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Further assessment of petroleum road fuel tankers in frontal impact and rollover

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In partnership with:

TriMech Simulation Solutions
Optimised Engineering Design

APOLLO VEHICLE SAFETY

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Executive summary

The Dangerous Goods Unit at the UK Department for Transport (DfT) aims to ensure that the transport of dangerous goods by road is undertaken safely, and that the regulations (mainly the ADR[1] and its referenced standards) used to achieve this are proportionate, do not needlessly hinder trade, and allow innovation. Should consideration be given to provisions that will facilitate innovation in design and trials of longer and heavier goods vehicles, these vehicles could include extra-large tank vehicles for the carriage of petroleum (flammable liquid).

The research reported here built on previous work which investigated the performance of petroleum road fuel tankers in rollovers and frontal impacts[2]. It focused on frontal impact collisions, mainly answering a question arising from previous work, but it also considered the appropriateness of the current regulations for potential extra-large tank vehicles, both in terms of frontal impact collisions and rollover.

Frontal impact

The previous research work identified that frontal impacts can present significant risks of substantial releases of flammable liquid, particularly for articulated vehicles with self-supporting trailers. Collision analysis work identified one frontal impact incident resulting in a substantial release of flammable liquid, (referred to as the baseline collision) in which the front of one articulated road fuel tanker struck the rear of another. This impact loading scenario was modelled to better understand damage and failure mechanisms and found that buckling failure could occur under comparatively low loading conditions. This posed the question of why weren't more examples identified in the collision analysis given the modelling predicted a high likelihood of failure?

A number of possible reasons for this were postulated including:

- The collision type (frontal impact with heavy vehicle) is rare.
- Few tankers in the fleet are susceptible to this failure type.
- The Finite Element (FE) models used may not be fully representative of real-world tanker designs and/or the associated loading conditions may not be fully representative of real-world collision loads.

DfT commissioned TRL and its partners, TriMech Simulation Solutions and Apollo Vehicle Safety, to perform further research to address this underlying question and meet the following objectives:

[1] UNECE Agreement concerning the International Carriage of Dangerous Goods by Road (ADR 2025): https://unece.org/adr-2025-files

[2] Edwards *et al.* 2023. 'Research on performance test procedures for petroleum road fuel tankers: Summary report'. TRL Published Project Report PPR2027. https://www.trl.co.uk/uploads/trl/documents/PPR2027-Research-on-performance-test-procedures-for-road-fuel-tankers---summary-report--v-final1.1-041023.pdf

- For petroleum road fuel tankers involved in frontal impacts, determine whether a noticeable real-world issue exists, either in general or with specific makes / models of tanker trailers.
- If any concerns are identified, determine how they can be best addressed with respect to ADR requirements, referenced standards, and applicable technical codes.

The research performed consisted of three main activities as follows:

1. Collision analysis to determine the frequency of flammable liquid (FL) tanker frontal impacts with heavy vehicles in GB with significant risk of product release.
2. Fleet analysis to understand the variation of flammable liquid (FL) tanker semi-trailer design in the current GB fleet.
3. Finite element (FE) modelling to compare and contrast the performance of generic semi-trailer designs.

The research found that the main answer to the question was that the type of collision in which these failures occur, i.e. frontal impact with another heavy vehicle, are very rare, with an average of one incident every five years. The research also concluded that even though all semi-trailer tankers in the fleet, including those of a supported design type, would buckle and fail in a frontal impact with another heavy vehicle of severity of the baseline collision, delta V ~ 15 mph (24 km/h), all of them maintain their integrity in frontal impacts with light vehicles. This is somewhat serendipitous given that frontal impact collisions with light vehicles are about 15 times more frequent than those with heavy vehicles. However, it also indicated that the integrity of new novel semi-trailer tanker designs, such as extra-large tankers, should not be allowed to fall below that of current ones because that could lead to them failing in collisions with light vehicles and ultimately result in many more flammable liquid releases. To address this concern, potential amendments to the ADR EN 13094 referenced standard were suggested for potential future extra-large tank vehicles.

Rollover

The previous research work developed 'performance based' rollover test methods, together with an understanding (from associated finite element modelling) of the test parameters relevant to current tanker designs, and a route to their future adoption in standards and regulation in the form of an outline technical code for rollover resilience. The idea was that this code could be used to help approve petroleum road fuel tankers with novel designs that otherwise would not satisfy the current 'design-based' requirements, *i.e. to provide an alternative means of approval that gives more freedom to innovate, while maintaining an equivalent (the same or a better) level of safety*.

The objective of the current work was to apply the outline technical code and associated understanding:

- To investigate the safety implications for extra-large tank vehicles in rollover, and
- If any concerns are identified, to determine how they may be best addressed in view of ADR requirements, referenced standards, and technical codes.

The work concluded that it may be necessary to consider the effect of joint design to assure that extra-large tank vehicles have adequate energy absorption capability in rollover.

However, confidence in the forming limit diagram / omega approach and curves used to derive this conclusion, although reasonable, was not high enough to support changes to the regulatory requirements

This conclusion was derived from the following findings:

- The current EN 13094 standard permits the construction of a potential extra-large tank vehicle which requires strengthening elements with a maximum energy absorption capability at the upper end of that seen for current conventional tankers, specifically one with a stuffed design.
- A banded design, which has a lower energy absorption potential before failure compared with a stuffed design, would also be permitted by the current EN 13094 standard for a potential extra-large tank vehicle. However, curves of omega against energy absorption potential per strengthening element for different joint designs, derived as part of the previous work, show that a banded design may not have sufficient energy absorption potential. However, it should be noted that although confidence in the derived curves was reasonable, it was not high because work to validate them was limited, for example no drop test work was performed.

Regarding next steps, should consideration be given to allowing the use of extra-large tank vehicles, it is suggested that tanker stakeholders are informed of the importance of consideration of the additional energy absorption required in topple impact for these vehicles due to their additional weight, and particularly the influence of joint design. Planned project dissemination activities should help achieve this objective.

It is also suggested that further work is performed to help validate the technical code developed in the previous project and demonstrate the influence of joint design which would provide more confidence in the forming limit diagram / omega approach and the curves used to derive the conclusion. A first part of this work could be the verification of the difference in behaviour for tanks with banded and stuffed type joint designs predicted by the approach and curves. This could be achieved by performing two sub-section drop tests with similar tank sections but with different joint designs at an appropriate impact energy. Further work could be performed to verify other aspects of the technical code such as its robustness for different tanker cross-sections and the magnitude of the safety factor recommended.

1 Introduction

1.1 Research background

The Dangerous Goods Unit at the Department for Transport (DfT) aims to ensure that the transport of dangerous goods by road is undertaken safely, and that the regulations used to achieve this are proportionate, do not needlessly hinder trade and allow innovation. Goods vehicles used for the carriage of dangerous goods must comply with the construction requirements set out in the UNECE Agreement concerning the International Carriage of Dangerous Goods by Road (ADR 2025)[3], which references European standards EN13094 'Design and Construction – gravity discharge' tanks and EN12972 'Testing, inspection, and marking of metallic tanks'. In Great Britain, these vehicles must be certified by the Driver and Vehicle Standards Agency (DVSA). Vehicles transporting flammable liquids by road are certified as "FL vehicles".

Many of the requirements of ADR currently applied to the tanks of FL vehicles are prescriptive design requirements. Typically, the design-based approach to regulation does not directly control the desired safety outcome, but controls easily defined proxies for that performance. For example, ADR defines the material types, thicknesses, and joining techniques used in the structure of the tank, reflecting a design that has evolved over time and has been shown to be safe. However, technology and demands for different tanker designs are always changing, and therefore, standards need to evolve to ensure that safety is not compromised. For example, the UK DfT is currently considering provisions to facilitate innovation in design and trials of larger heavy goods vehicles to:

- Reduce emissions and congestion
- Improve safety and productivity

Any introduction of longer and/or heavier vehicles for general haulage could see suggestions that extra-large tank-vehicles (i.e. those with a gross capacity which exceeds about 45,000 litres), including FL vehicles should also be permitted. This would have implications for the relevant standards to ensure safety is not compromised.

Previous work for the DfT investigated the performance of petroleum road fuel tankers in rollovers and frontal impacts[4]. The main focus of this work was on rollovers, but some initial research was carried out on frontal impacts.

The research identified that frontal impacts can present significant risks of substantial releases of flammable liquid, particularly for articulated vehicles with self-supporting trailers.

[3] UNECE Agreement concerning the International Carriage of Dangerous Goods by Road (ADR 2025): https://unece.org/adr-2025-files

[4] Edwards *et al.* 2023. 'Research on performance test procedures for petroleum road fuel tankers: Summary report'. TRL Published Project Report PPR2027. https://www.trl.co.uk/uploads/trl/documents/PPR2027-Research-on-performance-test-procedures-for-road-fuel-tankers---summary-report--v-final1.1-041023.pdf

When such a vehicle is involved in a frontal impact, collision forces are transmitted to the tank indirectly through the fifth wheel and king pin assembly.

Collision analysis work highlighted a specific incident in which the front of one articulated road fuel tanker struck the rear of another, resulting in around 7,000 litres of flammable liquid being released. This impact loading scenario was modelled to better understand damage and failure mechanisms. This modelling analysis found that factors such as the length of king pin support structure and its attachment to the tank had a large influence on the loads that could be sustained before buckling was predicted. The results further indicated that buckling failure could occur under comparatively low loading conditions.

However, this work did not fully answer the underlying question: given that results from the modelling indicated a high likelihood of failure, particularly for short king pin support structures, why were more examples not observed in the collision analysis? A number of possible reasons for this were postulated including:

- The collision type (frontal impact with heavy vehicle) is rare.
- Few tankers in the fleet are susceptible to this failure type.
- The Finite Element (FE) models used may not be fully representative of real-world tanker designs and/or the associated loading conditions may not be fully representative of real-world collision loads.

DfT commissioned TRL and its partners, TriMech Simulation Solutions and Apollo Vehicle Safety, to perform further research to address this underlying question and meet the following objectives:

- For petroleum road fuel tankers involved in frontal impacts, determine whether a noticeable real-world issue exists, either in general or with specific makes / models of tanker trailers.
- If any concerns are identified, determine how they can be best addressed with respect to ADR requirements, referenced standards, and applicable technical codes.

For rollover, the previous research aimed to develop ‘performance-based’ finite element modelling approaches and appropriate physical test procedures to approve petroleum road fuel tankers with novel designs that otherwise would not satisfy the current ‘design-based' requirements, *i.e. to provide an alternative means of approval that gives more freedom to innovate, while maintaining an equivalent (the same or a better) level of safety*.

The research found that the deflections and likelihood of major loss of containment experienced by road fuel tankers in real-world rollover events could be replicated using a suitably specified, two-compartment subsection drop test (or a full-scale physical topple test) supplemented by abrasion and penetration tests. It also developed performance-based rollover test methods, together with an understanding (from associated finite element modelling) of the test parameters relevant to current tanker designs, and a route to their future adoption in standards and regulation in the form of an outline technical code for rollover resilience.

In anticipation of the need for provisions for extra-large tank vehicles, DfT commissioned TRL and its partners to apply the insights and findings from the previous rollover research to investigate safety implications for extra-large tank vehicles in rollover events. This was based on the principle that, compared to conventional tankers which fulfil current ADR requirements, safety should not be compromised.

1.2 Structure of this report

This report is divided into two main parts: 'frontal impact' and 'rollover'. In each of these parts, background information, the research performed, its findings, and any implications for regulation are described. The majority of the research focussed on frontal impact because the understanding for this collision type was less advanced, primarily because the previous research focused on rollover type collisions which occur more frequently.

2 Frontal impact

As described in the 'Introduction' section above, the objectives of the frontal impact research were:

- For petroleum road fuel tankers involved in frontal impacts, determine whether a noticeable real-world issue exists, either in general or with specific makes / models of tanker trailers.
- If any concerns are identified, determine how they can be best addressed with respect to ADR requirements, referenced standards, and applicable technical codes.

To meet these objectives, it was decided to divide the work into two parts: the first addressing the initial objective, and the second building on its findings to address the second objective.

The first part consisted of three main activities, as follows:

1. Collision analysis to determine the frequency of FL tanker frontal impacts with heavy vehicles in GB with significant risk of product release.
2. Fleet analysis to understand the variation of FL tanker semi-trailer design in the current GB fleet.
3. Finite element (FE) modelling to compare and contrast the performance of generic semi-trailer designs.

The structure of this section is as follows. The first sub-section provides background information to facilitate understanding of the research results. The next sub-sections report the results of the three main work activities. The following sub-section discusses the key findings of these activities. The next sub-section details the implications of these findings for potential future extra-large tank vehicles and suggests amendments to the EN 13094 standard to address them. The final sub-section summarises the conclusions and outlines the next steps.

2.1 Background information

2.1.1 Baseline collision

As mentioned in the 'Introduction' section above, previous work found that frontal impacts of petroleum road fuel tankers can present significant risks of substantial flammable liquid releases, particularly for articulated vehicles. One example of a frontal impact collision was found in which there was a substantial release of flammable liquid.

To compare the performance of different semi-trailer tanker designs in frontal impacts and determine if some designs are more likely to fail than others, it was decided that this collision should be considered a baseline for subsequent modelling work. For reference, it is described in greater detail below.

The collision occurred in 2016. The front of one fuel tanker struck the rear of another fuel tanker and there was a substantial release of flammable liquid from the striking tanker. The lead tanker, which was nearly stationary (travelling at 2 km/h), was struck at the rear of the tank by the following tanker at an impact speed of 48 km/h, and with an overlap of approximately 75%.

The impact caused the rear bumper of the lead vehicle to be heavily deformed but, based on the photographs, only minor markings and minimal deformation was evident to the rear of the tank itself, which maintained its integrity. In contrast, the following striking tanker sustained extensive damage to the front of the vehicle across its full width, causing the cab to be crushed and pushed back on the chassis, although it did not appear to substantially under-run the lead vehicle (Figure 2-1).

Figure 2-1: Pictures of baseline collision showing minimal deformation of tank of struck tanker and failure of striking tanker tank just behind the king pin

Further examination of the collision concluded that the mechanism for the failure of the striking tanker was loading transmitted via the fifth wheel and king pin assembly, which would have formed the main load path for decelerating the mass of the trailer. As illustrated in Figure 2-2, the inertia of the loaded tank would have acted through the centre of mass, which is located at a greater height than the king pin providing the opposing force. This would have created a force couple and a rotational moment, resisted by the mass of the vehicle acting downward.

It appeared that the net effect of the forces applied by the trailer's own mass and the mass of the load it contained, caused the tank to deform and buckle in the region just behind the king pin. The rotational moment caused the front section to bend downward, causing tearing of the tank shell. The tank ruptured in the region of this deformation, above the king pin, and released almost all of the fuel in the front compartment of the tanker (c. 7,000 litres).

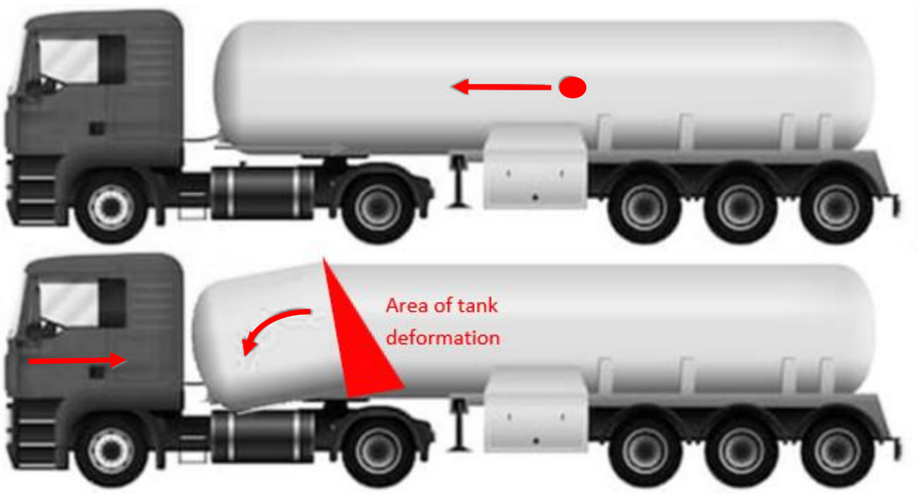


Figure 2-2: Illustration of crash loading of the following tanker through the king pin assembly in a tanker front to HGV rear collision and the resulting deformation (Top: before impact, Bottom: after impact)

2.1.2 Description of tanker design

There are a number of factors related to tanker semi-trailer design which are likely to influence its performance in frontal impact. Two main factors frequently referred to in this report are whether it is supported or self-supported, and whether it has a banded or a stuffed type design for the joints between the tanker shell and partitions / baffles. These design factors are described detail in Section 2.3.2.2, with examples of each type of design. However, in summary:

- Supported trailers have a longitudinal framework that runs the full length of the tank, upon which the tank is supported whereas self-supported trailers do not. Although self-supported trailers may have a longitudinal framework supporting some parts of the tank, it does not run the full length of the tank; usually there is no support for the conical section behind the king pin. This design allows the tank to be lower in this area, avoiding structural interference with movement of the tractor unit.
- Stuffed trailers have a different partition / baffle to joint construction to banded trailers, mainly because of the different ways in which they are manufactured. For stuffed trailers, the shell is made from pieces which are the length of the tank and are shaped and welded together longitudinally. Partitions / baffles are attached to the shell inner using lap joints. For banded trailers, the shell is made from shorter lengths, joined together longitudinally by welding them to extruded bands. These bands have

an upstand, which helps stiffen the structure, and to which the partitions / baffles are welded.

2.1.3 Current regulatory requirements for king pin loading

The main regulatory requirements for semi-trailers tankers for king pin loading are contained in EN 13094:2022 'Tanks for the transport of dangerous goods - Metallic gravity-discharge tanks - Design and construction.' Paragraph 6.3.2 of this standard requires that the design stress for tank shells and their attachments is not exceeded for loads of 2 g in the direction of travel.

Paragraph 6.3.2 further states that for the front-end (i.e. the king pin structure), only the maximum mass of the substance carried in the first (front) compartment shall be taken into account. However, this statement is caveated for trailers without a longitudinal framework, upon which the tank is supported, i.e. self-supported tanker designs. For this case, it states that for front attachments (i.e. the king pin structure), the maximum design mass of the trailer shall be deemed to act where the coupling device attaches to the tank.

From consultation with an Appointed Inspection Body (AIB), it is understood that, generally, linear Finite Element analysis is used by tank manufacturers to assure compliance with the EN 13094 2 g king pin longitudinal loading requirement, without exceeding the design stress.

It is also understood that the origin of the requirement was to address loads that may be experienced in service as a result of heavy braking. The loads are substantially higher than the maximum applied to the tanker through braking, circa 1.0 g, to account for the additional loading that can be caused by fuel movement and arrest, i.e. the fuel moves forward in the tank when the braking is initially applied, but then is brought to a sudden halt when its motion is stopped by the partitions and baffles which causes transient loads substantially above the braking deceleration applied.

2.2 Activity 1: Collision analysis

The main objective of this activity was to determine the frequency of FL tanker frontal impacts with heavy vehicles in GB that present a significant risk of product release.

Previous work on this topic completed by (Knight and Dodd, 2019)[5] analysed the STATS19 national collision data and the ADR collision reports. This study found that front to rear collisions can present a significant risk of substantial flammable liquid release , both when the tank of the FL vehicle is hit at the rear and damaged by direct contact, and, in the case of articulated vehicles, when the FL vehicle is hit at the front and collision forces are transmitted to the tank indirectly through the 5th wheel and king pin. However, in the ADR reports for the period 2014-2016, there were only two cases that involved a frontal impact. The first occurred when an icy road caused a tanker to leave the road and drive head-first into a stream. The tank was empty at the time, so no leakage was reported. The second case, which was mentioned previously in Section 1, was a tanker-front to tanker-rear collision that caused a leak from the tank of the striking vehicle. This implied that FL tanker frontal impacts with heavy vehicles in GB with significant risk of product release are rare.

To verify this conclusion and better determine the frequency of FL tanker frontal impacts with heavy vehicles in GB with a significant risk of product release, this activity performed work to:

- Update the STATS19 analysis previously completed by (Knight and Dodd, 2019)[5] to incorporate the most recent data.
- Expand analysis of ADR collision reports to identify further relevant collisions.
- Compare national (STATS19) data and ADR reports, to assess the reporting levels for incidents involving petroleum road fuel tankers and gain confidence that the reporting system is sufficiently robust to reliably capture the true frequency of such incidents.

The results of this work are reported below.

2.2.1 STATS19 collision analysis

The STATS19 database provides data from all police reported road collisions involving personal injury in Great Britain. This database was used to analyse collisions involving all goods vehicles. It is not possible to identify fuel tankers directly within STATS19. The database does include a field that describes the body type of a vehicle, but, especially in the case of articulated vehicles, this can be unreliable as often the coding reflects the classification of the towing vehicle (e.g. tractor unit) rather than the trailer (e.g. tanker, curtain-sided, etc). Therefore, an alternative approach was needed to identify incidents involving flammable liquid (FL) tankers.

The Driver & Vehicle Standards Agency (DVSA) holds a database of all vehicles and trailers that have applied for ADR certification. From this database, the DVSA compiled a list of Vehicle Registration Marks (VRMs) for all vehicles that had been granted ADR certification as

[5] Knight, I., & Dodd, M. (2019). Performance test procedures for petroleum road fuel tankers. DRAFT report for Part A – Review and analysis of accident data, impact conditions and regulations. London: Department for Transport.

an FL vehicle between 2009 – 2022. For privacy reasons, the publicly available version of the STATS19 database does not include the VRM of each vehicle involved in an incident. However, the DfT does hold this data, and can identify the VRM of vehicles involved in collisions recorded in STATS19. Therefore, the list of VRMs compiled by the DVSA was passed to the DfT, who then matched the list against their STATS19 database to produce a list of the unique collision reference numbers for all incidents in which any of those vehicles had been involved. The match identified 960 vehicles that had been involved in incidents in the period 2009 – 2022. This linked data was returned to Apollo for analysis, without the VRM, so Apollo were unable to identify which of the FL vehicles had been involved in any of the collisions, only that all 960 involved at least one FL vehicle.

2.2.1.1 Number of registered vehicles

Each year, the DfT publishes data on the number of goods vehicles licensed for use in Great Britain[6]. Since 2009, the number of licensed goods vehicles has remained broadly consistent at approximately 400,000 vehicles (Figure 2-3). Approximately 70% of licensed goods vehicles are rigid vehicles. Articulated goods vehicles represent approximately 30% of all goods vehicles, with approximately 80% of these being reported as having 3-axle tractor units.

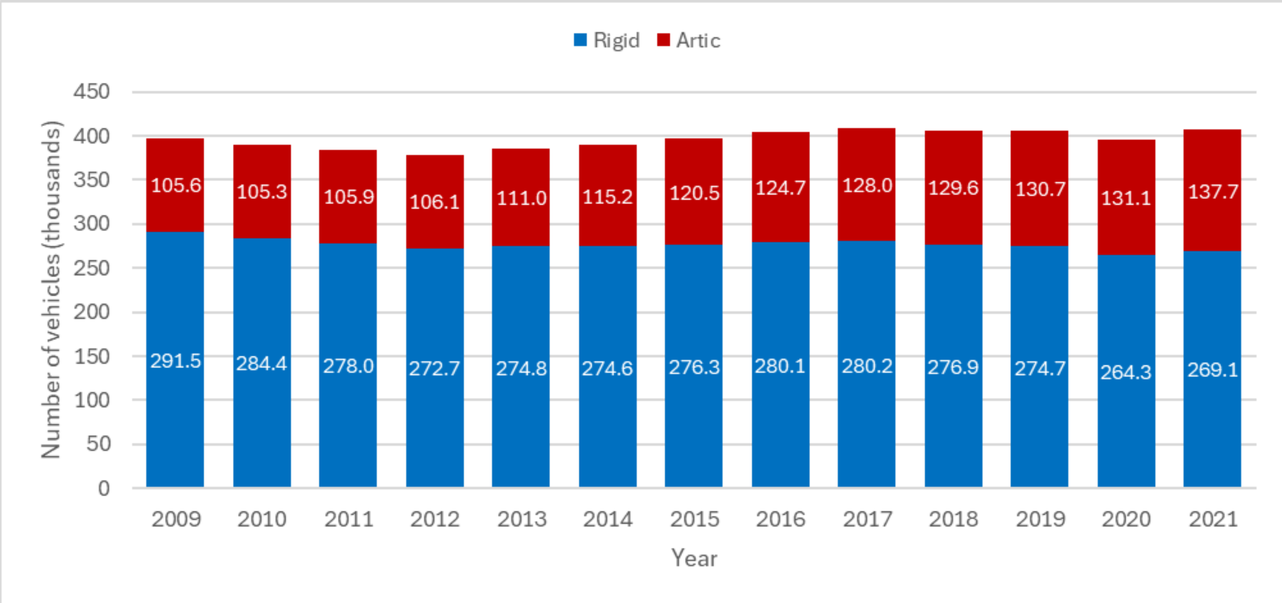


Figure 2-3: Number of goods vehicles (>3.5t) licensed in Great Britain.
Source: gov.uk Table: VEH0524

The DVSA's ADR certification database was expected to provide information about the number of FL Tankers licensed for use each year. Although the database contains data covering the last 10 years, the entries are overwritten each year with the latest available data. This meant that historical annual snapshots were not available, and only data for the latest year for which a vehicle has been approved is stored in the database. A snapshot for the end

[6] Table VEH0524: https://www.gov.uk/government/statistical-data-sets/veh05-licensed-heavy-goods-vehicles

of 2018 was requested as part of the original project by (Knight and Dodd, 2019),and a more recent snapshot for the end of 2023 was requested as part of this study.

