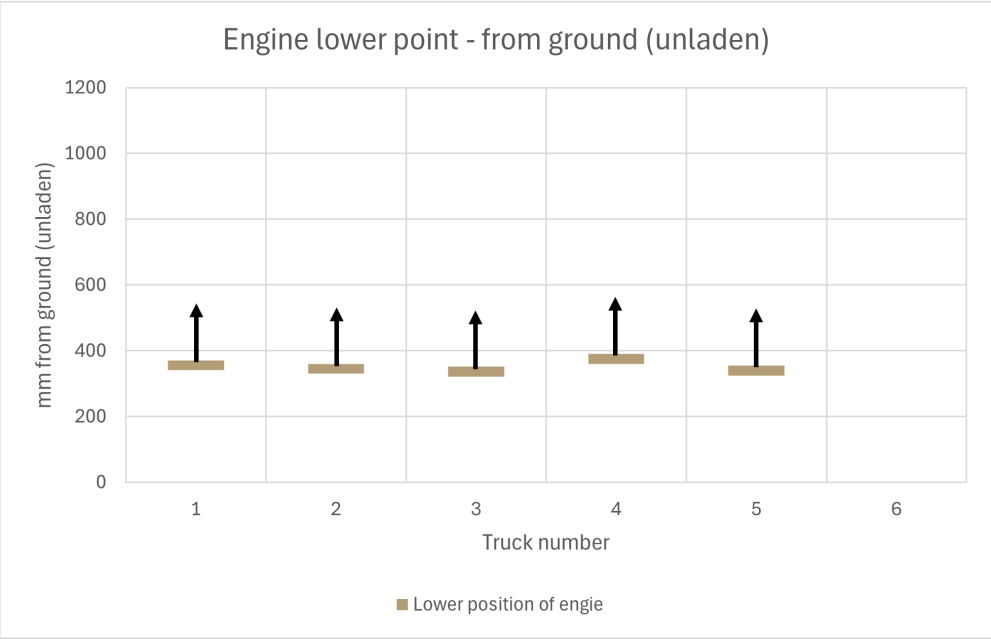


Figure 89: Vertical engine distribution.

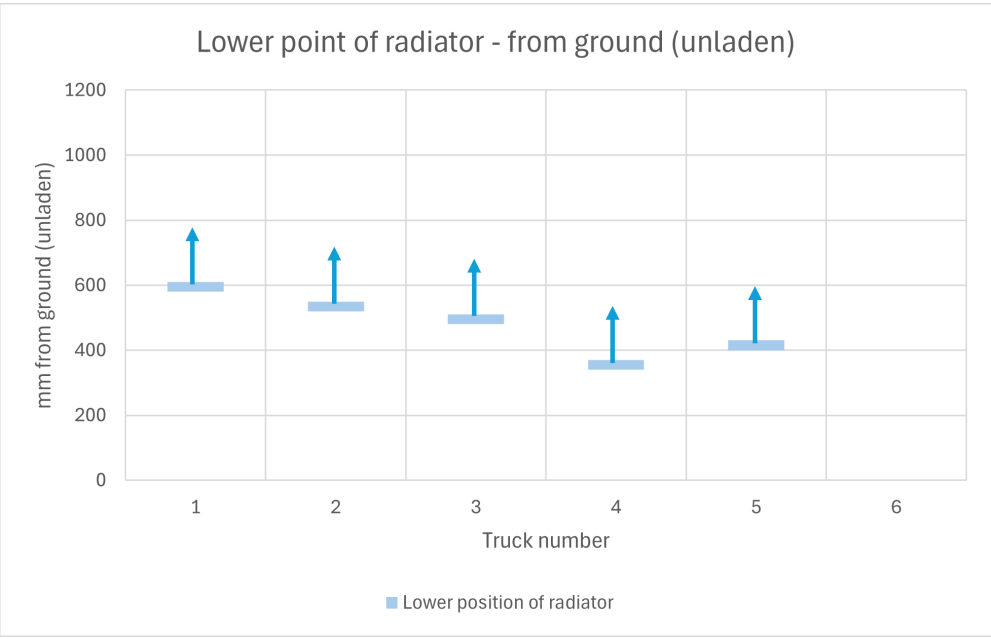


Figure 90: Vertical radiator distribution.

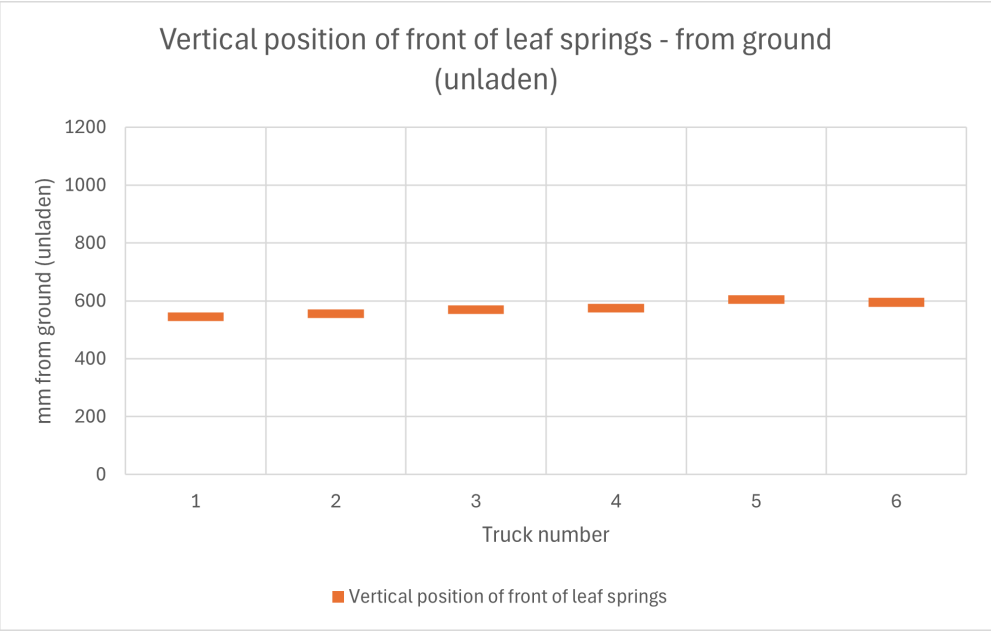


Figure 91: Vertical springs distribution.



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