Figure 2-4 shows the number of 6-axle articulated FL tankers by year presented by (Robinson *et al.*, 2015)[7] alongside the number of articulated FL tanker trailers that were certified for ADR use at the end of 2018 and 2023. Whilst it is possible that some of the vehicles included in the 2018 and 2023 figures are not 6-axle configuration, it is likely that a very high proportion will be 6-axle vehicles and so it provides a reasonable comparison. It suggests that there has been a gradual increase in the number of FL vehicles approved under the ADR certification process, but it has been level for recent years.

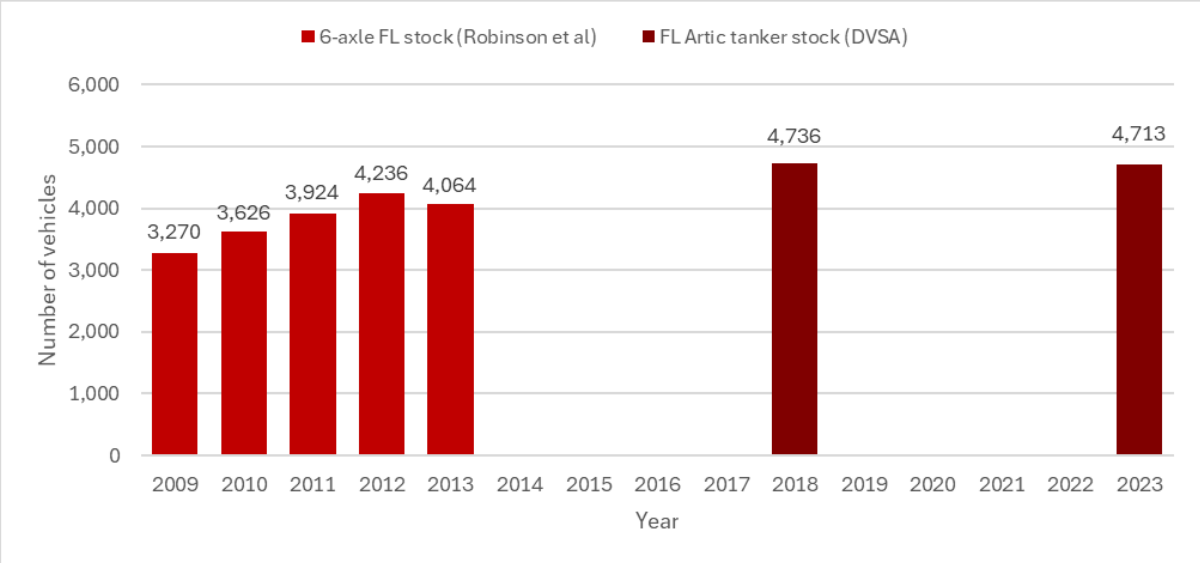


Figure 2-4: FL tanker vehicle stock by year.
Source: (Robinson *et al.*, 2015)[7] and DVSA

The UK Department for Business, Energy and Industrial Strategy (BEIS) publishes regular updates on the volume of UK deliveries of petroleum products for inland consumption. Figure 2-5 shows that there has been a gradual decline in petrol deliveries, offset by an increase in diesel deliveries, which has resulted in a comparable level in 2023 to that seen in 2009. The dip in volume seen in 2020 was likely an effect of the COVID-19 pandemic. However, the overall stability in the volume of deliveries supports the consistency of the size of the FL stock presented in Figure 2-4.

[7] Robinson, B., Robinson, T., Tress, M., & Seidl, M. (2015). Technical Assessment of Petroleum Road Fuel Tankers: Accident Data and regulatory Implications (with extensions). Crowthorne: TRL Published Project Report PPR761

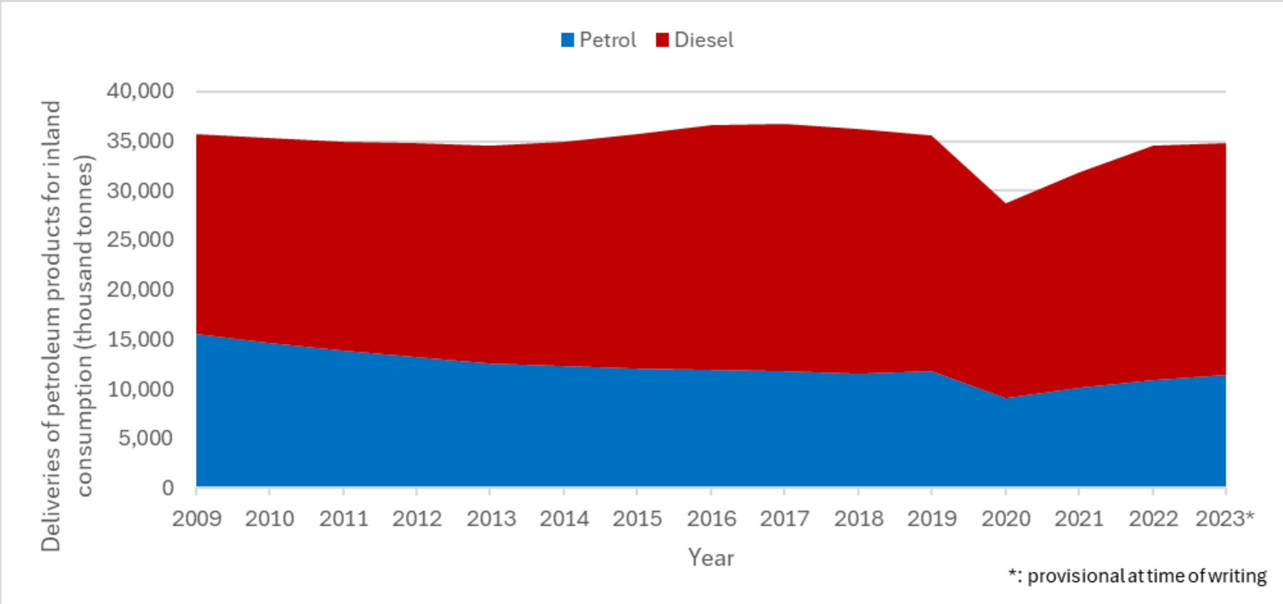


Figure 2-5: Volume of UK deliveries of petroleum products for inland consumption (2009-2018) Source: BEIS Table ET3.13

2.2.1.2 *Number of recorded collisions*

Analysis of the STATS19 database (Figure 2-6) showed that, since 2014 there has been a decline in the number of heavy goods vehicles (GVW >3.5t) involved in road traffic accidents each year, up until 2020 at the beginning of the COVID-19 pandemic. Since then, the numbers have remained largely constant at about 3,500 HGVs.

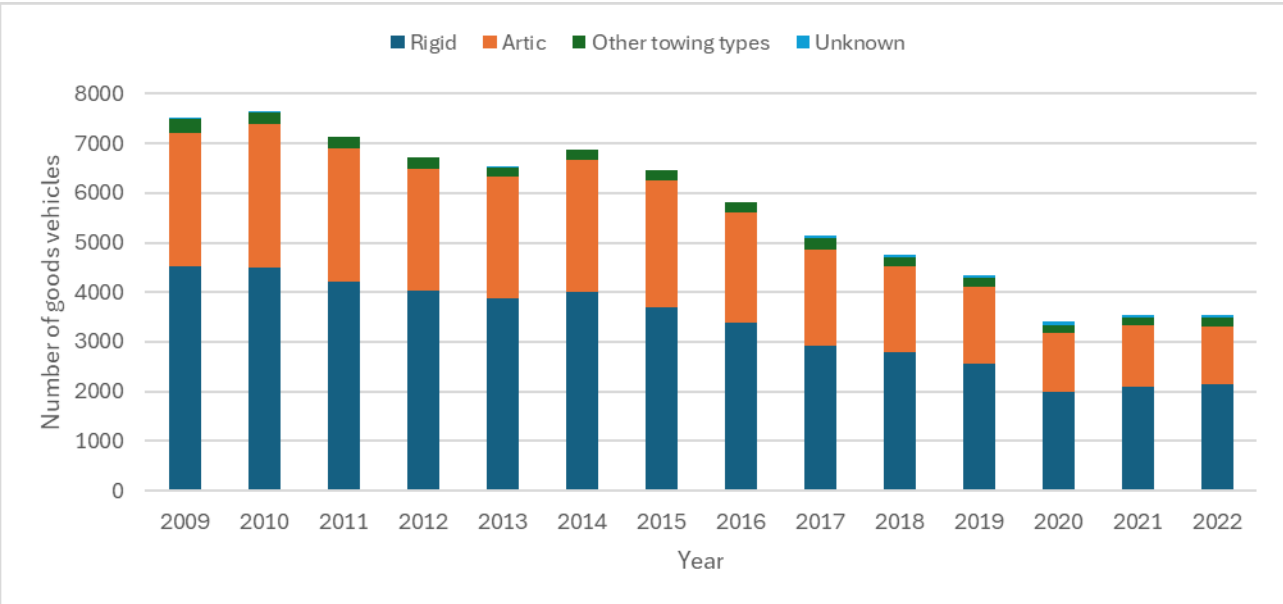


Figure 2-6: Number of heavy goods vehicles involved in road traffic accidents. Source: STATS19 database (2009-2022)

In comparison, the number of incidents involving FL vehicles (Figure 2-7) remained broadly level at approximately 75 per year until 2020, when there was a sharp reduction, again likely influenced by the COVID-19 pandemic and a reduction in demand for petroleum deliveries. Unlike the delivery volumes shown in Figure 2-5, there has not been a subsequent increase in collisions since that time. In 2022, there were 38 FL vehicles involved in accidents, roughly half the level seen prior to 2020.

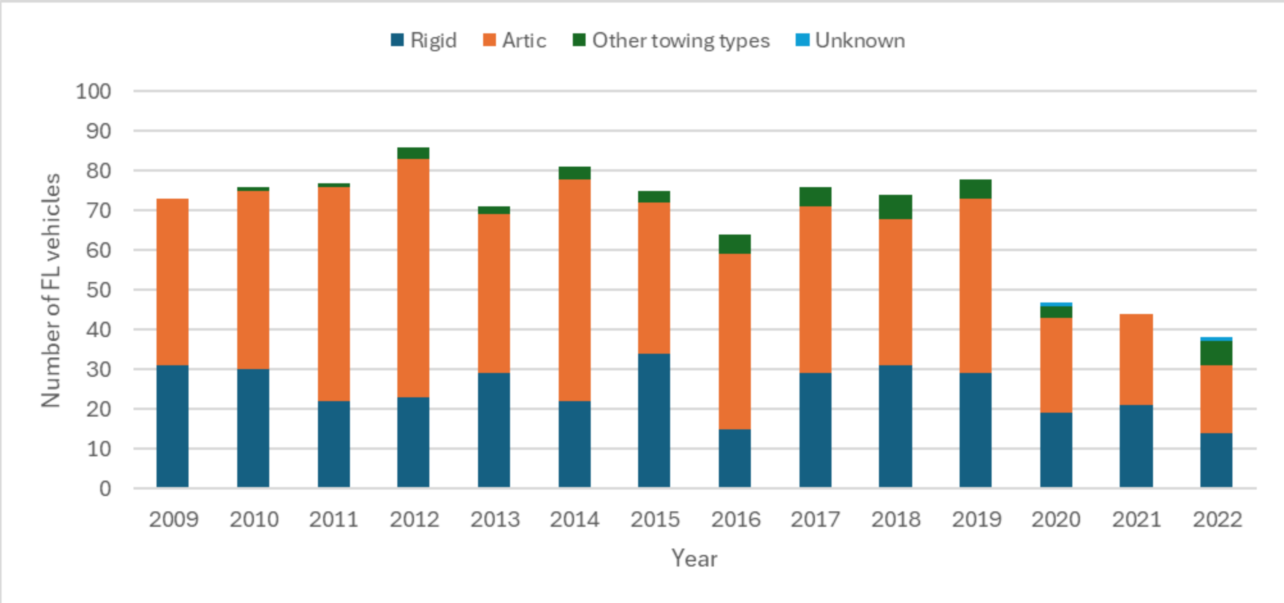


Figure 2-7: Number of FL vehicles involved in road traffic accidents.
Source: STATS19 database (2009-2018) & DVSA ADR certification database (2009–2022)

In general, the number of vehicles involved in collisions dropped substantially during 2020, likely as a consequence of Covid-19 lockdowns; a reduction of around 23% for all vehicles and 22% when considering only HGVs[8]. However, the rate for 'all vehicles' has rebounded to around 11% lower than 2019. For HGVs, it can be seen that the recovery has been somewhat less than for all vehicles, to around 19% lower than 2019. For FL tankers specifically, there has been no post-Covid recovery; instead, the reduction continued to around 50% of the 2019 figure. This is not explained by changes to exposure to risk (as measured by the total quantity of fuel deliveries), so the reason for the different pattern is unknown.

2.2.1.3 Incident types

Rollover incidents

Table 2-1 and Figure 2-8 show that the proportion of all rigid goods vehicles involved in a rollover collision has remained largely constant over the past 10 years with, on average, 2.8% of rigid HGVs in recorded collisions suffering a rollover. The proportion of articulated goods vehicles involved in a rollover was higher at, on average, 5.3%.

[8] Based on figures from RAS0101 DfT statistics table.

Table 2-1: Breakdown of rollover by rigid/articulated vehicles. Source: STATS19 database (2009-2022)

	All HGVs	FL vehicles
Rigid	2.8%	3.4%
Articulated	5.3%	5.7%

The numbers suggest that a greater proportion of FL vehicles were involved in rollover incidents, however the trend (Figure 2-8) shows there was considerable year to year variation. This was due to the smaller numbers involved. On average, there were three articulated FL vehicles involved in rollovers each year, but there were several years in which there were no reported rollover collisions.

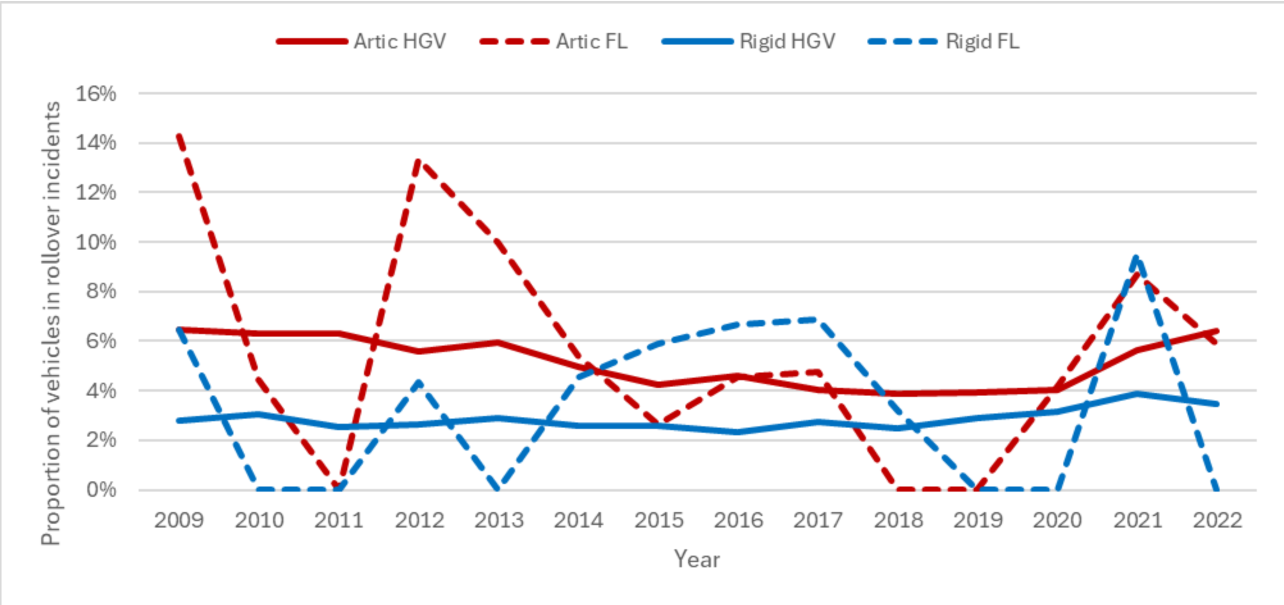


Figure 2-8: Proportion of vehicles involved in collisions involving rollover. Source: STATS19 database (2009-2022) & DVSA ADR certification database (2013-2022)

Again, unexpected post-Covid changes are apparent. For articulated vehicles, there has been a steady reduction in the absolute number of collisions involving articulated HGVs, but also the proportion of those that involved rollover has reduced from more than 6% to around 4% in 2020. (Knight and Dodd, 2019) considered this reduction to be consistent with expectations of the steady increase in market penetration of electronic roll stability controls within the fleet, even though the data does not prove that is the cause of the change. However, all this steady gain from 2009 to 2020 has been eliminated by a proportional increase from 2020 to 2022. In absolute terms, the number of articulated HGVs involved in rollovers dropped in 2019 but increased in 2021 and 2022 to a level slightly higher than 2019 (61 cases in 2019, 75 in 2022), but most of the difference in the percentage figure comes from the drop in the number of vehicles involved in collisions not involving rollover. That is, if the number of rollovers in 2022 was equal to 2019 at 61, then the percentage that rolled would have increased from 4.1%

in 2019 to 5.6% in 2022, instead of the observed 6.4%. The situation for FL vehicles is similar at 1-2 per year but, as stated above, the number of FL vehicles involved in all types of incidents (including non-rollover) has been considerably lower since 2020.

Collision incidents

For cases where, in STATS19, only two vehicles were involved in a collision and impact points were recorded for both vehicles, it was assumed that the two vehicles collided with each other. For collisions involving more than two vehicles, STATS19 did not contain enough detail about the sequence of events to confidently determine collision partners and so these cases were excluded from the following analysis. Furthermore, there were some single vehicle incidents, for which the HGV under analysis does not have an impact partner, despite an impact point being recorded in the database. These were also excluded.

For cases where the impact partner was known, Table 2-2 shows the proportion of all collisions with an FL vehicle, broken down by the impact point and the type of impact partner. Collision partners were grouped into the following categories:

- Heavy: e.g. HGV, bus, agricultural tractor
- Light: e.g. car, van
- Two-wheel vehicle: e.g. cyclist or motorcyclist
- Other: e.g. horse

Impacts with another heavy vehicle accounted for 8.7% of all collisions with a known impact partner. Collisions with small vehicles, such as cars or vans, make up the majority (80.7%) of collisions.

Table 2-2: Collisions with FL vehicles by impact point and collision partner. Source: STATS19 database (2009-2022)

Collision Partner	FL Front	FL Rear	FL Side	Total
Heavy vehicle	2.4% (n=14)	3.4% (n=20)	2.9% (n=17)	8.7% (n=51)
Light vehicle	37.8% (n=221)	12.0% (n=70)	30.8% (n=180)	80.7% (n=471)
Two-wheel vehicle	3.4% (n=20)	1.4% (n=8)	4.5% (n=26)	9.2% (n=54)
Other vehicle	0.7% (n=4)	0.0% (n=0)	0.7% (n=4)	1.4% (n=8)
Total	44.3% (n=259)	16.8% (n=98)	38.9% (n=227)	100% (n=584)

A comparison with all HGVs shows a similar pattern (Table 2-3). FL vehicles were involved in proportionally fewer collisions with two-wheel vehicles, which may reflect differences in the usage patterns of FL vehicles compared with other types of HGVs.

Table 2-3: Collisions with all HGVs by impact point and collision partner. Source: STATS 19 database (2009-2022)

Collision Partner	HGV Front	HGV Rear	HGV Side	Total
Heavy vehicle	4.0% (n=1,898)	2.6% (n=1,262)	2.9% (n=1,393)	9.5% (n=4,553)
Light vehicle	36.1% (n=17,242)	12.7% (n=6,077)	28.0% (n=13,358)	76.8% (n=36,677)
Two-wheel vehicle	3.7% (n=1,779)	1.8% (n=872)	7.3% (n=3,471)	12.8% (n=6,122)
Other vehicle	0.4% (n=181)	0.1% (n=71)	0.4% (n=179)	0.9% (n=431)
Total	44.2% (n=22,100)	17.3% (n=8,282)	38.5% (n=18,401)	100% (n=47,783)

Collisions with the front of the FL tanker

Between 2009 and 2022 there were 14 collisions recorded in STATS19 between the front of an FL vehicle and another "heavy" vehicle, an average of one per year. Of these, nine were impacts to the rear of the "heavy" vehicle, and three were front-to-front impacts with a "heavy" vehicle, as shown in Figure 2-9 below.

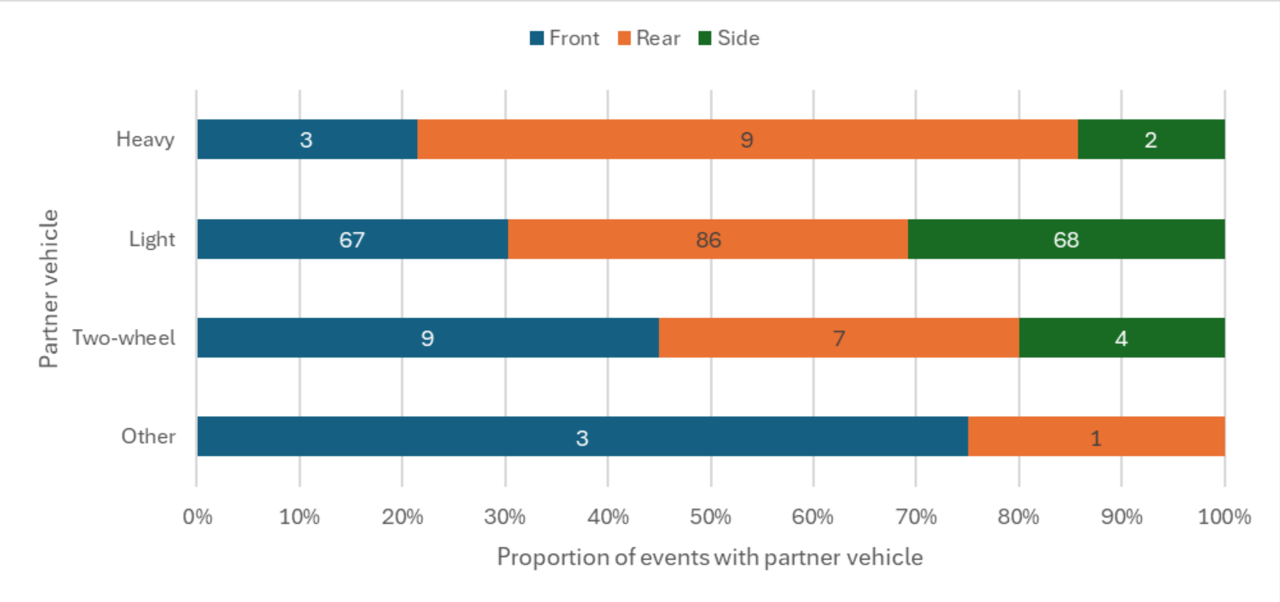


Figure 2-9: Breakdown of collisions between the front of an FL vehicle and another vehicle, by collision partner and impact location. Source: STATS19 database (2009-2022)

One front-to-rear collision between two heavy vehicles resulted in a pedestrian casualty. The incident involved the FL vehicle pulling away and hitting the rear of another stationary vehicle. In the STATS19 database, the pedestrian casualty was associated with the FL vehicles which, based on the limited information available, could imply that the FL vehicle hit the pedestrian as well as the other vehicle.

The 14 incidents resulted in 20 casualties (3 fatal, 4 serious, 13 slight). So, if it is assumed that the fatalities occurred in the most severe collisions, those most likely to damage the structure of a tank, this means there were just three such cases in 14 years, or approximately one every five years. Although infrequent, it is clearly a very severe crash type with 15% of all casualties being killed. This compares to all collisions involving an HGV, where 5% of all casualties are killed (based on figures for 2022, RAS 0601).

Collisions with the rear of the FL tanker

Similarly, for rear impacts to an FL vehicle, there were 20 collisions between the rear of an FL tanker and another "heavy" vehicle between 2009 and 2022, an average of 1.4 per year. These collisions resulted in 27 casualties (4 fatal, 5 serious and 18 slight), the majority of which were associated with the other vehicle involved in the collision.

The majority of the collisions (n=18) were impacts to/from the front of another heavy vehicle, as shown in Figure 2-10 below.

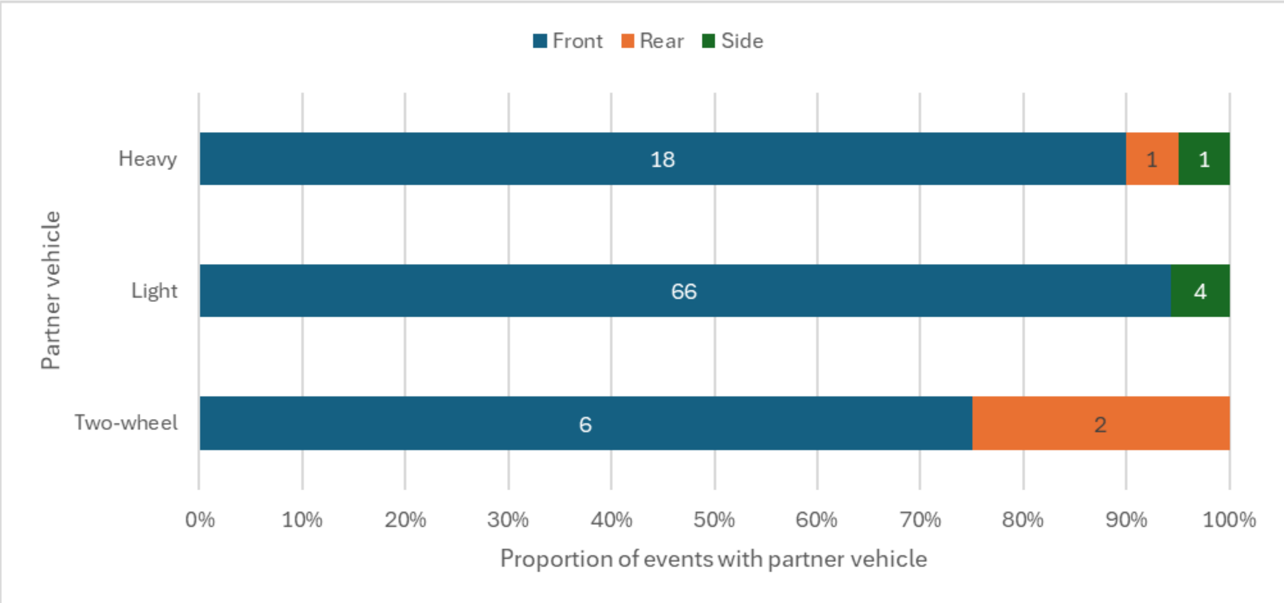


Figure 2-10: Breakdown of collisions between the rear of an FL vehicle and another vehicle, by collision partner and impact location. Source: STATS19 database (2009-2022)

2.2.2 *Expanded analysis of ADR collision reports*

(Knight and Dodd, 2019) studied reports made to DfT under the ADR, which requires operators to notify DfT where a serious collision occurs and there was an imminent risk of a loss of product (i.e. the release of dangerous goods)[9]. Six cases involving FL vehicles that occurred between 2014 and 2018 were provided by DfT, of which 5 involved an actual loss of product. An additional 6 cases were provided by DfT covering the period 2018 to 2024, but on closer examination, two of the cases were duplicates of those provided for the earlier study. In total, 10 cases were available from the period 2014 – 2024 via operators reporting to DfT under the ADR requirements.

In addition to this, DfT have been collecting reports from other agencies to assess the extent of under-reporting through the ADR mechanism. An additional 6 cases were identified through National Highways, the Environment Agency, and the Fire Service in 2022 alone, suggesting a significant level of underreporting.

Most cases had very little detail, but a summary of the incidents is provided below, together with some commentary on possible additional factors:

- 7 of 16 reports involved vehicles that had rolled over. Five of those seven cases specifically cited an icy road surface as contributory factor or direct cause. HGVs can rollover due to cornering forces alone, but when low friction is present it is more likely that the vehicle first lost directional control by slipping on the ice, then left the road and rolled over.
- One additional case was described as the HGV spinning out, resulting in the diesel tank rupturing. A spin in an HGV is more likely to occur where low road friction is present because in high friction conditions, they will often roll before sliding (if heavily laden). A spin alone is unlikely to cause tank rupture without a collision of some kind, either with another vehicle, roadside furniture, or with the ground after a rollover. It is possible, but not certain, that this incident involved a spin on ice, followed by a road departure and subsequent rollover.
- One front to rear collision was identified. This case is the primary reason for this study as described in more detail in Section 1.1.
- One additional frontal collision was identified, but in this case the vehicle left the road, drove into a stream and the frontal collision was with the bank of the stream.
- One case involved a vehicle colliding with and penetrating through or over the central reservation barrier onto the opposite carriageway. There was no report of a separate collision with a vehicle or rollover, but there was thought to be a release of product.
- Two cases did not involve a collision: one involved a fire, and the other was reported from a roadworthiness check.
- Three cases had no information at all about collision type.

[9] An incident is deemed reportable when the loss is 333 kg / 333 l or more if petrol, or 1 000 kg / 1 000 l or more if diesel (ADR 1.8.5.3)

2.2.3 *Comparison of STATS19 and ADR reporting levels*

One of the concerns raised by previous studies is the potential under-reporting of serious collisions involving ADR vehicles through the reporting scheme required by the ADR. In the context of this study, under-reporting was considered as a possible contributory factor in explaining the apparent discrepancy between simulation results and collision data. Specifically, that simulation suggested at least some tanks may be vulnerable to rupture from king pin loading in frontal collisions, and only one such incident involving a release of product was found from ADR reports

There are several different potential sources of information in relation to collisions, both generally, and those specifically targeting ADR vehicles. These are briefly summarised below along with their strengths and weaknesses in this respect:

- STATS19 – this is the national collision reporting system, populated mainly by police, but also local authorities. It covers only injury collisions in places considered public roads. It is generally considered that all fatal collisions are recorded on the system, but it is widely acknowledged that lower severity collisions may be substantially under-reported. DfT estimates (Road Accident Statistics table RAS4201) suggest that, in 2022, as few as 27% of lower severity casualties may have been recorded by STATS19. In the context of ADR analysis, a major limitation is that ADR vehicles cannot be separately identified.
- STATS19 linked with DVSA – is the technique used in this work to link DVSA records to STATS19 records via the Vehicle Registration Mark. This technique relies on the correct recording of VRM in both data sets. General experience with linking STATS19 to the DVLA wider registration data set, is that as many as one third of cases may fail to link correctly. If this is replicated with the DVSA data, which there is no way to prove, then it would lead to a potential under-estimate at all levels of severity.
- ADR reporting – ADR imposes a legal obligation on companies to report to DfT any serious incident if there is an immediate risk of (or actual) loss of, product, or if there is personal injury or fatality, there is material or environmental damage more than €50,000, or the authorities evacuate or close a route for more than three hours. This has the advantage of being very specific to ADR but the disadvantage that there are few details and few mechanisms for enforcing the requirement.
- DfT under-reporting investigation – in 2022, DfT undertook a special exercise to assess under-reporting by asking other agencies such as the Environment Agency, National Highways and the Fire Service to report incidents to them. This initiative was a one-off, not backed by any legal requirement, and was often based on first notifications with few technical details on the collision. Nevertheless, it identified substantial under-reporting via the ADR mechanism.
- Press reports – A fuel tanker collision involving a loss of product would be expected to be a newsworthy event such that internet press articles would exist. This can also help to identify any under-reporting, but there is no way to be sure every genuine collision is identified, and technical details are subject to misreporting. While a tanker collision is often easy to identify, the product it is carrying may not be. This could lead to non-flammable products being included in the sample.

As such, there is no absolute benchmark source of data that can be considered 'ground truth' in assessing the scale of under-reporting and imperfect assessments must be made by comparing different sources.

In respect of the ADR reports, the conclusions are clear. In a ten-year period, DfT were able to supply ten cases relevant to this study. In one year, in 2022, the extension of reporting to other Government agencies identified 6 relevant collisions. Numbers are too low for any kind of meaningful confidence, but in 2022 the ADR reporting process identified 2 cases of a total 8 identified through all agencies. Four of the eight cases were also identified in an internet search of press articles. That same search produced an additional 4 incidents, but only in one of those was there a report of product being released. In addition to this, STATS19 linked with DVLA reported one FL vehicle front to other heavy vehicle rear collision that was not reported via ADR, and one articulated FL tanker rollover that was not reported to ADR. However, both were slight injury collisions and there is no way of knowing whether there was an imminent risk of product release. So, it is not known whether these were officially reportable under ADR.

As such, with considerable uncertainty due to low numbers, it can be estimated that the ADR reporting process captures between around 18% and 25% of the incidents involving flammable liquids that it should cover.

There is also strong evidence that press reports would under-estimate the scale of relevant collisions. Only one of the FL vehicle front-to-rear or rollover incidents recorded in STATS19 (46 in total) was identified in press reports on the internet. Although many of those STATS19 incidents would not have involved risk of product release, it might have been expected that they were more newsworthy.

Other than the original incident for FL tanker front to other heavy vehicle rear that triggered this study, there were no confirmed similar incidents identified in new ADR reports, the DfT under-reporting investigation, or press reports. Press reports did identify a frontal impact into a ditch which may have some similarities, and one further "HGV to tanker" collision that resulted in frontal damage to at least one vehicle, but it was unclear which vehicle sustained the damage.

As such, although under-reporting is clearly an issue and overall risks can be considered higher than suggested by these data, no specific evidence has been found of under-reporting of front to rear collisions between FL tankers and other heavy vehicles. It remains unclear whether this is evidence that there is no such under-reporting or simply a lack of evidence.

2.2.4 Summary of findings

Collisions involving the front of FL tankers impacting the rear of other heavy vehicles are rare. Based on the figures available for the period 2014-2024 there was:

- An average of one incident of any kind per year,
- An average of one incident every five years that was sufficiently severe to cause a fatality (a possible proxy for high crash forces through the king pin risking tank rupture),
- Only one incident in the 10-year period where tank rupture was confirmed.

Under-reporting is a significant issue, both within STATS19 (for non-fatal collisions) and in the ADR reports. Based on average under-reporting rates within STATS19, the 11 non-fatal incidents in 14 years could in fact be 41 cases. However, the fatal cases where forces on the tank might reasonably be expected to be highest, would remain at 3. On this basis, three such incidents per year might be expected, with one fatal incident every 5 years.

ADR reports may only be identifying somewhere in the region of 15%-25% of relevant cases. As such, the one confirmed case in ten years, could in fact be 4 to 7 cases in ten years or, very roughly, one per year to one every 3 years. However, with no other confirmed incidents, it is possible that the base rate is less frequent than one in ten years – i.e. it is quite possible that no further confirmed incidents arise in the next 10 years, in which case the base rate becomes one every 20 years.

In summary, it is unquestionable that this crash type is rare, and the exact frequency is extremely uncertain due both its rarity and the under-reporting inherent in the available data sources.

- In the best case, a repeat of the central collision type could, on average, occur substantially less often than once every 10 years.
- In the worst case, it could occur as often as once every year or two
- A best estimate may lie somewhere in the region of once every 5 years

With all these estimates, the random nature of rare events could render averages misleading. For example, there could be no incidents for 20 years, followed by 3 occurring in a single year.

It was also noted that the number of HGVs involved in collisions reduced by 22% during 2020 as a likely consequence of Covid-19 lockdowns, but there has been a small increase since then to a level around 19% lower than 2019. For FL tankers specifically, there has been no post-Covid increase, but a continued reduction to around 50% of the 2019 figure. This is not explained by changes to exposure to risk (as measured by fuel deliveries), so the reason for the different pattern is unknown. However, it could possibly be related to the benefits of new safety technologies being realised by rapid fleet penetration into the FL tanker fleet.

2.3 Activity 2: Fleet analysis and understanding of tanker design

The objective of this activity was to understand the variation in tanker semi-trailer design within the current GB fleet; in particular, whether the tank is self-supporting behind the king pin, and the variation in the design of the king pin support structure.

To meet this objective, this activity performed work to:

- Identify the common makes of tanker semi-trailers in the GB fleet.
- Develop an understanding of the range of tanker design in the GB fleet, their main design requirements and influencing factors.

which is reported in the sections below.

2.3.1 Common makes of tanker semi-trailers in the GB fleet

In GB, to be used on the public road, ADR tankers must pass a supplementary test performed annually, normally in conjunction with the annual roadworthiness (MOT) test. The DVSA maintain records for these supplementary tests, which includes the expiry date.

From these records, DVSA supplied the project with a list of all semi-trailer tankers with a test expiry date in 2023 to give a snapshot of the road legal fleet in 2022. For each individual tanker, the following data were also included:

- Semi-trailer manufacturer.
- Tank product list.
- Tank manufacture year.

The focus of the project was petroleum road fuel tankers, i.e. class 3 Flammable Liquid (FL) tankers and specifically ones that transport road fuels. Examination of the product list found that many of the tankers listed transported dangerous goods other than fuel. Therefore, to determine the most common petroleum road fuel semi-trailer tanker makes, the product list was filtered for the following:

- UN 1202: Gas oil or diesel fuel or heating oil (light)
- UN 1203: Gasoline or petrol or motor spirit
- UN 1223: Kerosene

It should be noted that product list was as declared to DVSA and contained named standard lists, such as Zurich Engineering product list PS22-0033-03, the content of which was not readily known. These lists may have contained the items filtered for, e.g. UN 1202, but because they were effectively hidden in the product list, the filtering performed would not have identified the associated tankers.

The top 20 most common petroleum road fuel semi-trailer tankers in the GB fleet that were identified are shown in Table 2-4.

Table 2-4: Twenty most common makes of petroleum fuel semi-trailer road tankers in GB fleet (2022)

No.	Semi-trailer make	No.	% of All (1776)
1	LAKELAND	580	32.7%
2	LAG	362	20.4%
3	COBO	353	19.9%
4	MAGYAR	98	5.5%
5	GRW	84	4.7%
6	NORTHERN	80	4.5%
7	HEIL	49	2.8%
8	FELDBINDER	39	2.2%
9	TASCA	37	2.1%
10	RTN	26	1.5%
11	CLAYTON	20	1.1%
12	CROSSLAND	9	0.5%
13	VAN HOOL	6	0.3%
14	KASSBOHRER	5	0.3%
15	RTN LTD	4	0.2%
16	ACERBI	4	0.2%
17	GENERAL TRAILERS	3	0.2%
18	BURG	2	0.1%
19	STOKOTA	2	0.1%
20	INDOX	2	0.1%
TOTAL		1765	99.4%

Summing the makes produced by the Road Tankers Northern (RTN) group, which includes Lakeland, Northern, RTN and Clayton Vallely, shows that it forms by far the largest segment of the fleet at 40% (No. = 713). Cobo tankers form a noticeable segment of the fleet, at approximately 20%.

Because the year of semi-trailer manufacture was also included in the data provided, it was possible to examine the age distribution of the petroleum fuel semi-trailer tanker fleet. This is shown below for the top three most common and all makes (Figure 9). It is seen that fewer tankers were sold in 2022, most likely a consequence of the Covid-19 pandemic, and that the Cobo proportion of the market has increased in recent years since a low in 2018.

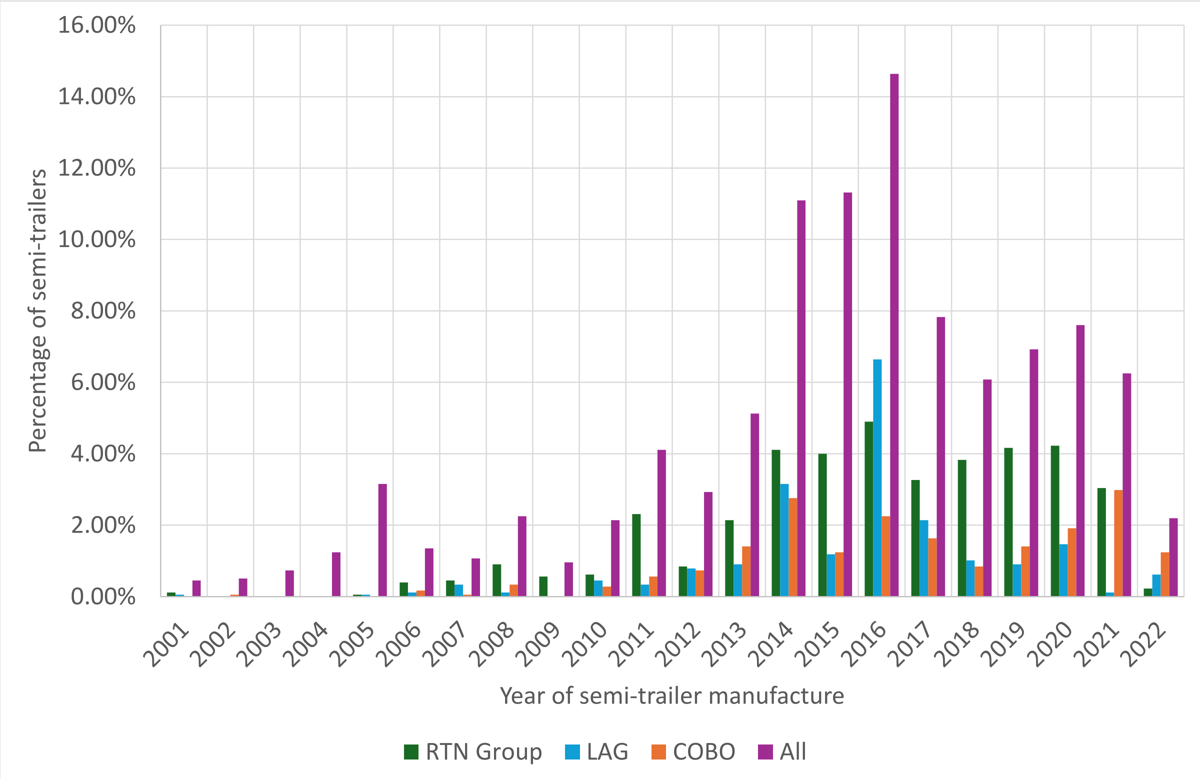


Figure 2-11: Age distribution of petroleum road fuel semi-trailer tankers in GB fleet for top three most common and all makes

2.3.2 Range of tanker semi-trailer designs in the GB fleet, their design requirements and influencing factors

From a literature review and interviews with tanker manufacturers and appointed inspection bodies (AIBs), the main regulatory requirements which control tanker semi-trailer design, current tanker design differences, and drivers for those design differences, were identified. These are reported in the sub-sections below. Following this is a sub-section focused on the king pin and supporting sub-assembly and a discussion sub-section.

2.3.2.1 Regulatory requirements

The main regulatory requirements identified which control and influence semi-trailer design were weight, length, height of centre of gravity, and tank design stress under static and dynamic loading in normal conditions of carriage and prescribed minimum stresses (ADR 6.8.2.1.1). Further description of these requirements is given below:

Weight:

- Maximum gross weights permitted for goods vehicles are set out in the Road Vehicles (Construction and Use) Regulations 1986 as amended (C&U) and the Road Vehicles (Authorised Weight) Regulations 1998 as amended (AWR)[10].
    - The maximum weight for articulated HGVs with 6 axles is 44,000kg.
    - These regulations require that for operation of articulated HGVs with a gross weight above 40,000kg, the drive axle(s) must not exceed 10,500kg and have road friendly suspension and each part of the combination must have 3 axles and the trailer must have road friendly suspension.

Length:

- Maximum overall lengths permitted for goods vehicles are set out in the Road Vehicles (Construction and Use) Regulations 1986 as amended (C&U)[11].
    - These regulations require that the overall length of an articulated (tanker type) HGV must not exceed 16.5 m and the length of the semi-trailer (king pin to rear) must not exceed 12 m.

Height of centre of gravity above ground:

- The ADR Chapter 9.7.5 'Stability of tank-vehicles' requires that:
    - 9.7.5.1 The overall width of the ground bearing surface (distance between the outer points of contact with the ground of the right-hand tyre and the left-hand tyre of the same axle) of the axle with the greatest width shall be at least equal to 90% of the height of the centre of gravity of the laden tank vehicle. In an articulated vehicle, the mass on the axles of the load carrying unit of the laden semi-trailer shall not exceed 60% of the nominal total laden mass of the complete articulated vehicle.

      Note: Given a maximum vehicle width of 2.5 m, this sets a maximum laden centre of gravity height of 2.78 m.
    - 9.7.5.2 In addition, tank-vehicles with fixed tanks with a capacity of more than 3 m^3 intended for the carriage of dangerous goods in the liquid or molten state tested with a pressure of less than 4 bar, shall comply with the technical requirements of UN Regulation 111 for lateral stability, as amended, in accordance with the dates of application specified therein. The requirements are applicable to tank-vehicles which are first registered as from 1 July 2003.

      Note that the petroleum road fuel tankers being considered have a capacity more than 3 m^3 and thus must comply with UN Regulation No. 111.
- UN Regulation No. 111 requires that one of the following is met for rollover stability:

[10] https://www.gov.uk/government/publications/hgv-maximum-weights/hgv-maximum-weights

[11] https://www.gov.uk/government/publications/maximum-length-of-vehicles-used-in-great-britain/maximum-length-of-vehicles-used-in-great-britain

- Calculation method: withstand 4 m/s^2 lateral acceleration **OR**
- Tilt table method: withstand tilt angle 23 degrees.

Tank design stress under static and dynamic loading in normal conditions of carriage:

- The ADR Chapter 6.8.2.1.1 requires that tank shells, their attachments and their service and structural equipment shall be designed to withstand without loss of contents (other than quantities of gas escaping through any degassing vents) static and dynamic stresses in normal conditions of carriage and prescribed minimum stresses as specified in other sections of the ADR.
  - These requirements are also detailed in EN 13094:2022, para 6.3.2 'dynamic conditions'. For loads in the longitudinal direction, it is specified that the design stress for tank shells and their attachments is not exceeded for longitudinal accelerations (2 g) in the direction of travel.

    It is further specified that for the front-end (i.e. the king pin support structure), that only the maximum mass of the substance carried in the first (front) compartment shall be considered. However, this requirement is caveated for trailers without a longitudinal framework upon which the tank is supported, i.e. self-supporting tankers – see section 3.2 below. For this case, it states that for front attachments (i.e. the king pin support structure), the maximum design mass of the trailer shall be deemed to act where the coupling device attaches to the tank.

  It should be noted that the requirement above was developed to account for dynamic loading from the fluid in the tank experienced in potential in-service braking type conditions – for rail transport, this requirement is not specified because of low braking decelerations experienced by rail vehicles. Longitudinal loads far greater than 2 g are likely to be experienced in frontal collisions, especially ones with another HGV, which is the subject of this project. Typical load requirements related to frontal collisions specified in regulations include:

  - UN Regulation No. 80 'Strength of seats of buses' (M_2 and M_3 category vehicles); dynamic tests of seats with crash pulse peak between 8 g and 12 g, average 6.5 g to 8 g.
  - UN Regulation No. 100 'Electrical safety'. Mechanical shock of Rechargeable Energy Storage System (REESS), i.e. Battery. For large buses and HGVs (M_3 and N_3 category vehicles), dynamic test with crash pulse peak between 6.6 g and 12 g.
  - UN Regulation No. 110 'Specific components for CNG and/or LNG and their installation on motor vehicles' Strength of fuel container attachments. For large buses and HGVs (M3 / N3 category vehicles) withstand load of 6.6 g without damage occurring.
  - UN Regulation No. 134 'Hydrogen safety' Fuel system integrity (e.g. tank attachments). For large buses and HGVs (M3 / N3 category vehicles) withstand load of 6.6 g and tank remain attached to vehicle at a minimum of one attachment point.

2.3.2.2 *Tanker design differences*

The main tanker design differences that might be relevant to the failure of the conical / taper section behind the king pin assembly / support structure seen in the collision investigation, are described below.

The first is whether the tanker is supported or self-supported in the region of the conical / taper section at the bottom of the tank behind the king pin assembly / support structure, as illustrated in the examples shown below (Figure 2-12).

Supported

Self-supported

Figure 2-12: Examples of semi-trailer tanker designs which are supported and self-supported in the region of the conical / taper section behind the king pin assembly / support structure (indicated with red circle)

It is interesting to note that, for both tanker examples, the rollover damage protection device along the top of the tank provides longitudinal support for the tank in the region of the conical section. However, although this device is required by EN 13094 para 6.14 'Protection of service equipment mounted on the tank top', several designs are permitted, some of which, such as rollover bars, may not provide longitudinal support as in the examples shown.

The second design difference is related to the design of the joint between the tank shell and the tank partitions and baffles. There are two main types of design, referred to as banded and stuffed (Figure 2-13). For the banded design the tanker shell is discontinuous and not connected (welded) directly to the partition / baffle. Instead, an extrusion acts as an interface with the tanker shell welded circumferentially to both sides of the extrusion and the partition / baffle similarly welded inside to the middle of the extrusion. For the stuffed design, the tanker shell is connected directly to the partition / baffle with a lap joint.

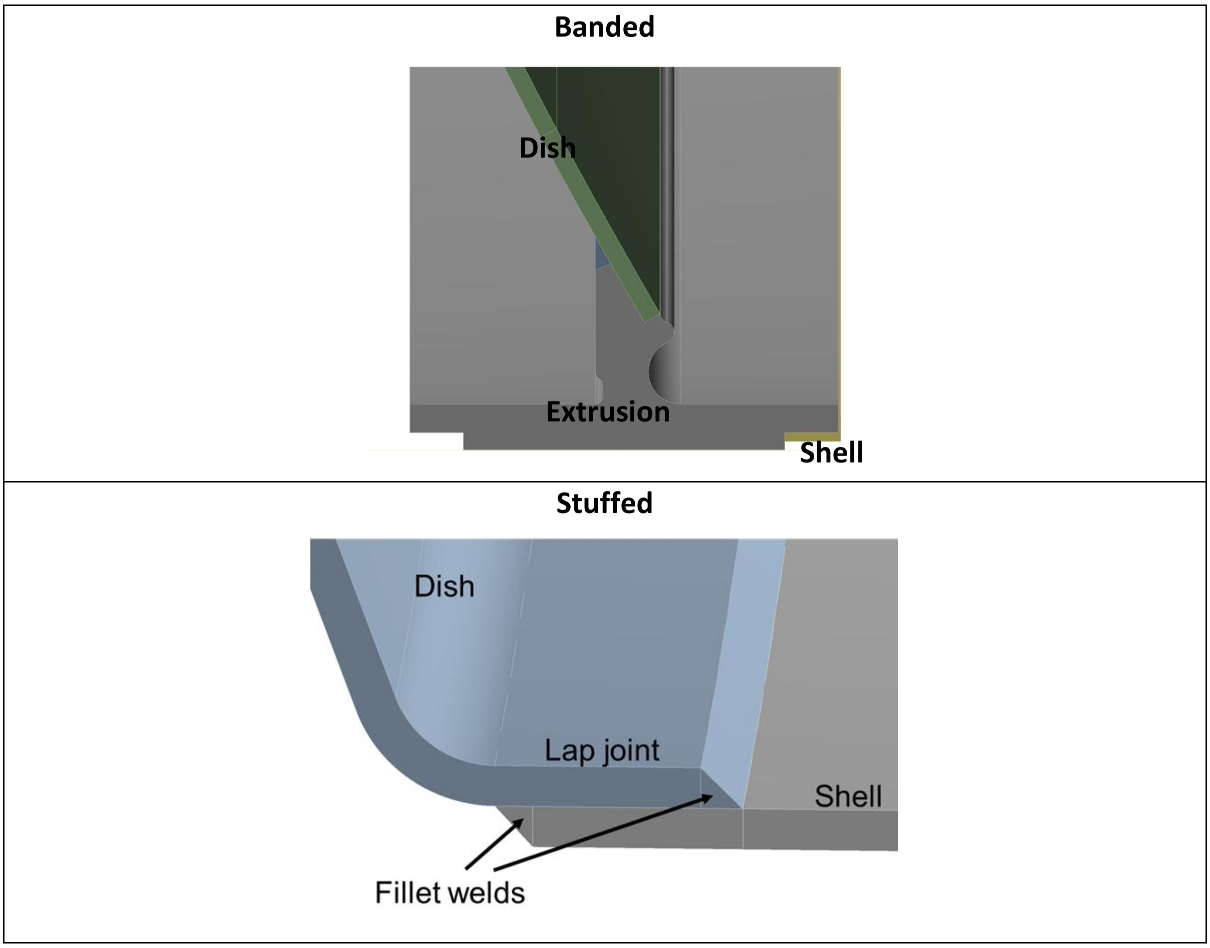


Figure 2-13: Examples of banded and stuffed tanker shell to partition / baffle joint designs

2.3.2.3 Main design influences

The main drivers of semi-trailer tanker design include desires for:

- The largest capacity possible, which translates to a low unladen weight. This is one of the main reasons why road fuel tankers are constructed from aluminium.
- A low centre of gravity height for good rollover stability.

These desires are constrained substantially by regulatory requirements, particularly those related to gross weight, axle weight, length and centre of gravity height.

The main reason for a tanker design which is self-supporting in the area behind the king pin assembly / support structure is that it allows for a higher capacity compartment in this area because the bottom of the conical section compartment can be lower without causing interference into the space required for the rear of the tractor unit when the vehicle turns. In turn this can help allow:

- Design of higher capacity trailers with a low centre of gravity because the fuel in the conical section compartment has a lower centre of gravity.
- Design of a shorter trailer overall, because of greater fuel carrying capacity in the forward part of the tanker. This in turn can help improve the manoeuvrability of the tanker; for example, for delivery to fuel stations with tight access. An example of a semi-trailer with such a design is the Lakeland Maxivator[12].

2.3.2.4 King pin and supporting assembly

Most manufacturers purchase the king pin (Figure 2-14), with an appropriate rating according to UN Regulation No. 55 'Provisions concerning the approval of mechanical coupling components of combinations of vehicles', from a supplier such as Jost. The supplier provides recommendations as to how the king pin should be welded to the bed plate (sometimes called skid plate).

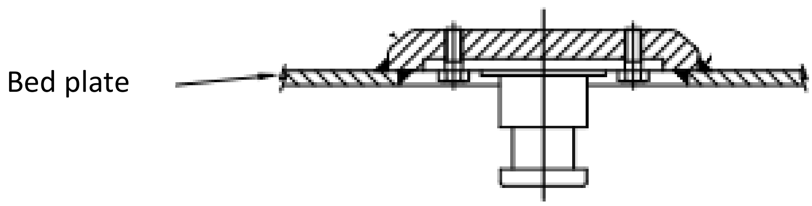


Figure 2-14: King pin and associated bed plate

Both the king pin and associated bed plate are usually made from of high tensile steel (e.g. grade S55). The bed plate has a thickness of about 8 mm to 12 mm, often 10 mm, and is supported by a structure which can be aluminium. The bed plate is bolted to the aluminium raft structure which connects to the tank shell. The bolts enable easy replacement of the king pin and associated bedplate when required because of wear. The raft structure is custom designed for each different tanker design. Load paths from the king pin into the tank are an important consideration for the raft design. To spread loads into the tank shell structure, often saddles and gussets are used as shown in Figure 2-12. Also, from observation of alignment of structures, it appears that designs often use partitions / baffles to try and spread loads further. For supported type designs the raft structure connects with the longitudinal support structure which in turn connects to the tank shell. For self-supported type designs the raft structure connects directly to the saddles / horns and the tank. For this type of design,

[12] Lakeland / RTN Maxivator: https://www.rtnltd.co.uk/new-tankers/semi-trailers/

the length of the raft structure is seen to vary considerably for current designs, from about 1.8 m to about 2.5 m.

Previous work in this project[13] suggests a high likelihood of tank failure, in particular for shorter king pin assembly raft lengths. In principle, if all other factors are equal, a shorter raft assembly length will result in higher moment loads into the tank and potentially a higher likelihood of tank failure. However, the previous work did not take into account the different load paths into the tank for these different designs, which could possibly offer an explanation as to why particular designs may or may not be more susceptible to tank failure than others. Modelling work was performed (see Section 2.4) to investigate this further.

2.3.2.5 Discussion

In the baseline collision referred to in the Section 2.1.1, the front of one fuel tanker struck the rear of another fuel tanker and there was a ADR reportable level[14] release of flammable liquid from the striking tanker. The lead tanker, which was almost stationary, was struck at the rear of the tank by the following tanker at an impact speed of 48 km/h and with an overlap of approximately 75%.

The impact caused the rear bumper of the struck tanker to be heavily deformed but, based on the photographs, only minor markings and minimal deformation was evident to the rear of the tank shell which maintained its integrity. In contrast, the tractor unit of the striking tanker sustained extensive damage across the full width of the front of the vehicle, causing the cab to be crushed and pushed back on the chassis, although it did not appear to substantially under-run the struck tanker.

It was evident that the rear of the cab was modestly damaged in a collision with the front dish of the tank shell. However, the main damage to the tank shell which caused the release of flammable liquid was in a region behind the king pin assembly / support structure. The mechanism for this damage appeared to be loading via the king pin assembly, which would have formed the main load path for decelerating the mass of the trailer. As illustrated in Figure 2-15, the inertia of the loaded tank would have acted through the centre of mass, which is at a greater height than the king pin providing the opposing force. This would have created a force couple and a rotational moment, resisted by the mass of the vehicle acting downward. It appeared that the net effect of the forces applied by the trailer's own mass and the mass of the load it contained, potentially compounded by the bucking of the rear of the tractor unit in a concertina effect, caused the tank to deform and buckle in the region just behind the king pin assembly and the rotational moment caused the front section to bend downward causing tearing of the tank shell. The tank ruptured in the region of this

[13] Edwards et al. 2023. 'Research on performance test procedures for petroleum road fuel tankers: Summary report'. TRL Published Project Report PPR2027. https://www.trl.co.uk/uploads/trl/documents/PPR2027-Research-on-performance-test-procedures-for-road-fuel-tankers---summary-report--v-final1.1-041023.pdf

[14] An incident is deemed reportable when the loss is 333 kg / 333 l or more if petrol or 1,000 kg / 1000 l or more if diesel (ADR 1.8.5.3)

deformation, above and immediately to the rear of the king pin assembly / support structure, and almost all of the fuel in the front compartment of the tanker (c. 7,000 litres) was released.

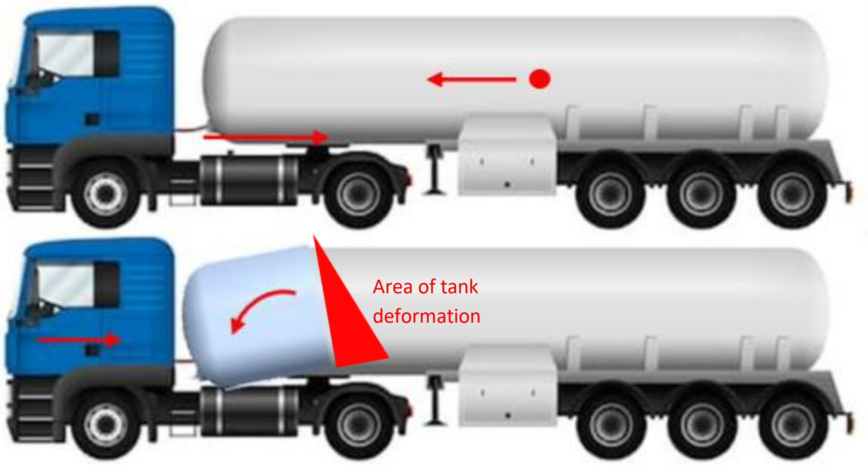


Figure 2-15: Illustration of crash loading of striking tanker through king pin assembly in tanker front to rear collision and the resulting deformation (Top: before impact, Bottom: after impact)

From the sub-sections above, it is understood that of the type of design where the semi-trailer is supported would have more strength in the conical / taper area where the semi-trailer in the collision was seen to buckle. It is also understood that for lateral deflections the type of design with banded shell to partition / baffle joints is stiffer than for a stuffed design and thus, should be greater able to resist the moment applied in the baseline collision.

Initial thoughts from this are that it can be simply concluded that, assuming other factors are equal, the order of the different types of design to better withstand the loading seen in the baseline collision are likely to be as follows:

- Supported tanks.
- Self-supported tanks with banded joints.
- Self-supported tanks with stuffed joints.

It should be noted that the semi-trailer that failed in the baseline collision was self-supported with stuffed joints.

Other likely influencing factors include:

- Length of raft structure and difference between centre of gravity and king pin height which will affect the moment forces.
- Design of load paths from king pin into tank structure, e.g. use of partitions / baffles.
- Support from the roll over protection / vapour capture device structure on the top of the tank.
- Conical / taper section length.
- Partition dish / baffle design in region of the conical / taper section, convex or concave relative to the rear of tanker (Figure 2-16).

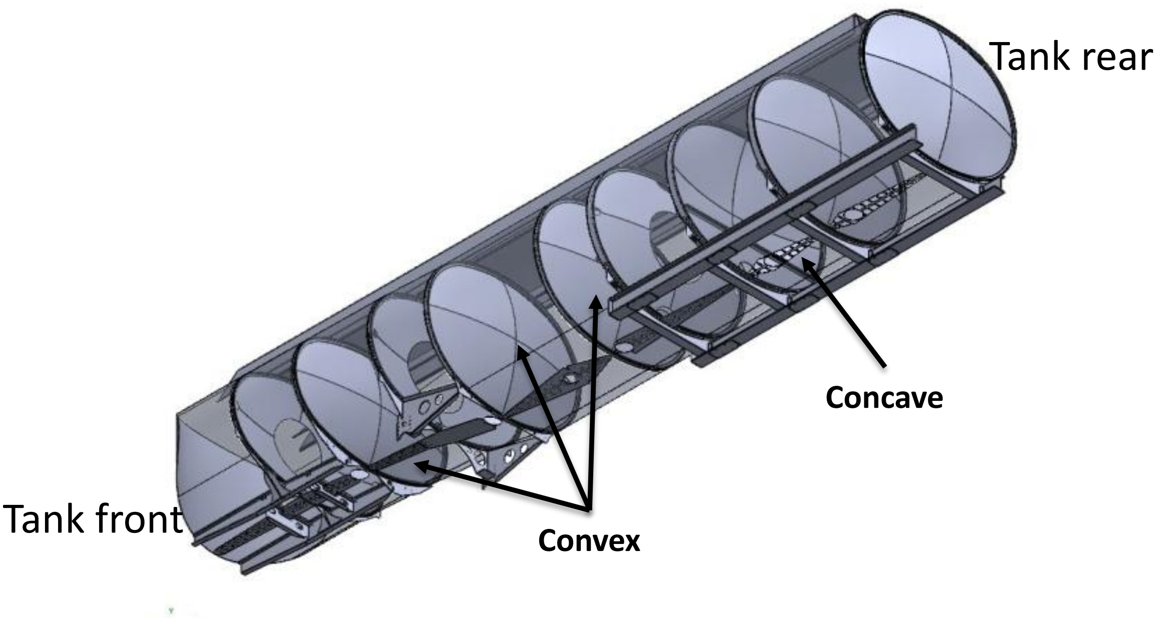


Figure 2-16: Illustration of convex and concave partitions relative to rear of tanker

Convex partitions will push tank shell out and concave partitions pull it in when loaded by fluid pushing forward in the compartment as in a frontal impact. Pulling the shell in is likely to encourage buckling behaviour and thus concave partitions could contribute to this.

2.3.3 Summary of findings

- The RTN group of tankers form the largest segment of the petroleum road fuel semi-tanker fleet ~ 40%. The make of semi-trailer tanker involved in the baseline collision forms a noticeable segment of the fleet ~ 20%.
- The vast majority of the GB fleet of petroleum semi-trailer road fuel tankers have a self-supported type design.
- Key regulatory requirements which influence semi-trailer design include:
    - Weight, maximum gross and axle weights.
    - Length, maximum.
    - Height of centre of gravity above ground, maximum.
    - Design stresses for in-service loads, maximum.
- Typical regulatory requirements for crash pulse loads for frontal impacts are between 6.6 g and 12 g.
- The main semi-trailer design differences likely to influence performance in frontal collisions include:
    - Self-supported or supported,
        - Note, the main reason for self-supported design is to allow higher capacity compartment in area just behind fifth wheel which in turn can

help allow design of higher capacity trailer with low centre of gravity and/or design of a shorter semi-trailer.

- Shell to partition / baffle joint design banded or stuffed.

- Other potential influencing design differences include:
  - Length of raft structure associated with king pin.
  - Design of load paths from king pin into tank structure, e.g. extent of support / raft structure and use of partitions / baffles.
  - Conical / taper section length.
  - Partition / baffle design in region of conical / taper section, convex or concave relative to the rear of tanker.
  - Support from the roll protection / vapour capture device structure on the top of the tank.

2.4 Activity 3: Finite Element (FE) modelling

The objective of this activity was to compare and contrast the performance of generic tanker semi-trailer designs across a range of collision conditions, including one representative of the ‘baseline collision’. The design range included a semi-trailer design representative of the semi-trailer involved in the ‘baseline collision’.

To achieve this objective the following work was performed:

- Build of three generic tanker semi-trailer FE models.
- LS-Dyna3D explicit crash modelling of baseline collision and parameter sweeps.
- Linear buckling modelling.
- Non-linear buckling modelling.

The LS-Dyna 3D software is a powerful, advanced simulation software used for performing complex, nonlinear, 3-dimensional dynamic analysis of structures, primarily focused on high-impact events like crashes, explosions, and penetrations. It utilises an explicit non-linear finite element method to accurately model the behaviour of materials under extreme loading conditions in which materials are deformed into their plastic regime and fail. It is considered an industry standard for crashworthiness simulations, particularly in the automotive industry, and is now part of the Ansys software suite, originally developed by Livermore Software Technology Cooperation (LSTC) as ‘Dyna3D’.

Linear (elastic) buckling analysis is used to identify the buckling modes of structures and loads at which they occur. This type of analysis assumes that loads to induce buckling occur well below what is required to cause compressive yielding, and the buckling is of a bifurcation type, i.e. there is a rapid transition from axial loading response to a lateral response, which is usually catastrophic. For linear buckling analyses, the FE solver performs an Euler type of calculation and extracts eigenvalues to determine what scaling factors to the nominal static load applied will cause the critical buckling mode shapes (eigen vectors). The lowest scaling factor usually indicates the most likely buckling mode at failure.

Non-linear (elastic / plastic) buckling analysis is also used to identify the buckling modes of structures and loads at which they occur. However, unlike linear buckling analysis, generally, it can also be used to analyse the post buckling behaviour of the structure.

It should be noted that the advantage of the LS-Dyna3D crash modelling compared to the linear and non-linear buckling analyses is that it is much more representative. For example, fuel movement and its structural loading was included in the model, but its disadvantage is that it is much more expensive than the linear and non-linear modelling. The linear and non-linear modelling were included in the work programme because they could identify buckling loads and modes and thus could potentially offer a more cost-effective route to implement regulatory change, assuming it was found necessary.

The results of the FE modelling work are described in the sub-sections below, structured as follows. The first sub-section describes the semi-trailer models, details of their build and where the information required to build them was obtained from. The second sub-section describes the analysis of the baseline collision and the development of the boundary conditions for the crash modelling with the LS-Dyna3D software. The next subsections detail the results of the LS-Dyna 3D crash simulations, including the parameter sweeps, the linear buckling simulations, and the non-linear buckling simulations, respectively. This is followed by the final sub-section, which gives a summary of the main findings of the FE modelling work.

2.4.1 Build of semi-trailer FE models

This section is divided into three parts: the first describes the semi-trailer models built, the second describes the information required to build them and the third how the fuel load was modelled.

2.4.1.1 Semi-trailer models

The fleet analysis work (see Section 2.3) found that the main design differences likely to influence performance in frontal impact included:

- Whether self-supported or supported.
- Whether shell to partition / baffle joint design banded or stuffed.
- Length of raft structure associated with king pin.
- Design of load paths from king pin into tank structure, e.g. extent of support / raft structure and use of partitions / baffles.

The analysis noted that whether self-supported or supported was likely to be the main influencing factor. Based on a desire to investigate the differences in performance for self-supported and supported semi-trailer tankers, and to compare performances of typical semi-trailer tankers with the semi-trailer tanker involved in the 'baseline collision', three semi-trailer FE models were built as follows:

#1: typical example of self-supported banded design

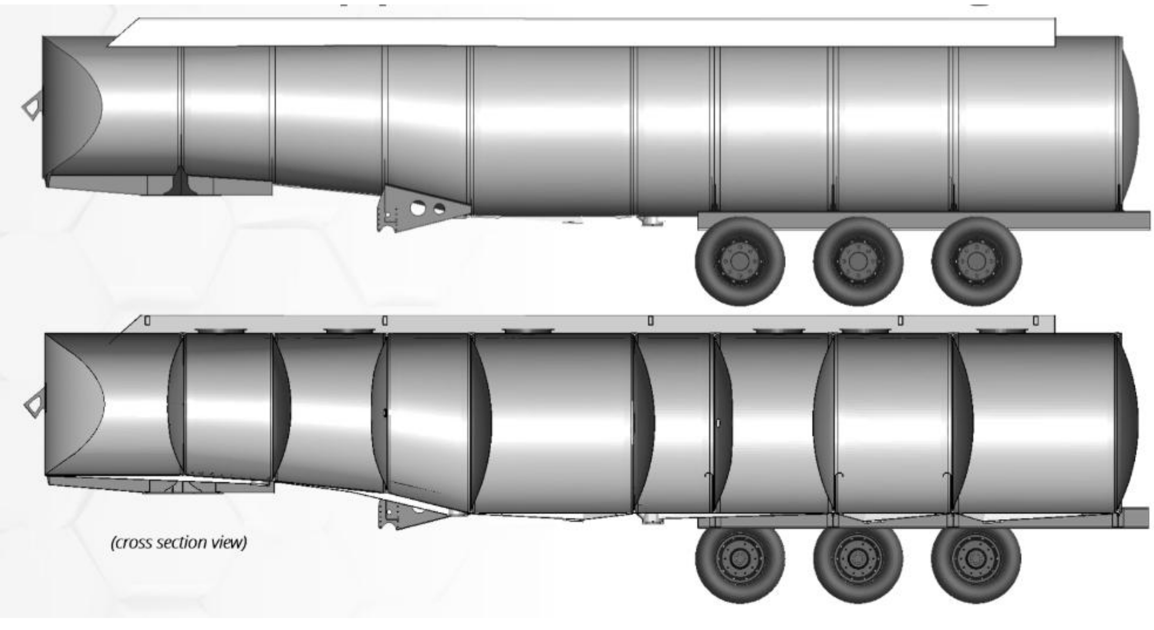


Figure 2.1: FE model of typical self-supported banded design semi-trailer

#2: typical example of supported stuffed design

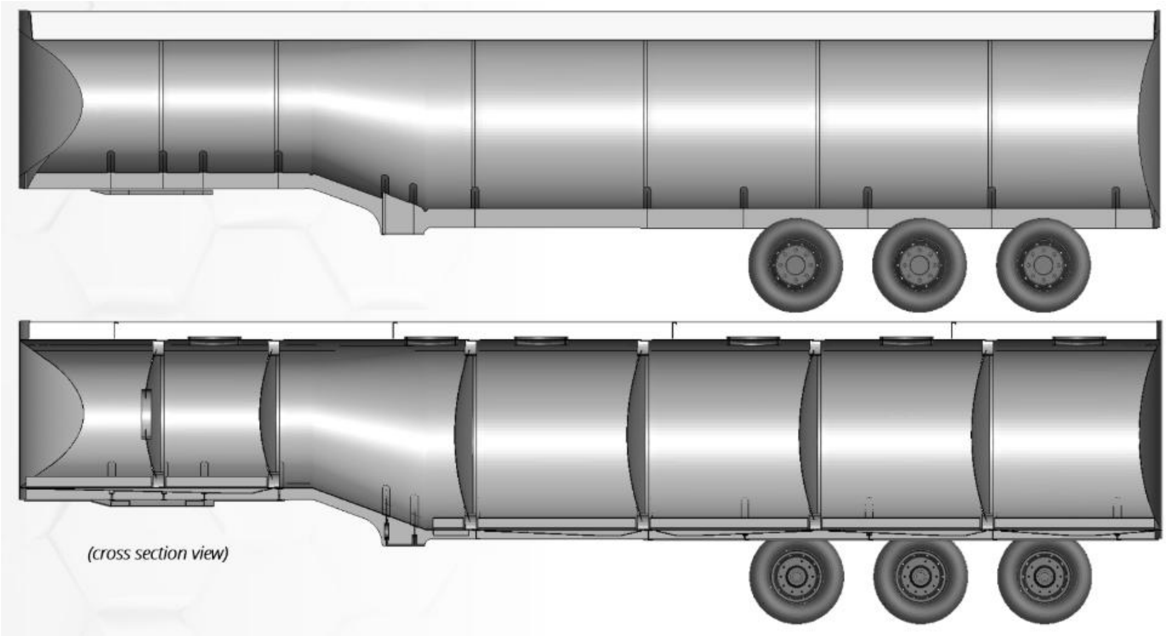


Figure 2.2: FE model of typical supported stuffed design semi-trailer

#3 representative model of self-supported stuffed tanker involved in baseline collision

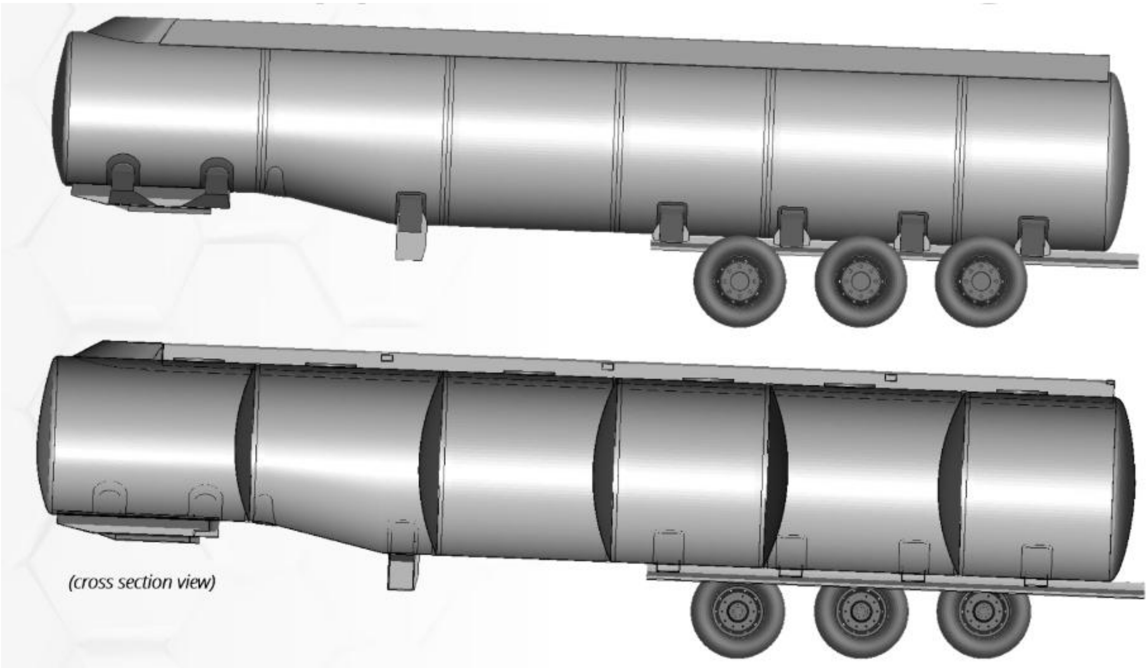


Figure 2.3: FE model of self-supported stuffed design semi-trailer involved in baseline collision

2.4.1.2 Information for model build

Tractor and semi-trailer weights

The tractor and semi-trailer weights used for the modelling are detailed below and were the same for all three models.

- Tractor weight: 8,000 kg
    - Chosen based on internet search which found 6 x 2 tractor weight range was approximately 7.1 t to 8.5 t and advice from an expert who DfT recommended.
- Semi-trailer weight: 36,000 kg
    - It was decided that a worst case loading scenario should be modelled for the following reasons:
        - The tankers involved in the baseline collision were fully loaded.
        - Regulatory type approval is based on a worst-case scenario.

      so given that the maximum gross combination weight (GCW) for vehicle is 44,000 kg in UK, and tractor weight 8,000 kg, this resulted in a semi-trailer weight of 36,000 kg.
    - Fuel loading in tankers is complex because petrol or diesel which have different densities can be carried in different compartments and there are limits on the fill of a compartment allowed, namely maximum fill to allow ullage for expansion (see ADR 4.3.2.2.1 (a)) and degree of fill for roll over stability (see ADR 4.3.2.2.4). Therefore, to enable a comparable loading of the tankers the following approach was taken.

      Tanks were filled to capacity (with required ullage allowed) and density of fuel load adjusted to give GCW of 44,000 kg without exceeding permitted axle loads. OEM supplied load tables were used to achieve this.

      The justification for using this approach included:

        - Tank at maximum capacity will provide the highest centre of gravity, and hence worst case for moment at king pin about the lateral axis.
        - Fuel density is variable in real-world.
        - Approach minimises the effect of different loads when comparing structural performances of semi-trailers.

Geometry and materials

Tankers #1 and #2 were built using information supplied by the OEM, e.g. drawings, component CAD dxf files, etc.. The FE model build process involved constructing a CAD model which was then meshed. To ensure model quality the complete CAD model was provided to the OEM for checking before it was meshed. Material grades were obtained from drawings and properties from TriMech's material database.

It was not possible to obtain the Information to build Tanker #3 from the OEM, so an alternative approach was taken. This involved sourcing a physical example of that tanker model and measuring it. The tanker model required was identified from the tanker plate of the collision involved tanker. Geometric information was obtained by using laser scanning and ultra-sound measurements for material thickness. Material grade information was obtained from inspection reports supplied by the DfT.

2.4.1.3 Fuel model

LS-DYNA 3D crash simulations

For the LS-Dyna3D crash simulations, the fuel was modelled using a smoothed particle hydrodynamic (SPH) method with an appropriate equation of state (EOS) for petroleum.

SPH discretizes the fluid into a set of particles, where each particle carries its own mass, velocity, and other fluid properties, allowing for flexible movement and accurate representation of free surfaces. Its advantages are that it excels at modelling large fluid deformations, such as splashing, impact, and free surface flows, which can be challenging with traditional mesh-based methods. Also, it can handle complex structural geometries with ease, allowing for accurate representation of realistic components.

Linear and non-linear buckling simulations

For the linear and non-linear buckling simulations, the fuel was not modelled because its weight was included in the loading condition. For example, for linear buckling the critical load for the first buckling mode is calculated from the nominal load multiplied by the eigenvalue. For this project, this load was scaled in terms of the weight of the semi-trailer by dividing it by mg to provide a buckling load in terms of g.

2.4.2 Development of baseline boundary conditions

The objective of this work was to develop a lumped parameter model of the baseline collision to estimate and apply boundary conditions for the LS-Dyna3D semi-trailer models, in particular king pin loading Figure 2-17.

The reason that a lumped parameter model was required to apply the king pin loads to the semi-trailers was that initial work found that failure of semi-trailer influences king pin loads; namely it reduces loads with earlier / larger failures giving greater reductions. Therefore, because king pin loads were not independent of semi-trailer behaviour, it was necessary to incorporate this feedback mechanism into the boundary condition application and a lumped parameter model was developed to do this.

Note that the comparison of results for rigid and compliant models later in this section will illustrate the effect of this feedback mechanism.

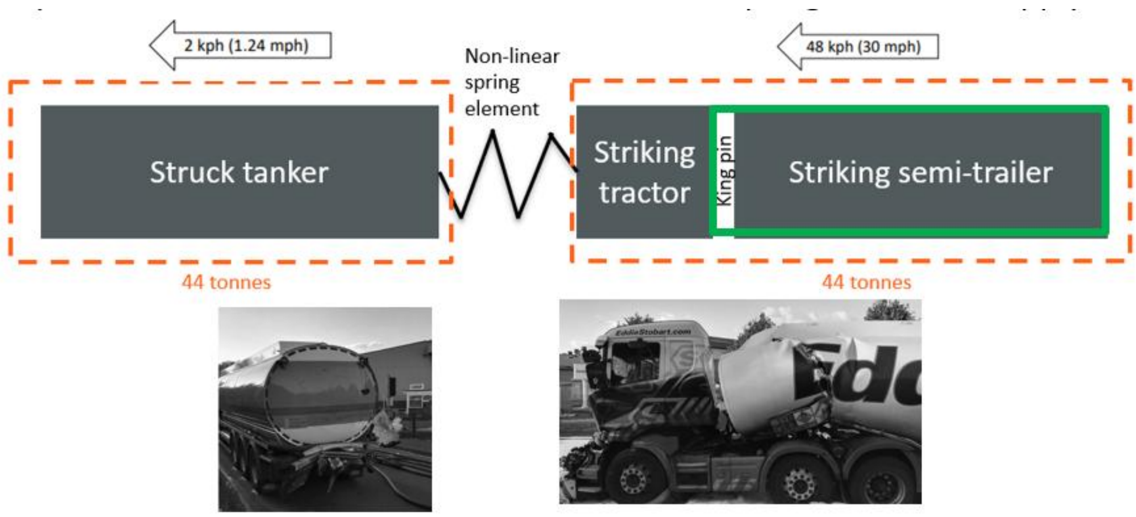


Figure 2-17: Illustration of lumped parameter model of the baseline collision showing the configuration of the model; struck tanker (rigid), non-linear spring element, striking tractor (rigid), king pin and striking semi-trailer (rigid or compliant).

To develop the lumped parameter model, the following steps were performed:

- Step 1: Estimate non-linear spring element force / deflection characteristics.
- Step 2: Develop lumped parameter model.
- Step 3: Test model with rigid and compliant semi-trailer tankers.

2.4.2.1 Step 1: Estimate non-linear spring element force / deflection characteristics

To estimate the non-linear spring force / deflection characteristic, background information was collated and the baseline collision was analysed. The background information collated consisted of regulatory crash pulse load requirements and rigid HGV crash loads measured in Load Cell Wall (LCW) tests (Appendix A.1). Information available about the baseline collision consisted of police reports supplied by the DfT. These contained photographs of the scene post collision and tachograph data from both vehicles involved in the collision.

Analysis of the baseline collision identified that two main load paths were active during the collision, as shown in Figure 2-18. Crush of the structure was estimated from photographs of the collision used to identify the main load paths and vehicle components involved.

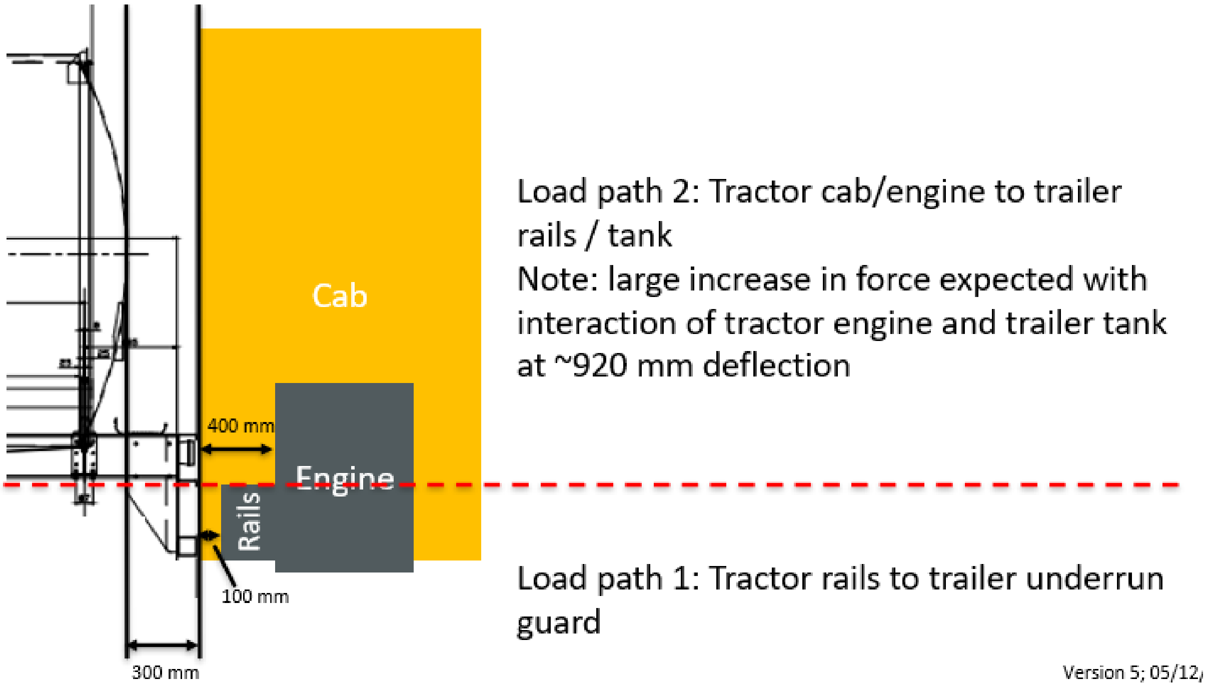


Figure 2-18: Schematic of main components of vehicles and load paths

Some of the photographic evidence used to identify these load paths is detailed below (e.g. Figure 2-19 and Figure 2-20). The photographic evidence was also used to estimate the crush of the structure.

Load path 1: Striking tanker chassis rails to struck tanker underrun guard.

Figure 2-19: Identification of load path 1 between striking tanker chassis rails and struck tanker underrun guard

Load path 2: Striking tanker cab/engine to struck tanker chassis rails

Figure 2-20: Identification of load path 2 between striking tanker cab/engine and struck tanker chassis rails

In the collision, the striking tanker travelling at approx. 30 mph (48 km/h) impacted the rear of nearly stationary tanker, travelling at just over 1 mph (2 km/h). Tachograph data indicated that the speed of the struck tanker rapidly increased to 12 mph (20km/h) after the impact. Assuming that both tankers had a mass of 44 tonnes, and applying the principle of conservation of momentum, a speed of about 15 mph (24 km/h) was predicted for the coalesced vehicles, which agreed reasonably well with tachograph information.

The load path force deflection profiles were developed using knowledge of load path forces, e.g. from LCW data, and an iterative process to tune them with checks on outputs, such as alignment with expected decelerations from regulatory requirements. The following was the result of the tuned profile:

Load path 1: Striking tanker chassis rails to struck tanker underrun guard

The force levels through this load path depend mainly on the strength of the underrun guard because it is the weaker structure.

Assumptions to derive estimate:

- Force level: Regulation No. 58 requires minimum reaction force at bottom of guard of 180 kN, strike is higher up guard, so it was assumed that the force doubled, i.e. circa 360 kN.
- Displacement: Regulation No. 58 para 25.3 requires maximum distance from rear of trailer when minimum required force reached 300 mm. ~100 mm was added to account for crush of weak structure in front of tractor unit rails to give crush of 400 mm.
- Guard does not break off, i.e. keeps providing a reaction force.

Deflection / force estimate: (0 mm, 0 kN), (400 mm, 360 kN), (1200 mm, 400 kN)

Load path 2: Striking tanker cab/engine to struck tanker chassis rails / tank

The force levels through this load path depend mainly on the strength struck tanker trailer chassis rails because it is the weaker structure.

Assumptions / estimate:

- Crush of cab structure:

- Deflection 0 to 100 mm, force 0 to 100 kN.
  - Force level estimated from LCW test data (see Appendix A.1.2).
- Tractor engine engages trailer chassis rail:
  - Deflection 100 to 400 mm, force increases to 1500 kN.
    - Force level estimated from LCW test data (see Appendix A.1.2).
- Trailer chassis rail deforms:
  - Deflection 400 to 920 mm, force increase to 1800 kN.
    - Force level estimated from LCW test data (see Appendix A.1.2).
- Tractor engine engages trailer tank:
  - Deflection 920 mm to 1200 mm, force increase to 3000 kN.
    - Deflection estimated from crush measurements taken from collision.
    - Force level estimated from LCW test data (see Appendix A.1.2).

Deflection / force estimate: (0 mm, 0 kN), (100 mm, 100 kN), (400 mm, 1500 kN), (920 mm, 1800 kN), (1200 mm, 3000 kN).

The force deflection curve for the sum of both load paths used for the non-linear spring element is shown in Figure 2-21 below.

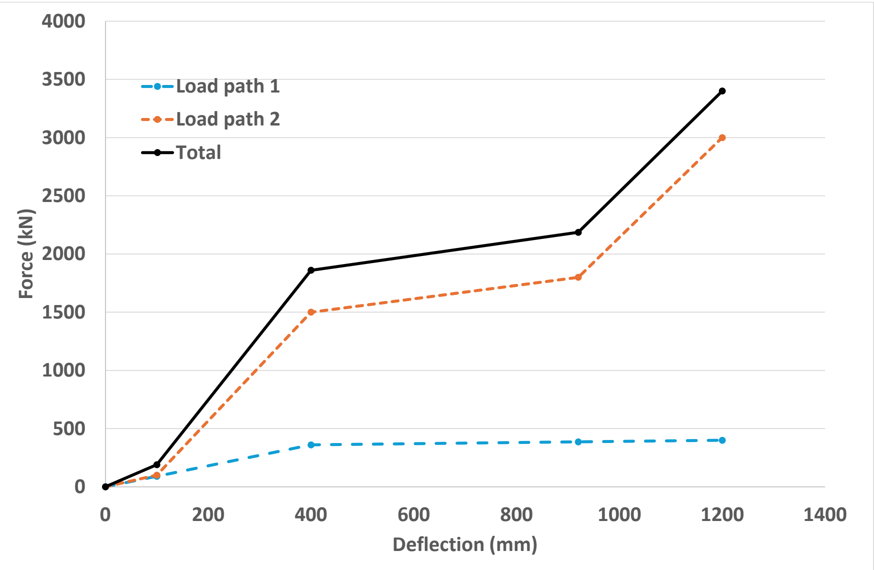


Figure 2-21: Total non-linear spring force deflection profile

It should be noted that because the deformation in the collision was mainly plastic, an unloading curve having a stiffness 10 – 50 times greater than that of the loading curve was incorporated into the non-linear spring. This ensured that elastic recovery of the lumped parameter model was representative of the collision, i.e. minimal.

2.4.2.2 *Step 2: Develop lumped parameter model*

Two lumped parameter models were developed: one with a rigid striking tanker and the other with a compliant semi-trailer tanker. Both used the non-linear spring with unloading curve described in step 1 above. The boundary conditions for each of these models were slightly different and are described below.

Rigid model – boundary conditions

The rigid model consisted of two masses connected by the non-linear spring element. The lumped masses were constrained to move in x only (fore /aft) and initial velocity conditions were imposed on them as shown (Figure 2-22).

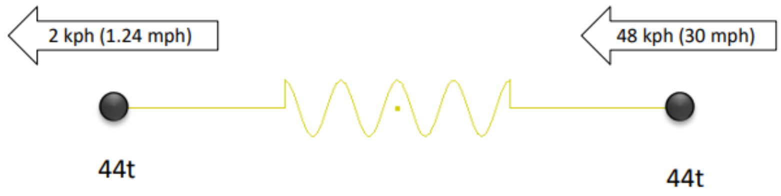


Figure 2-22: Rigid lumped parameter model

Compliant model - boundary conditions

The compliant model consisted of:

- Struck tanker – lumped mass constrained to move in x only.
- Striking tractor – lumped mass constrained to move in x only.
- Non-linear spring element – connecting struck tanker and striking tractor.
- King pin joint between striking tractor and striking compliant semi-trailer.

Initial velocities were applied as shown in Figure 2-23.

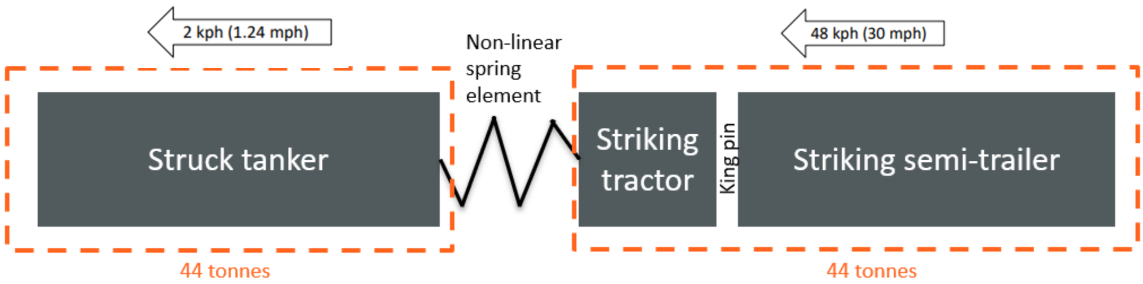


Figure 2-23: Compliant lumped parameter model

The striking compliant semi-trailer was constrained as below:

- Axles:
  - Constrain Y (vertical), Z (lateral) translation and X (longitudinal) rotation.
- King pin:
  - Planar and spherical joints which permit translation in YZ plane and full rotation, i.e. constrain to striking tractor in X only.

- Gravity – none applied.

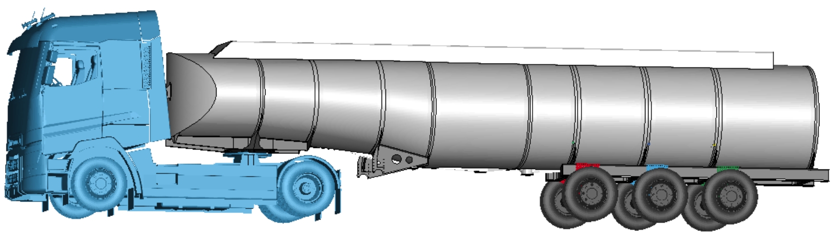


Figure 2-24: Striking compliant semi-trailer model

The king pin constraint was chosen because it represents a worst-case scenario with minimal support from the fifth wheel. It does not constrain the motion of the tanker at the king pin in the vertical direction at all, which a king pin in the real world would. However, at the start of the impact, this constraint will be representative of the real world because the tractor unit will compress down on its suspension. Later on in the impact it will not be representative because resistance to vertical movement will be provided. Consideration to modelling this resistance was given, but it was decided not to model it because of the variability that different suspension designs could introduce, and that the project was mainly interested in the initial stages of the impact when buckling of the semi-trailer started to occur and vertical constraint would be minimal. It should be noted that because of the lack of constraint in the x direction at the king pin, gravity was not applied to the model, to ensure that the weight of the front of the trailer was not added to the moment caused by the impact of the mass of the trailer acting around the king pin.

2.4.2.3 Step 3: Test model with rigid and compliant semi-trailer tankers

A snapshot from animations of the rigid and compliant models in the baseline collision towards the end of the collision is shown (Figure 2-25). The tankers are represented by pictures of standard articulated HGVs because suitable pictures of tanker HGVs were not available. It is seen that the tractor of the striking tanker has penetrated the trailer of the struck tanker. This penetration illustrates the crush of the struck trailer and striking tractor, which is represented in the model by the non-linear spring. It is also seen that the compliant model buckles and fails, resulting in a rotation of the front of the tanker which penetrates the tractor unit. This occurs at a time of about 130 msec and represents a time after which the results are less representative because this penetration would not occur in the real-world.

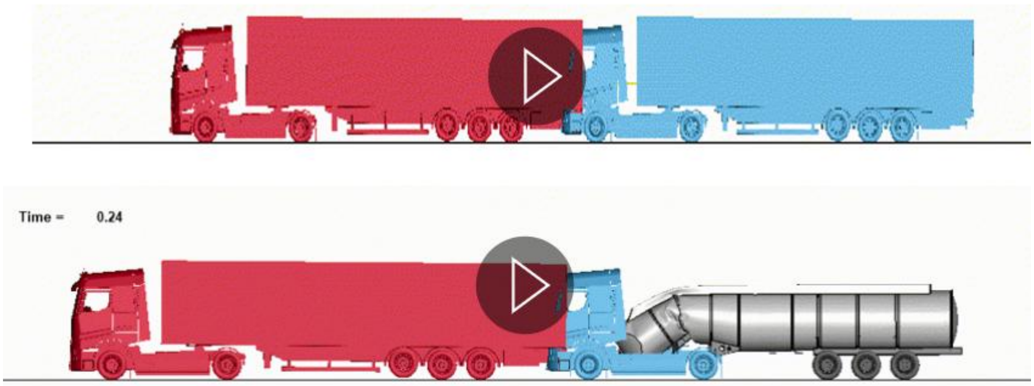


Figure 2-25: Snapshot from animations of rigid and compliant tanker models towards end of collision

Figure 2-26 shows a comparison of velocity histories for the rigid and compliant tanker lumped parameter models. For the rigid model, it is seen that the velocity of the struck tanker (1) decreases, and the velocity of the striking tanker (2) increases until they coalesce at about 140 msec, after which there is some elastic recovery and the tankers continue at slightly different velocities. Note that if there were no elastic recovery, they would continue at the velocity at which they coalesce. For the compliant model, it is seen that the velocity histories are quite different, the main reasons being:

- the effect of the fuel movement which is included in the model of the compliant tanker, and
- the buckling and failure of the compliant tanker semi-trailer.

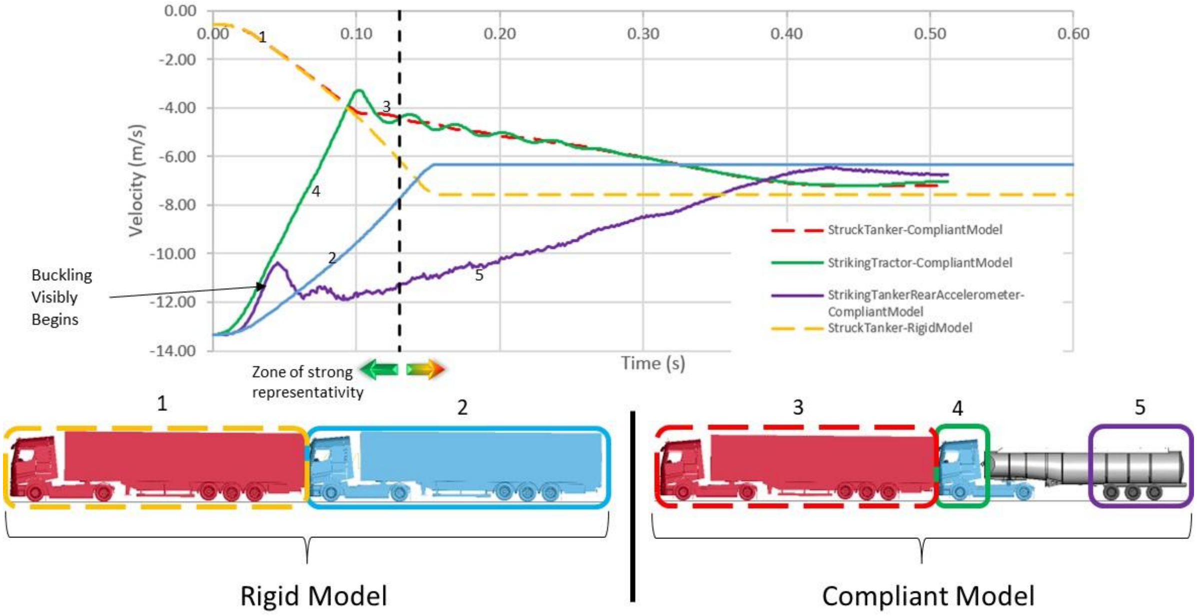


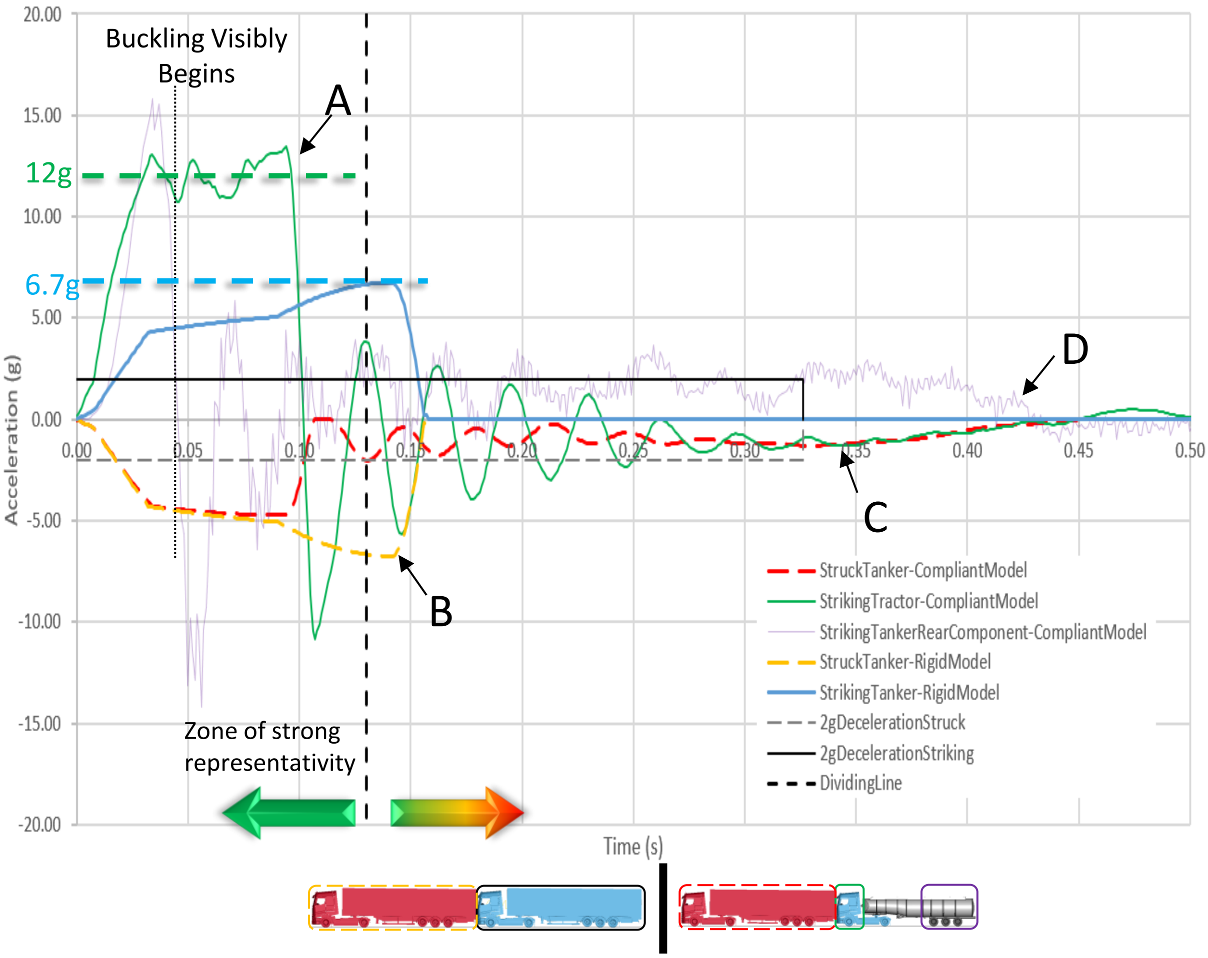
Figure 2-26: Comparison of velocity histories for rigid and compliant tanker lumped parameter models

The effect of modelling the fuel movement is to partially decouple its mass from the semi-trailer at the beginning of the impact, which results in a higher deceleration of the uncoupled mass, namely the tractor unit and semi-trailer tank, because of newtons law, namely f = ma; the force is constant, the mass less and so the acceleration is greater. This is seen as a divergence of the curves for the tractor velocity (4) and the semi-trailer rear velocity (5) from that of the striking rigid tanker velocity (2) at the beginning of the impact.

The effect of the buckling and failure of the compliant tanker semi-trailer is seen after about 40 ms into the impact. The failure reduces the deceleration load on the rear of the semi-trailer and further decouples it. It is seen as a divergence between the velocities of the rear of the semi-trailer and the tractor unit.

It should be noted that towards the end of the impact, all velocities tend to one constant velocity which is predicted by the conservation of momentum, which is turn demonstrates that the physics of the models is correct.

Further explanation of the effect of the buckling and failure of the compliant lumped parameter, compared to the rigid, model is given below in Figure 2-27 in terms of acceleration histories.



- In comparison to the rigid body model higher decelerations of the **striking tractor** are seen in the compliant model. (A)
- The discrepancy between acceleration of the struck tanker in the **rigid** and **compliant** models is due to the buckling of the fuel tanker. (B)
- **Struck tanker** and **striking tractor** continue to accelerate in the compliant model as the impulse from the **striking tanker** continues to act on the model. (C)
- The **striking tanker** continues to decelerate in the compliant model as it equalises velocity with the **struck tanker** and **striking tractor**. (D)

Figure 2-27: Comparison of acceleration history traces for rigid and compliant lumped parameter models showing the effect of buckling and failure of the compliant semi-trailer model. Note 2 g deceleration shown for reference because related to EN 13094 structural requirement

The effect of the buckle and failure of the compliant lumped parameter model compared to the rigid model also has an effect on the non-linear spring element. Because this failure absorbs energy, the non-linear spring element absorbs less energy for the compliant model and hence peak forces are less (Figure 2-28).

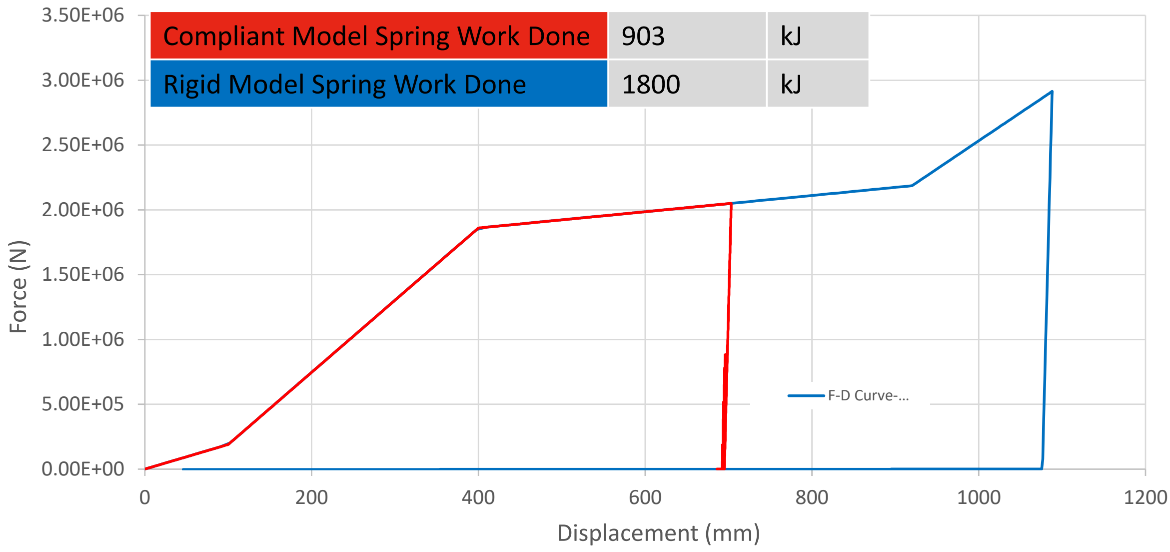


Figure 2-28: Comparison of non-linear spring force displacement histories for rigid and compliant models showing less energy absorbed and lower peak force for compliant model

2.4.2.4 Summary

- A lumped parameter model incorporating a non-linear spring element was developed to simulate the interaction between tankers during a collision. This model was used to apply kingpin loads to semi-trailer models.
- Comparison of the results from rigid and compliant semi-trailer models shows how buckling failure of the semi-trailer and the modelling of fuel movement changes the collision loads.
    - Peak load was reduced by about 30% for the compliant semi-trailer model which fails, compared to rigid semi-trailer model which does not fail.
    - Interpretation of results provided on the basis of velocity time history plots.

2.4.3 LS-Dyna3D crash simulations

2.4.3.1 Simulations performed

A baseline simulation and three sets of parameter sweeps were performed as follows (Table 2-5):

Baseline:

The baseline collision is described in Section 2.1.1. In this collision, the front of one fuel tanker struck the rear of another and there was a substantial release of flammable liquid from the striking tanker. The striking tanker was travelling at 30 mph (48 km/h) and the struck tanker was nearly stationary at 1 mph (2 km/h). Development of the boundary conditions to simulate this collision are described in Section 2.4.1 above.

Parameter sweep 1 - lower initial velocities:

This parameter sweep consisted of simulations with lower initial velocities for the striking tanker (12 km/h – a quarter, 18 km/h – three eights, 24 km/h – a half). They were chosen to encompass a range where tankers buckled and did not buckle. They aimed to help understand better the collision velocities at which a current tanker could be expected to maintain its integrity in a frontal impact with the rear of an HGV.

Parameter sweep 2 - fixed king pin decelerations:

This parameter sweep consisted of simulations with velocity boundary conditions on the king pin to give decelerations of a 2 g, 1.6 g, 1.4 g and 1.0 g. The velocity boundary conditions were applied for a time calculated to give the striking tanker a delta v of 23 km/h, the same as it experienced in the baseline collision. These decelerations were chosen to encompass a range where tankers buckled and did not buckle. The original basis for them was the EN 13094 regulatory requirement for a 2 g load in longitudinal direction at the king pin without design stresses being exceeded (see Section 2.1.3).

Parameter sweep 3- struck tanker lower masses:

This parameter sweep consisted of simulations with lower masses of the struck tanker to approximately represent collisions into a light vehicle (mass of struck tanker changed to 2,000 kg) and a battery electric van (mass of struck tanker changed to 4,250 kg[15]). They aimed to help understand better the mass of a struck vehicle at which a current tanker could be expected to maintain its integrity in a frontal impact with.

It should be noted that, although force deflection characteristic of the non-linear spring developed for the baseline collision would not be expected to be precisely representative of the interaction at the lower velocities, and struck tanker masses simulated in the parameter sweeps, it was still used. This was because the spring stiffness will be overestimated for these simulations, thus peak forces calculated will be overestimated, and therefore the likelihood of buckling also overestimated. This means if buckling is not predicted, then this prediction will be correct. This was important to check alignment of the modelling results with the collision analysis results. For example, the collision analysis found that the frequency of collisions of tankers with the rear of light vehicles was medium to high, but no tank failures were found. This indicates that the risk of a tank failure in these types of collisions must be negligible. Therefore, it would be expected that the FE modelling should predict 'no buckling' for these simulations in order for it to be consistent with the collision analysis results.

[15] A standard UK category B driving license (car license) allows the holder to drive vehicles, such as vans and pickups, up to a maximum authorised mass (MAM) limit of 3.5 tonnes. Because electric vans are typically heavier than their diesel counterparts, due to the weight of the batteries, a derogation has been issued to allow holders of standard UK car licenses to drive electric vans up to a MAM limit of 4.25 tonnes. See: https://www.fleetnews.co.uk/news/driver-training-requirement-dropped-for-425-tonne-electric-van-licensing

Table 2-5: Matrix of crash simulations performed

Simulation name	Description (Initial tanker velocities)	Tanker #1: self-supported banded	Tanker #2: supported stuffed	Tanker #3: self-supported stuffed (representative of collision involved tanker)
1: Baseline	Striking tanker: 48 km/h delta v 23 km/h; Struck tanker: 2 km/h	✓	✓	✓
Parameter sweep 1: Striking tanker velocity lower initial velocities				
2: Velocity 12 km/h	Striking tanker: 12 km/h Struck tanker: 0 km/h	✓	✓	✓
3: Velocity 18 km/h	Striking tanker: 18 km/h Struck tanker: 0 km/h	✓	✓	✓
4: Velocity 24 km/h	Striking tanker: 24 km/h Struck tanker: 0 km/h	✓	✓	✓
Parameter sweep 2: Striking tanker fixed king pin decelerations				
5: Fixed deceleration 1.4g	Striking tanker: 1.4g, delta v 23 km/h Struck tanker: N/A	✓	✓	✓
6: Fixed deceleration 1.6g	Striking tanker: 1.6g, delta v 23 km/h Struck tanker: N/A	✓	✓	✓
7: Fixed deceleration 2.0g	Striking tanker: 2.0g, delta v 23 km/h Struck tanker: N/A	✓	✓	✓
8. Fixed deceleration 1.0g	Striking tanker: 1.0g, delta v 23 km/h Struck tanker: N/A	x	x	✓
Parameter sweep 3: Struck tanker lower masses				
9: Struck tanker mass 2,000 kg	Striking tanker: 48 km/h Struck tanker: 2 km/h	x	x	✓
10: Struck tanker mass 4,250 kg	Striking tanker: 48 km/h Struck tanker: 2 km/h	x	x	✓

2.4.3.2 Results

Baseline

A snapshot from the animations of the baseline collision simulations towards the end of the collision shows that all three designs fail in a similar manner, buckling just behind the king pin, followed by a rotation of the front of the semi-trailer which penetrates the tractor unit (Figure 2-29). As mentioned previously in Section 2.4.1, which describes the development of the lumped parameter model, the penetration of the tractor unit occurs at a time of about 130 msec and represents a time after which the results are less representative because this penetration would not occur in the real-world.

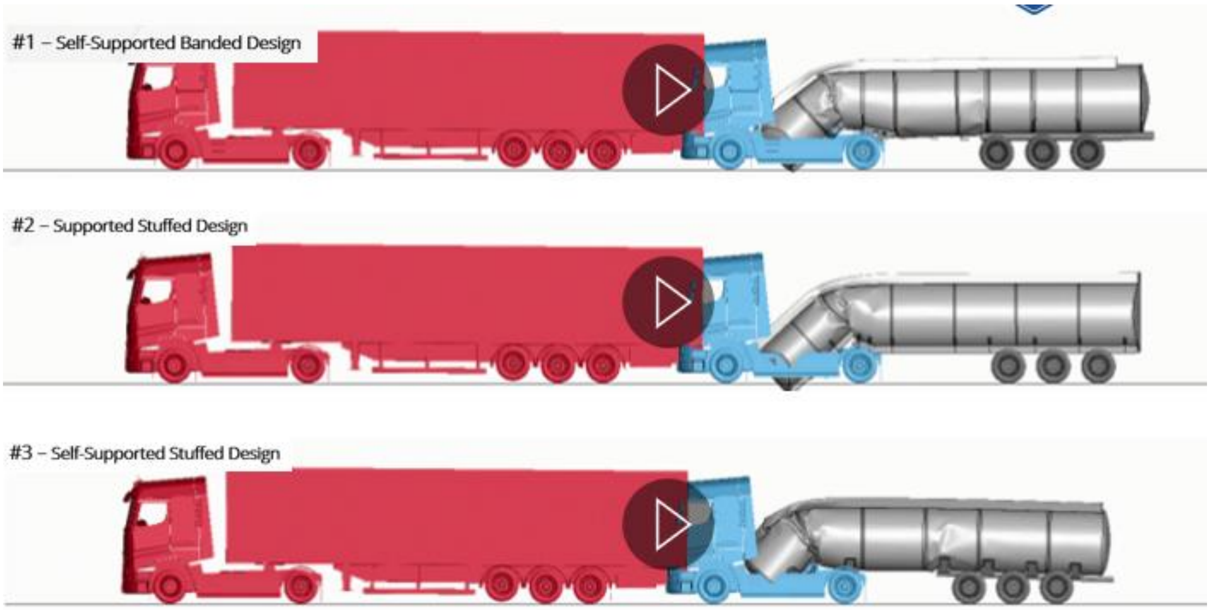


Figure 2-29: Snapshot from the animations of the baseline simulations towards end of the collision

Figure 2-30 shows a comparison of the longitudinal forces at the king pin for each semi-trailer design. It is seen that the semi-trailer designs which buckle earlier, sustain less load at the king pin than those that buckle later with times and peak loads as follows:

#3 Self-supported stuffed design	0.034 s	900 kN ~2.5 g
#1 Self-supported banded design	0.037 s	1150 kN ~3.3 g
#2 Supported stuffed design	0.047 s	1425 kN ~4.0 g

This indicates that the self-supported stuffed design, which is representative of the tanker in the baseline collision, is the most susceptible to buckling and failure in this type of collision, whereas the supported stuffed design is the least susceptible.

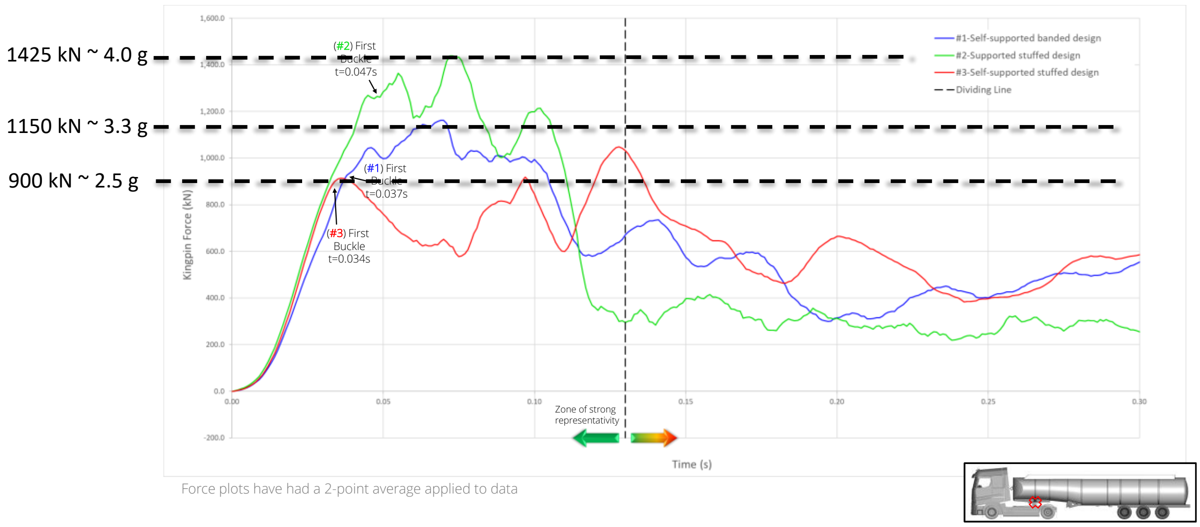


Figure 2-30: Comparison of semi-trailer king pin force in longitudinal (x) direction for baseline simulations

Parameter sweep 1 – striking tanker velocity lower initial velocities

Figure 2-31 shows a comparison of the maximum deformations for the three semi-trailer designs for the lower initial velocity parameter sweeps. It is seen that there is no significant buckling for any semi-trailer design at an initial velocity of 12 km/h (quarter of the baseline velocity) but all designs buckle at 24 km/h (half the baseline velocity).

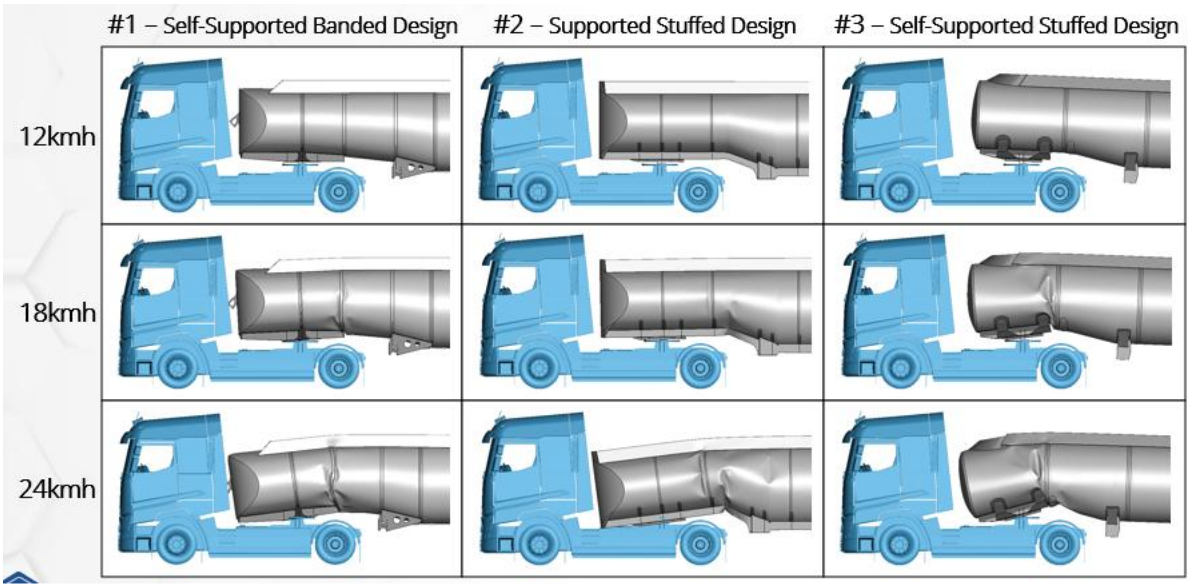
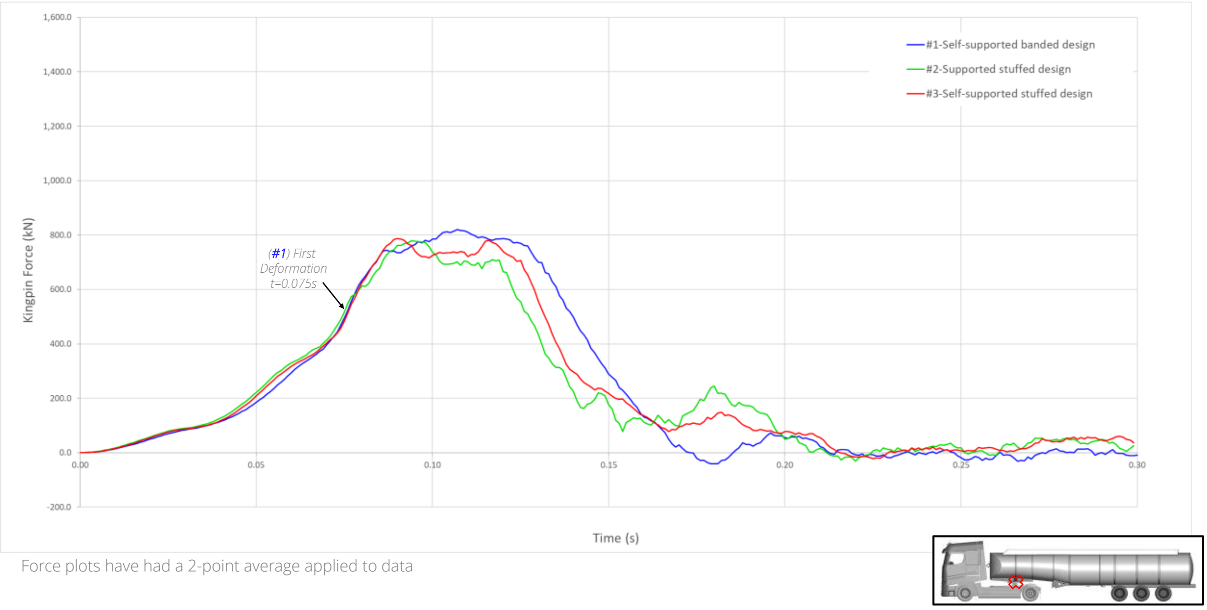


Figure 2-31: Comparison of the maximum deformations for the three semi-trailer designs for the lower initial velocity parameter sweeps

Figure 2-32 shows a comparison of the king pin longitudinal forces for the three semi-trailer designs for the 12 km/h lower initial velocity parameter sweep. It is seen that the peak forces are similar, at about 800 kN (equivalent to ~2.3 g), because no semi-trailer buckles. A semi-trailer which buckles reduces peak loads.



Force plots have had a 2-point average applied to data

Figure 2-32: Comparison of semi-trailer king pin force in longitudinal (x) direction for 12 km/h lower initial velocity parameter sweep

Similar plots showing the comparison at higher lower initial velocities (18 km/h and 24 km/h) in Appendix B.1 show a difference in the peak force values because some tankers buckle in these simulations. A trend similar to that noticed for the baseline collision in terms of susceptibility to buckling and capability to withstand load at the king pin is seen, namely that the self-supported stuffed design is the worst, the self-supported banded design better, and the supported stuffed design the best.

Parameter sweep 2 - fixed king pin deceleration

Figure 2-33 shows a comparison of the maximum deformations for the three semi-trailer designs for the fixed king pin deceleration parameter sweeps. It is seen that there only semi-trailer #3 self-supported stuffed design buckles significantly at 1.4 g, but at 2.0 g all semi-trailer designs buckle significantly.

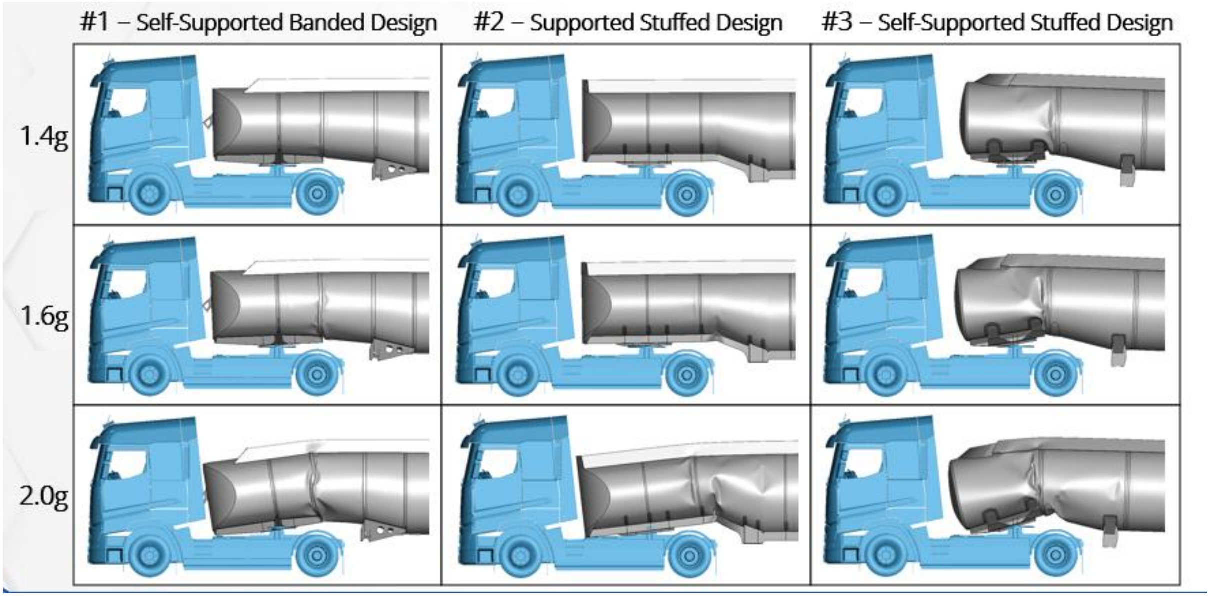


Figure 2-33: Comparison of the maximum deformations for the three semi-trailer designs for the fixed king pin deceleration parameter sweeps

Figure 2-34 shows a comparison of the semi-trailer rear (axle) velocities for 1.4 g fixed king pin deceleration parameter sweep. It is seen that the velocity of semi-trailer #3 'Self-supported stuffed design' deviates significantly from the deceleration boundary condition constraint applied to the king pin, whereas the other semi-trailers do not. This is because semi-trailer #3 buckles significantly, which causes a reduction of the forces on the trailer rear of the buckle position with a corresponding reduction in its deceleration (F=ma) which is seen in the deviation of the velocity.

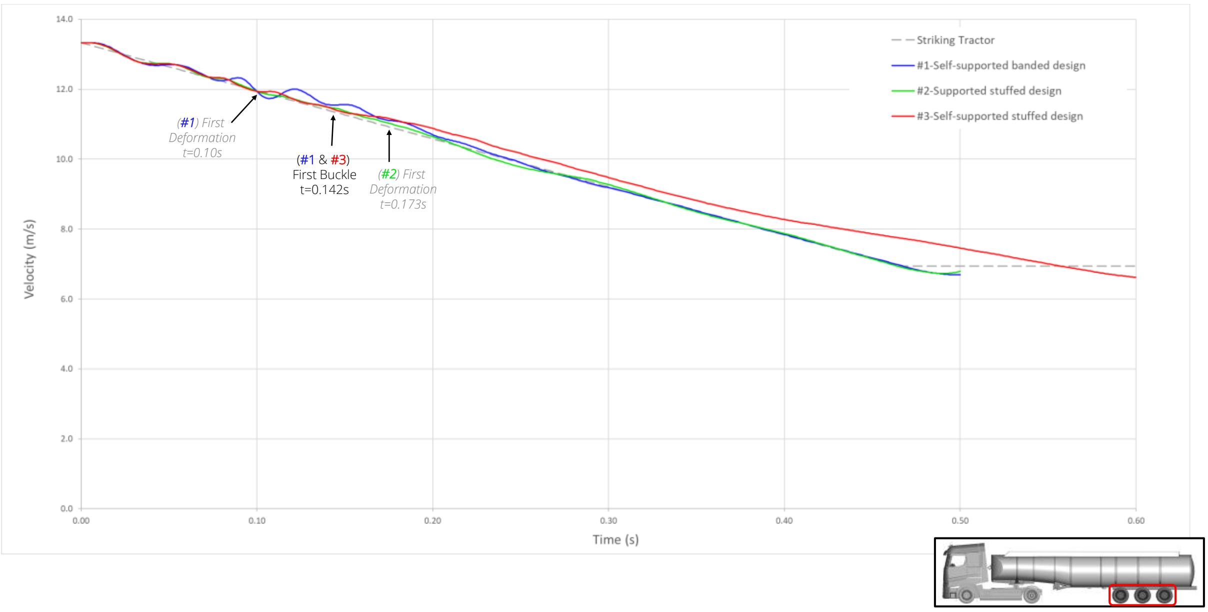


Figure 2-34: Comparison of semi-trailer rear (axle) velocities for 1.4 g fixed king pin deceleration parameter sweep. Note that dashed grey line represents the deceleration boundary condition constraint applied to the king pin

Even though semi-trailer designs #1 and #2 designs did not buckle significantly, on close examination some small deformations of the tank were seen in the animations in the region where buckling occurred for simulations with higher loads. In order that these deformations were not confused with the onset of buckling 'first deformation' and 'first buckle' were defined as follows:

- **First deformation**: minor deformation which **does not** influence the motion of the rear of the tanker, i.e. **small deformation without any collapse**
- **First buckle**: major deformation which **does** influence the motion of the rear of the tanker, i.e. **large deformation leading to collapse**.

Figure 2-35 shows a comparison of the king pin longitudinal forces for the 1.4 g fixed king pin deceleration boundary condition parameter sweep. The mass of the semi-trailer including the fuel load is 36,000 kg so using F = ma, a deceleration of 1.4 g applied to it could be expected to result in a force of 1.4 x 9.81m/s^2 x 36,000 kg = 494 kN. However, the peak force recorded is much higher at 900 kN. The reason for this is because of the fuel movement. When the semi-trailer body is decelerated initially, the fuel load does not decelerate with the body of the semi-trailer and keeps moving forward, which reduces the king pin loads until eventually it is arrested by the partitions, at which point its deceleration increases significantly which increases the king pin loads above those calculated for the nominal deceleration level. Effectively, the fuel load is decoupled from the semi-trailer body. The degree of decoupling and the associated difference between the nominal deceleration load and peak load experienced can be controlled by the number of partitions and baffles that are included in the semi-trailer design.

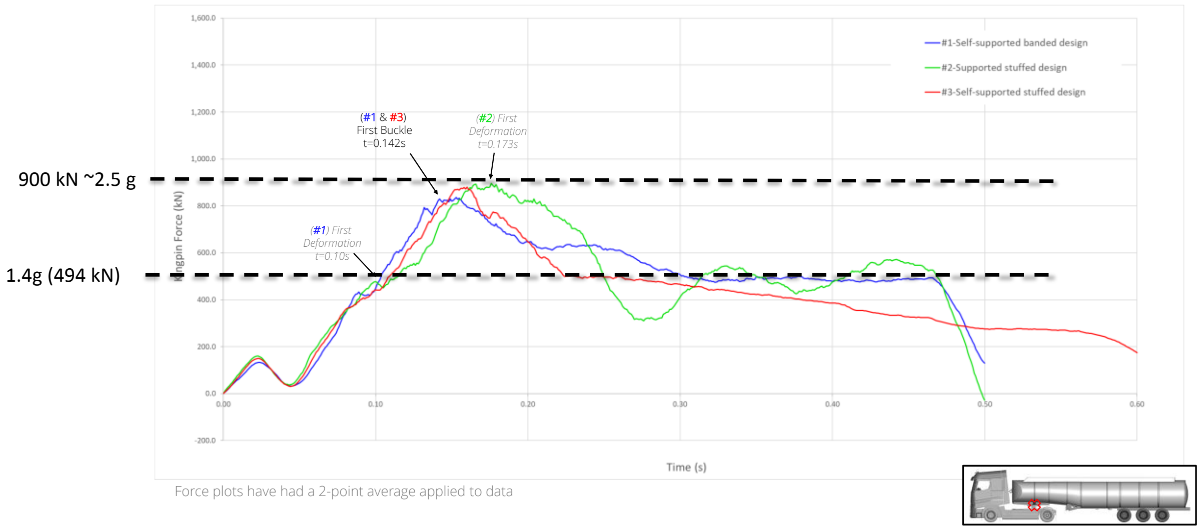


Figure 2-35: Comparison of king pin longitudinal forces for 1.4 g fixed king pin deceleration boundary condition parameter sweep

Plots of the velocities and king pin longitudinal forces for the 1.0 g 1.6 g, and 2.0 g parameter sweeps can be found in Appendix B.2. It is interesting to note that for the 1.0 g parameter sweep, peak loads of about 2.2 g were measured because of the effect of fuel motion. This is in alignment with the 2.0 g regulatory requirement for king pin loading in EN 13094 (see Section 2.1.3) which is related to in-service braking loads. These would give maximum nominal decelerations of circa 0.6 to 1.0 g, which are then increased to compensate for the higher peak loading caused by the arrest of fuel movement.

Parameter sweep 3 – struck tanker lower masses

A snapshot from the animations of the parameter sweep with the struck tanker lower masses towards the end of the collision shows that the semi-trailer #3 self-supported stuffed design does not buckle significantly in either of the two collisions modelled (Figure 2-36). Peak king pin loads of 700 kN (~ 2.0 g) and 900 kN (~ 2.5 g) were measured for the 2 t and 4.25 t struck vehicles, respectively.

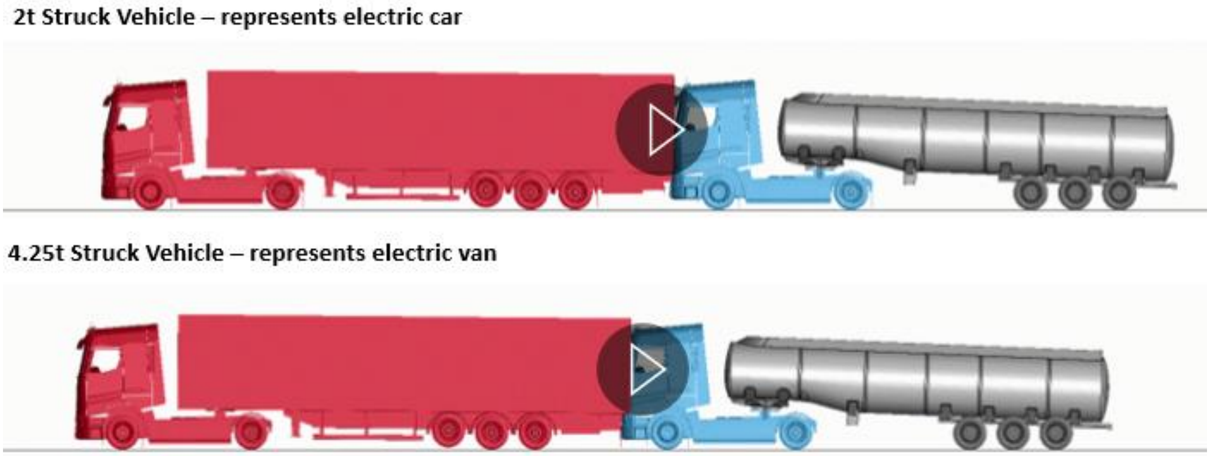


Figure 2-36: Snapshot from the animations of parameter sweep struck tanker lower masses towards the end of the collision for semi-trailer #3 self-supported stuffed design

Close examination of the collision with the 4.25 t electric van shows a small deformation which indicates that, if the loading were a little higher, buckling may have occurred (Figure 2-37).

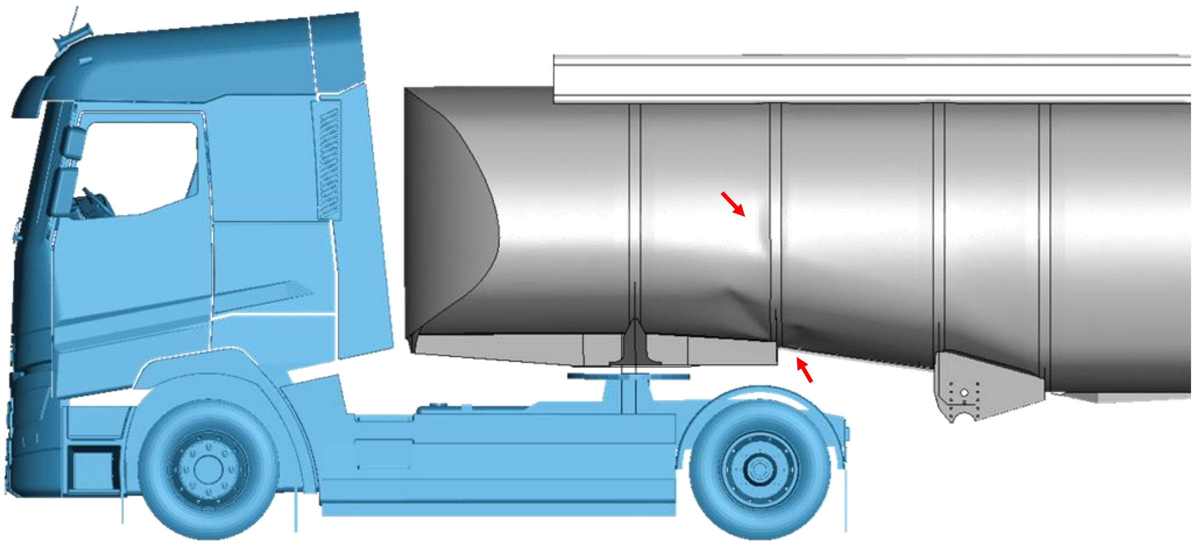


Figure 2-37: Close up of semi-trailer #3 self-supported stuffed design at end of collision with 4.25 t electric van showing some minor deformation of the tank

However, it should be noted that the stiffness of the non-linear spring was not changed from its representation of an HGV to HGV collision for these simulations. Therefore, it will be too stiff to correctly represent a collision with an electric car or van and result in peak loads on the semi-trailer above those that would be experienced in the real-world. Thus, if buckling does not occur in the simulation, it can be inferred with some confidence that buckling will not occur in the real-world because loads are over-predicted. However, if buckling or near buckling is predicted in the simulation, there is a reasonable likelihood that it would not occur in the real-world.

2.4.4 Linear buckling simulations

2.4.4.1 Model overview and simulations performed

The boundary conditions for the buckling model were made to imitate the constraints in the LS-Dyna3D explicit crash model. The axles were constrained in all translational and rotational degrees of freedom and a nominal load of 353 kN (=36,000 kg x 9.81 m/s^2) was applied to the kingpin in the same location as in the LS-Dyna3D explicit crash model (Figure 2-38). Because the linear analysis solver outputs a buckling factor based on the nominal load applied, by applying a nominal load of 1 g, this factor was effectively in units of g.

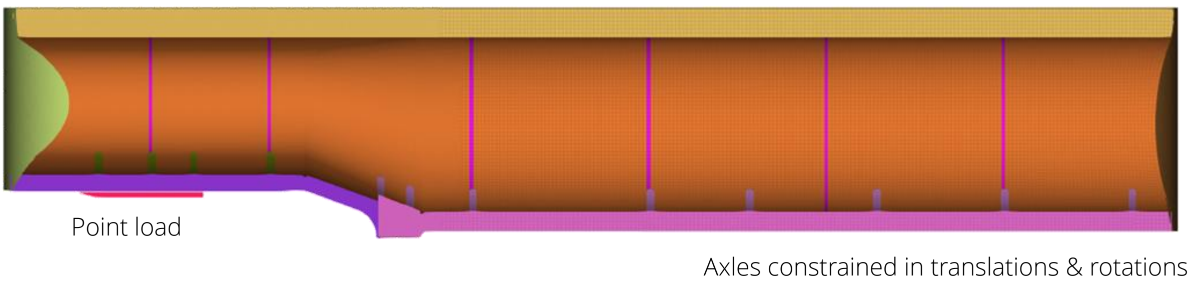


Figure 2-38: Illustration of boundary conditions for linear buckling models

Simulations were performed for the three semi-trailer designs:

- Tanker #1 – self-supported banded design (typical example)
- Tanker #2 – supported stuffed design (typical example)
- Tanker #3 – self-supported stuffed design (representative of one involved in baseline collision)

2.4.4.2 Results and discussion

The buckling modes predicted by the linear analysis are compared with those from the crash analysis for tanker #1, #2, and #3 in Figure 2-39, Figure 2-40 and Figure 2-41 below, respectively. Buckling factors and buckling loads are shown in Table 2-6.

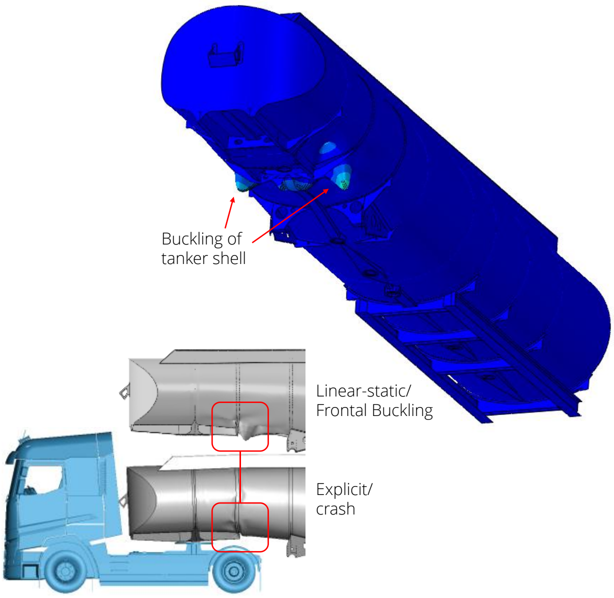


Figure 2-39: Comparison of buckling mode predicted by linear analysis with that predicted by LS-DYNA 3D crash analysis for tanker #1 self-supported banded design

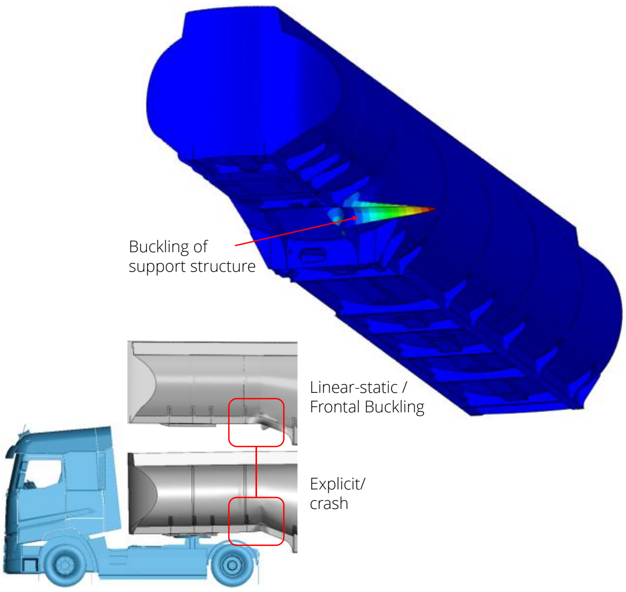


Figure 2-40: Comparison of buckling mode predicted by linear analysis with that predicted by LS-DYNA 3D crash analysis for tanker #1 supported stuffed design

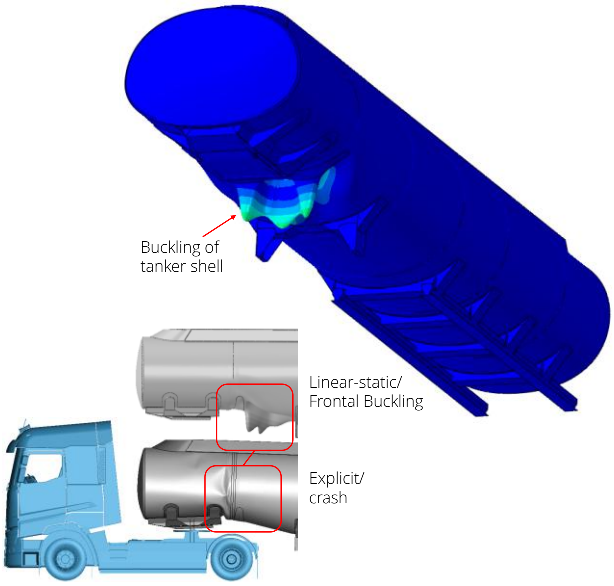


Figure 2-41: Comparison of buckling mode predicted by linear analysis with that predicted by LS-DYNA 3D crash analysis for tanker #3 self-supported stuffed design

Table 2-6: Buckling factors and forces predicted by linear buckling analysis.

	Buckling Factor (g)	Buckling Force (kN)
#1 Self-Supported Banded Design	2.82	997
#2 Supported Stuffed Design	2.64*	934*
#3 Self-Supported Stuffed Design	2.57	907

*Note: the 2.64g buckling factor recorded for tanker #2 occurred on the supporting structure and not the tanker shell itself. This means that if the supporting structure were better designed buckling force of the tanker shell would be substantially higher.

For tanker #1 self-supported banded design, it is seen that the crash and linear buckling analyses predicted similar initial points of buckling of the shell near the front of compartment 2. A buckling factor of 2.82 g was predicted.

For tanker #2 supported stuffed design, it is seen that the crash and linear buckling analyses predicted similar initial points of buckling of the support structure with a buckling factor of 2.64 g. The crash analysis shows that loads much higher than the load at which the supporting structure buckles are sustained before the tanker shell buckles. This was expected, because in principle, a supported type design should sustain a much higher load than an unsupported one. Indeed, if the supporting structure were designed so that it did not buckle at such low loads, it is expected that tanker #2 would have a much higher buckling factor.

For tanker #3 self-supported stuffed design, it is seen that the crash and linear buckling analyses predict differences in the initial point of buckling, with buckling predicted to occur ahead of the banded partition and reinforcing section in the crash model, whilst it is predicted behind this region in the linear buckling model. The likely reason for this difference is that the effect of fuel motion was included in the crash model, whereas it was not included in the linear buckling model. Compared to the linear buckling model, in the crash model arrest of the fuel motion will result in higher loads on the partition and thus increase the likelihood of buckling occurring ahead of the partition.

2.4.5 Non-linear buckling simulations

2.4.5.1 Model overview and simulations performed

As for the linear buckling simulations the boundary conditions for the non-linear models were made to imitate the constraints in the dynamic crash model, i.e. the axles were constrained in all translational and rotational degrees of freedom, and a nominal load was applied to the kingpin in the same location as in the LS-Dyna3D explicit crash model.

A nominal load of 2.93 g was applied to the kingpin. This load value was selected because it was greater than the buckling loads predicted by the linear analysis and allowed an additional 10% overload to capture the post buckling behaviour of the supported stuffed tanker design (i.e. #2) for which a 2.64 g buckling load was predicted by the linear analysis.

To determine the onset of buckling in the models and examine behaviour post buckle, the Abaqus non-linear 'Riks' solver was used. This is an equilibrium, path-based solution method i.e. the solution is viewed as the discovery of a single equilibrium path in a space defined by the nodal variables and the loading parameter. This algorithm yields solutions regardless of if the structural response is stable or unstable. Therefore, the transition from stability to instability was used to determine the onset of buckling in the tankers based on the force history as illustrated in Figure 2-42.

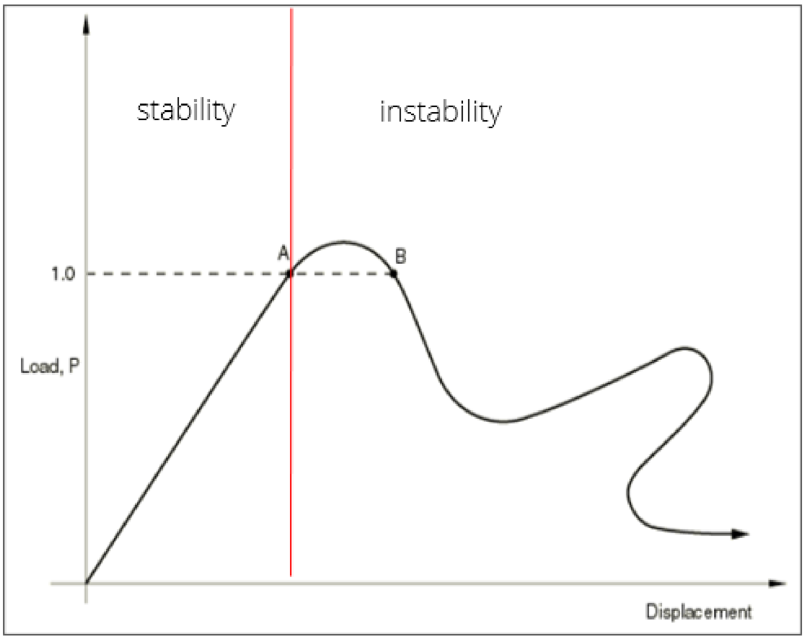


Figure 2-42: Example of transition from stability to instability using the Abaqus Riks solver which was used to define the onset of buckling

It should be noted that non-linear analyses typically predict lower buckling loads than linear analyses because:

- Linear buckling analyses only consider linear material behaviour and small deformation theory, whereas non-linear analyses are capable of modelling non-linear material and geometric behaviour, in addition to large deformations (e.g. as a result of yielding).
- Structural 'imperfections' cannot be effectively represented using linear analysis techniques and have the potential to result in large decreases in model stiffness and second order bending—leading to the overestimation of critical buckling loads.

Simulations were performed for the three semi-trailer designs:

- Tanker #1 – self-supported banded design (typical example)
- Tanker #2 – supported stuffed design (typical example)
- Tanker #3 – self-supported stuffed design (representative of one involved in baseline collision)

2.4.5.2 Results and discussion

The buckling modes predicted by the non-linear analysis are compared with those from the LS-Dyna3D explicit crash and linear analyses for tanker #1, #2, and #3 in Figure 2-43, Figure 2-44 and Figure 2-45 below, respectively. Buckling loads are compared in Table 2-7.

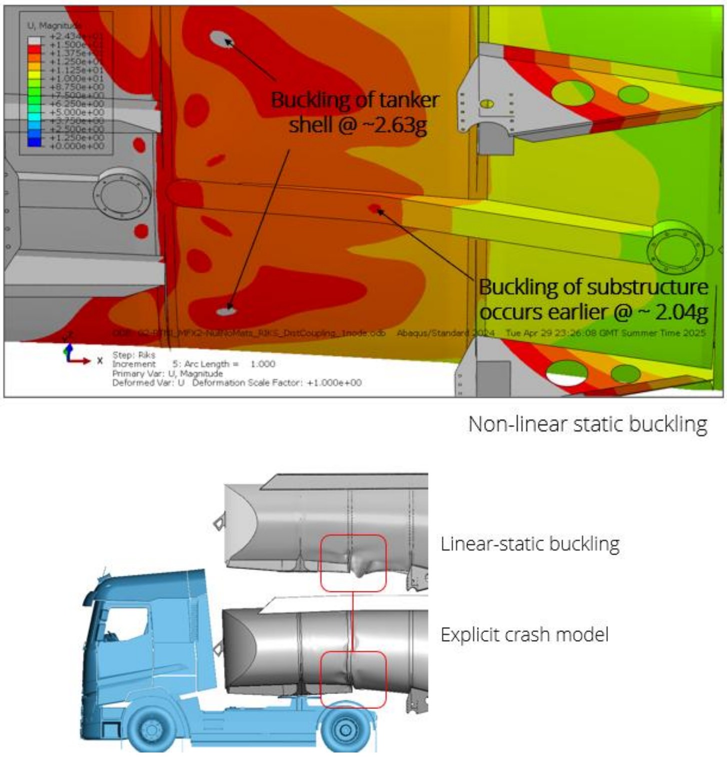


Figure 2-43: Comparison of buckling mode predicted by non-linear analysis compared with that predicted by LS Dyna3D crash (explicit) and linear analyses for tanker #1 self-supported banded design

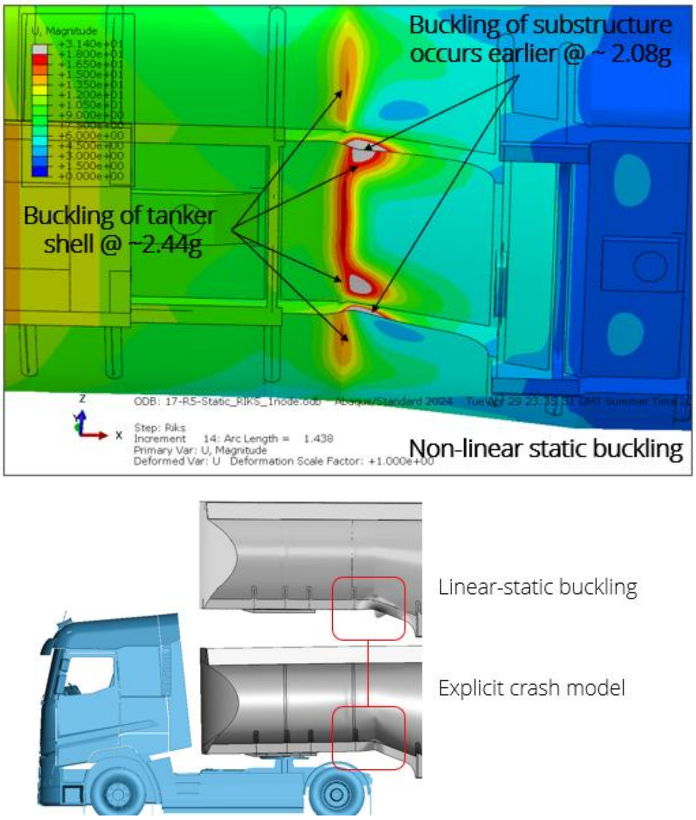


Figure 2-44: Comparison of buckling mode predicted by non-linear analysis compared with that predicted by LS Dyna3D crash (explicit) and linear analyses for tanker #2 supported stuffed design

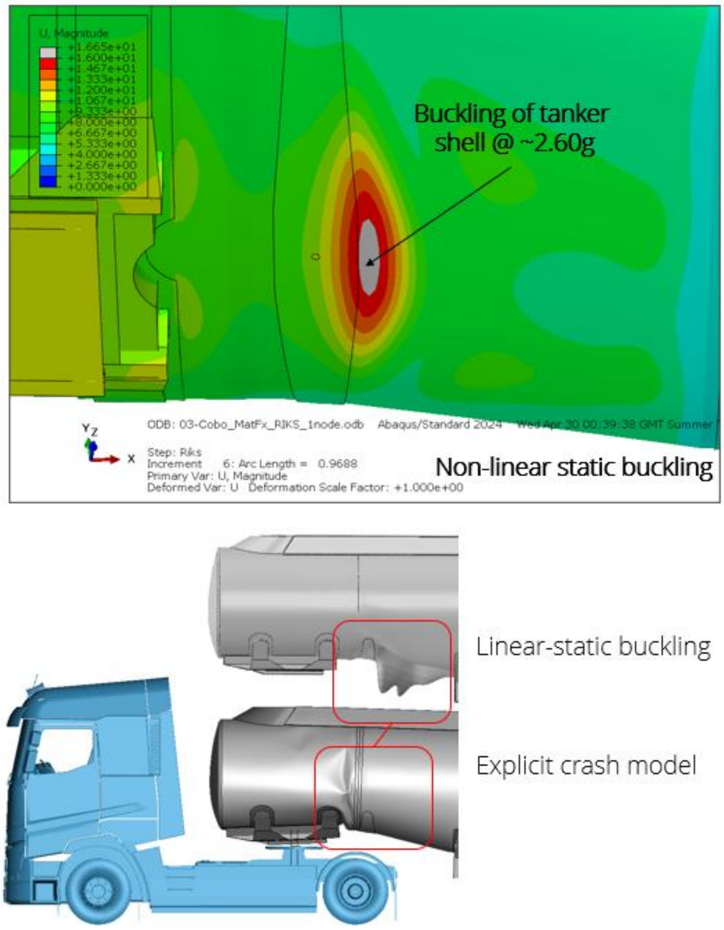


Figure 2-45: Comparison of buckling mode predicted by non-linear analysis compared with that predicted by LS Dyna3D crash (explicit) and linear analyses for tanker #3 self-supported stuffed design

Table 2-7: Comparison of buckling factors predicted by non-linear analysis with those predicted by LS-Dyna3D explicit crash and linear analyses. *Note the 2.64 g predicted by the linear analysis for tanker #2 occurred on the supporting structure and not the tanker shell itself

Tanker	LS-Dyna3D crash analysis: baseline collision (g)	Linear analysis (g)	Non-linear analysis (g)
#1 Self-Supported Banded Design	2.5	2.82	2.63
#2 Supported Stuffed Design	3.5	2.64*	2.44
#3 Self-Supported Stuffed Design	2.5	2.57	2.60

For tanker #1 self-supported banded design, it is seen that all three types of analyses predicted similar initial points of buckling of the shell near the front of compartment 2.

However, the buckling loads predicted were variable. Both the non-linear and linear analyses predicted higher loads than the LS-Dyna3D crash analysis, with the non-linear analysis predicting a little less load than the linear. This was expected because it typically occurs (see Section 2.4.5.1)

For tanker #2 supported stuffed design, it is seen that all three types of analyses predicted a similar initial buckling point of the support structure. However, very different buckling loads were predicted, with the LS-Dyna3D crash analysis predicting a much higher load. Whilst the load predicted by the linear analysis was expected to be lower because it identified the buckling of the support structure only, the load predicted by the non-linear analysis for the buckling of the shell was unexpectedly low.

For tanker #3 self-supported stuffed design, it is seen that both the non-linear and linear analyses predicted a difference in the initial buckling position compared to the crash model, namely, buckling predicted to occur ahead of the banded partition and reinforcing section in the crash analysis, whilst it was predicted to occur behind this region in the non-linear and linear buckling analyses. As mentioned previously, the likely reason for this difference is that the effect of fuel motion was included in the crash model, whereas it was not included in the non-linear and linear buckling models (see Section 2.4.4.2).

It was expected that the non-linear buckling analyses would show an advantage compared to linear analyses in that they would predict behaviour of the tank post the initial buckle. This turned out not to be the case, with the exception of tanker #2, as seen in Figure 2-46, which shows the reaction force against x displacement of the king pin. A likely explanation for this is that tanker #2 possesses some structural stability post buckle because of its supported design, and therefore the solver can find static equilibrium solutions post buckle. In contrast, tankers #1 and #3 are of a self-supported design and have very little structural stability post buckle, i.e. they collapse catastrophically, and therefore the solver cannot find static equilibrium solutions for significant displacements post buckle.

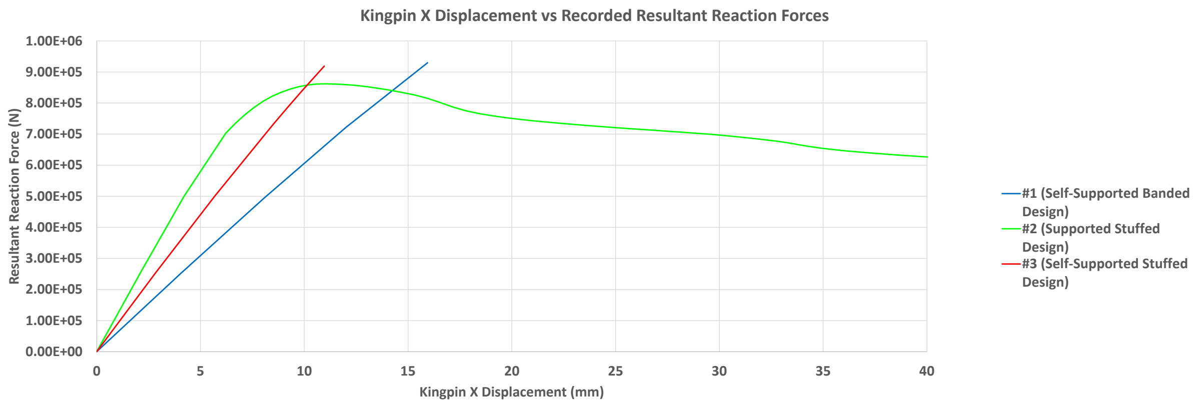


Figure 2-46: Comparison of reaction force against king pin displacement for non-linear tanker analyses

2.4.6 Summary of findings

The LS-Dyna3D crash modelling showed that:

- In a collision with another HGV of severity similar to that of the baseline scenario, all tankers modelled buckled and failed.
- In a collision with a 2 t light vehicle (e.g. electric car) tanker #3 (predicted to withstand lowest loads) did not deform, but in collision with a 4.25 t light vehicle (e.g. electric van), there was some small deformation.
- As expected, supported design #2 sustained higher loads than self-supported designs #1, #3. However, it did not sustain as much load as expected because of buckling of the support structure.
- The unsupported banded design sustained force post buckle better than unsupported stuffed design, because stuffed design has less connection to supporting structure, i.e. partition / baffle, and hence, once buckle starts, there is no structure to provide much resistance to collapse.
- Fuel motion causes increase in peak loads; for g level boundary condition load cases:
    - 1.4 g case, peak loads are ~2.5 g (~+80%).
    - 1.0 g case, peak loads are 2.2 g (~+120%).

- The FE modelling showed that linear and non-linear buckling methods can be used to predict buckling loads. These methods are less expensive than the LS-Dyna3D crash explicit method (linear buckling least expensive), but do not account for the effect of fuel movement. In theory, non-linear buckling should be able to predict post-buckle behaviour, but this was not achieved for self-supported tanker designs, and for the supported design the result did not align with the LS-Dyna3D crash one.
- On the basis of the above and cost, it is recommended that a combination of LS-Dyna3D or linear buckling analysis methods are used for future study and design work, depending on the output required.

Table 2-8: Comparison of tanker peak and buckling loads predicted by LS-Dyna3D crash, linear buckling and non-linear buckling FE analyses

Tanker	LS-Dyna3D Crash model												Linear buckling		Non-linear buckling	
	Baseline (48 km/h) collision				24 km/h collision				2.0 g boundary condition				Buckling loads		Buckling loads	
	Peak longitudinal loads		Buckling loads		Peak longitudinal loads		Buckling loads		Peak longitudinal loads		Buckling loads					
	Load (kN)	Load (g)	Load (kN)	**Load (g)**	Load (kN)	**Load (g)**	Load (kN)	**Load (g)**	Load (kN)	**Load (g)**	Load (kN)	**Load (g)**	Load (kN)	**Load (g)**	Load (kN)	**Load (g)**
#1 self-supported banded	1150	3.3	900	**2.5**	1050	**3.0**	800	**2.3**	900	**2.5**	800	**2.3**	997	**2.82**	930	**2.63**
#2 supported stuffed	1425	4.0	1250	**3.5**	1250	**3.5**	1180	**3.3**	1050	**3.0**	1050	**3.0**	934*	**2.64***	861	**2.44**
#3 self-supported stuffed	900	2.5	870	**2.5**	950	**2.7**	930	**2.6**	900	**2.5**	870	**2.5**	907	**2.57**	919	**2.60**

2.5 Discussion

The research before that reported here (see Section 1.1) found that frontal impacts can present significant risks of substantial releases of flammable liquids, particularly for articulated vehicles with self-supported trailers in collisions with other heavy vehicles. It identified one case in which there was a substantial release and performed modelling that indicated that buckling of a semi-trailer's structure and subsequent failure leading to release could occur at low loads. This raised the question of why more cases involving substantial releases were not identified. Potential elements of an answer to that question included:

- Collisions severe enough to cause buckling of a semi-trailer's structure and associated failure are rare.
- There are few tankers in the fleet susceptible to this failure type.
- The finite element (FE) modelling incorrectly predicted the loads at which buckling and failure occurred, possibly because they were not representative of real-world designs.

The current research has provided information to fully answer the question through performing additional collision analysis, tanker fleet analysis, and FE modelling as follows:

- Collisions severe enough to cause buckling of the semi-trailer's structure and associated failure are rare.

  Collision analysis found that frontal impact collisions with other heavy vehicles severe enough to cause buckling and failure of a semi-trailer's structure **are rare, with an average of one incident every five years**. It also found that frontal impact collisions with light vehicles were about 15 times more frequent, i.e. on average three incidents annually.

- There are few tankers in the fleet susceptible to this failure type.

  The tanker fleet analysis found that the vast majority of tanker semi-trailers in the GB fleet are of a self-supported type design, the same type of design of the semi-trailer tanker involved in the baseline collision which buckled and failed, i.e. the **vast majority of tankers in the fleet are susceptible to this failure type.**

- The finite element (FE) modelling incorrectly predicted the loads at which buckling and failure occurred, possibly because they were not representative of real-world designs

  The FE element modelling found that both the self-supported and the supported typical designs modelled would buckle and fail in a collision with another HGV of similar severity to the baseline collision case identified in the collision analysis. Indeed, the parameter sweep results showed that the severity of the collision - in terms of the impact velocity - would have to be reduced by about 75% (48 km/h to 12 km/h) of the baseline collision for none of the semi-trailers modelled to buckle, and about 60% (48 km/h to 18 km/h) for two of them to buckle, but probably not fail. This indicates a high risk of buckle and failure for all semi-trailers in a collision with another HGV of severity similar to that of the baseline collision. The parameter sweep results, in which the mass of the impacted vehicle was reduced to approximately represent an impact with a light vehicle, showed that significant buckling was unlikely to occur in this type of collision, even with an electric van with a mass of 4.25 t.

These results showed that **the risk of buckling/failure predicted by FE modelling for different collision configurations and the collision analysis findings were consistent**, i.e. high risk of buckling/failure in collision with another HGV, but the frequency of these collisions is very low, explaining why only one example of this failure type was identified from collision data. The greater frequency of collisions with light vehicles identified in the collision data did not result in tank failure, and FE modelling also showed extremely low risk of buckling/failure in these collisions. This consistency helps to provide confidence in the FE modelling results and the representativeness of the models, i.e. **the FE models used in the previous work predicted semi-trailer tanker failure at low loads – a result that is also observed in the current, improved models**.

The FE modelling work also showed some nuances in the buckling and failure loads and mechanisms between the different types of semi-trailer tanker designs modelled. Firstly, it showed that peak (failure) loads were much higher for the supported type design compared to the self-supported ones (4.0 g c.f. 2.5 – 3.3 g)[16], although they were not high enough to offer any possibility of the design maintaining its integrity in a collision with another heavy vehicle with a severity equivalent to the baseline collision. Secondly, it showed that the design of the king pin assembly and its attachment to the tank could also affect the magnitude of the peak (failure) loads withstood; 3.3 g for the # 1 self-supported stuffed design which had a king pin assembly structure connected well to the tank structure, i.e. structural elements to spread load on tanker shell and connect to the partition structure, but only 2.5 g for the # 3 self-supported banded design which only had structural elements to spread load on tanker shell.

In summary, the main explanation for the apparent discrepancy between the FE modelling—which predicted failure of semi-trailer tankers under relatively low loads during frontal impacts—and the limited number of observed failures in real-world collision data, is that such collisions are very rare.

Interestingly, the work performed also found that, even though all semi-trailer tankers in the fleet, including those of a supported design type, would buckle and fail in a frontal impact with another heavy vehicle of severity of the baseline collision, delta V ~ 15 mph (24 km/h), all of them maintain their integrity in frontal impacts with light vehicles. This is somewhat serendipitous given that frontal impact collisions with light vehicles are about 15 times more frequent than those with heavy vehicles. However, it also indicates that the integrity of new novel semi-trailer tanker designs should not be allowed to fall below that of current ones

[16] Note: The support structure buckled at much lower loads than the tanker shell which indicates that improvements to the design of the support structure could probably further increase the peak (failure) loads for this type of design, although probably not enough to maintain integrity in a collision with another heavy vehicle with a baseline collision level of severity.

because that could lead to them failing in collisions with light vehicles and would ultimately result in many more flammable liquid releases.

2.6 Implications of findings for extra-large tanker designs

As mentioned in Section 2.1.3, the main regulatory requirements for semi-trailers tankers for king pin loading are contained in EN 13094:2022, para 6.3.2. The requirement relevant for self-supported tankers for frontal impact is the longitudinal one which requires that the design stress for tank shells and their attachments is not exceeded for loads of the maximum design mass multiplied by 2 g acting in the direction of travel. The authors understand that this requirement is related to loads which could potentially be experienced in service because of heavy braking. Even though it is possible that structures could potentially buckle elastically and fail under loads of this magnitude, the authors understand that the requirement is related to the assessment of design stress only and does not include a requirement to assess elastic buckling.

However, the FE modelling showed that for current typical tankers with a maximum mass of 36 t buckling occurs at loads greater than 2 g, i.e. 2.5 to 3.5 g depending on their design[17], which indicates that there is no need to add an elastic buckling requirement for these tankers.

Should the widespread use of longer and/or heavier vehicles for general haulage prompted by the European Modular System (EMS) in Great Britain be considered, this could see the introduction of extra-large tank-vehicles (i.e. those with a gross capacity which exceeds about 45,000 litres), including FL vehicles should also be permitted. These extra-large tank vehicles might have weights of up to 60 tonnes and lengths of up to 25.5 m compared to the current normal maximum in GB of 44 tonnes and 16.5 m.

If it assumed that an extra-large tank vehicle is developed with a gross combination weight of 60 t, then the semi-trailer weight would be 52 t (if 8 tonne tractor unit weight assumed as for the modelling of the current typical conventional tankers). If it also assumed that the design of this vehicle in the cone area is similar to current conventional designs[18], the following calculation indicates that elastic buckling could occur at loads below 2 g for extra-large tank vehicles:

- For conventional tankers, buckling within cone area occurs at loads of ~ 2.5 g
- Assuming tanker decelerations are similar, king pin loads for 52 t extra-large tank vehicle will be greater than for conventional tanker by a factor of their weight ratio, i.e. 52 / 36 = 1.44
- Therefore, if design in cone section is similar buckling of extra-large tank vehicle could occur at loads of 2.5 g / 1.44 = 1.7 g which is less than 2.0 g.

[17] Measured in simulation of baseline collision, see Table 2-8;

Note that in parameter sweep with 1.4 g loading, peak king pin loads of ~2.5 g measured for all semi-trailers without significant buckling of any semi-trailer although there was some deformation of tanker #3, see Section 2.4.3.2 (Parameter sweep 2) and Appendix B.2

[18] Calculations have been performed that show that for the typical UK fleet semi-trailer designs studied, the design stress safety factors for the cone area behind the king pin for the typical UK fleet semi-trailers design required to show compliance with EN 13094 para 6.3.2 for 2 g loads in the direction of travel are sufficiently large to also allow compliance for extra-large semi-trailer tanker designs with weight of up to 52 t

On this basis, to help future proof international regulation, it is suggested that EN 13094 is updated to add a requirement to paragraph 6.3.2 to ensure the structure of extra-large tankers in the king pin area can withstand a load of at least 2 g as follows:

> *6.3.2 Shells, their attachments and their structural equipment shall be designed to withstand the forces and dynamic pressures resulting from the combination of (P_{ta} + P_{ts}) with, separately, each of the following, without exceeding the design stress in 6.7:*
>
> *- in the direction of travel, an acceleration of 2 g on the maximum design mass (for the front end, only the maximum mass of the substance carried in the first (front) compartment shall be taken into account);*
>
> *………….*
>
> *In addition, for longitudinal accelerations (2 g) of self-supporting tanks (trailers without a longitudinal framework upon which the tank is supported), the following shall be taken into account:*
>
> > *a) for front attachments, the maximum design mass of the trailer shall be deemed to act where the coupling device is attached to the tank; and*
> >
> > *b) for rear attachments, the maximum design mass of the tank shall be deemed to act on the attachments of the tank to the running gear. The non-uniform mass distribution on the tank saddles shall be taken into account.*
>
> **Where the total gross capacity exceeds [45,000 litres] irrespective of whether the tank is supported or self-supported the shell cylindrical and conical sections, structural connecting and coupling device shall also be checked for longitudinal accelerations (2 g) where required in the regulation including potential of buckling load collapse and stresses in Table A.1.**
>
> Note: Current standard text in light grey, suggested added text in bold.

As part of this project, this suggestion was discussed with the CEN/TC 296/WG2 expert group who advise on amendments to EN 13094. Some of the experts considered the current version of EN 13094, para 6.3.2 to include a requirement for the assessment of buckling already. If this interpretation is correct, then the suggestion above for amendment of EN 13094 would not be necessary, but amendments to clarify the current requirement may be beneficial.

2.7 Overall conclusions and suggested next steps

2.7.1 Overall conclusions and summary of key findings

2.7.1.1 Overall conclusions

The aim of the frontal impact research performed was to answer the question arising from previous research. This was that given that FE modelling work predicted failure of a semi-trailer tankers at low loads, why weren't more examples of failures found in the collision analysis**. The research concluded that the main answer to this question is that the type of collision in which these failures occur, i.e. frontal impact with another heavy vehicle, are very rare, with an average of one incident every five years**.

The research also concluded that even though all semi-trailer tankers in the fleet, including those of a supported design type, would buckle and fail in a frontal impact with another heavy vehicle of severity of the baseline collision, delta V ~ 15 mph (24 km/h), all of them maintain their integrity in frontal impacts with light vehicles. This is somewhat serendipitous given that frontal impact collisions with light vehicles are about 15 times more frequent than those with heavy vehicles. However, **it also indicates that the integrity of new novel semi-trailer tanker designs, such as extra-large tankers, should not be allowed to fall below that of current ones because that could lead to them failing in collisions with light vehicles and ultimately result in many more flammable liquid releases.** To address this concern, potential amendments to the ADR EN 13094 referenced standard are suggested in Section 2.7.2 below for potential future extra-large tank vehicles.

2.7.1.2 Summary of key findings

A summary of the key findings of the frontal impact research work is provided below.

Activity 1: Collision analysis

- Frontal impact collisions with other heavy vehicles severe enough to cause buckling and failure of a semi-trailer's structure are rare, with an average of one incident every five years. Collisions with light vehicles, e.g. cars, are about 15 times more frequent, i.e. an average of 3 incidents annually.
- From ten years of collision data, only one incident in which tank rupture was confirmed was found. *Note that throughout this report this is referred to as the baseline collision.*

Activity 2: Fleet analysis

- The vast majority of tanker semi-trailers in the GB fleet are of a self-supported type design, the same type of design of the semi-trailer tanker involved in the baseline collision which buckled and failed.

Activity 3: Finite element (FE) modelling

- To enable a representative LS-Dyna3D crash model of the semi-trailer to be built the following was required:

- A lumped parameter model approach to apply decelerations to the king pin because they were not independent of the semi-trailer's behaviour, i.e. failure of the semi-trailer modified (reduced) the decelerations applied.
- Detailed representation of the fuel load to account for its movement when the semi-trailer is decelerated, i.e. the fuel moves forward in the tank when the deceleration is initially applied but then is brought to a sudden halt when its motion is stopped by the partitions and baffles which causes transient loads substantially above the deceleration applied to the semi-trailer.

- The results of the modelling of the baseline collision and the parameter sweeps indicate that:
  - All current typical designs of semi-trailer tankers would buckle and fail (rupture) in a frontal impact with another heavy vehicle of similar severity to that of the baseline collision.
  - No current typical designs of semi-trailer tankers should fail in a frontal impact with a light vehicle with impact speeds similar to those in the baseline collision.
  - Typical initial buckling loads for current conventional semi-trailer tankers are in the range of 2.5 to 3.5 g, depending on their design.
- Linear buckling analysis provided a reasonable indication of the semi-trailer buckling loads predicted by the LS-Dyna3D crash analysis and thus could potentially be used to provide a lower cost alternative for use in tanker design by manufacturers. An exception was for the supported type design because buckling of the support structure prevented identification of the buckling load for the shell structure. It should also be noted that because linear buckling analysis does not represent the fuel arrest loading fully as with the LS-Dyna3D crash model, the buckling location identified differed between the Dyna3D and linear analyses for the self-supported stuffed semi-trailer design modelled.
- Non-linear buckling analysis did not provide information about the behaviour of the semi-trailer structure post initial buckle as expected, except for tanker #2 supported stuffed design. This was because the self-supported designs have little structural stability post initial buckle and collapse in a catastrophic manner, thus the solver cannot find any static equilibrium solutions. For tanker #2, the supported design which had some structural stability post initial buckle, some behaviour post buckle was found, but only for a limited king pin displacement of 40 mm. Given that this technique is also more expensive than linear buckling analysis it does not offer any advantage for use in tanker design by manufacturers.

Collision analysis and FE modelling:

- Comparison of these results showed that the risk of buckling/failure predicted by FE modelling for different collision configurations and the collision analysis findings were consistent, i.e. high risk of buckling/failure in collisions with another HGV, but the frequency of these collisions is very low, so only one example of failure found; also frequency of collision with light vehicle is more frequent, but FE modelling showed extremely low risk of buckling/failure in these collisions, explaining why no examples of failure were identified in the collision data.

Extra-large tank vehicles:

- For extra-large semi-trailer vehicles, if their design in the conical area behind the king pin is similar to current conventional designs, which is potentially feasible, simple calculations show that compared to conventional semi-trailer tankers, their additional weight could compromise their elastic buckling strength and thus safety. For example, a 52 t extra-large semi-trailer vehicle could have a buckling strength of 1.7 g compared to a value of 2.5 g for a conventional semi-trailer.

2.7.2 Suggested next steps

The research work showed that based on the current EN 13094 requirements, it is theoretically feasible to obtain approval for an extra-large tank semi-trailer vehicle design with structure in the cone area behind the king pin that could have a buckling strength less than 2 g, substantially less than that of conventional semi-trailer tankers. This could possibly lead to failure of these designs in collisions with light vehicles. On this basis, to help future proof regulation, it is suggested that EN 13094 is amended to add a requirement to paragraph 6.3.2 to ensure the structure of extra-large tankers in the king pin area can withstand a load of at least 2 g, as follows:

> 6.3.2 *Shells, their attachments and their structural equipment shall be designed to withstand the forces and dynamic pressures resulting from the combination of (P_{ta} + P_{ts}) with, separately, each of the following, without exceeding the design stress in 6.7:*
>
> *- in the direction of travel, an acceleration of 2 g on the maximum design mass (for the front end, only the maximum mass of the substance carried in the first (front) compartment shall be taken into account);*
>
> …………..
>
> *In addition, for longitudinal accelerations (2 g) of self-supporting tanks (trailers without a longitudinal framework upon which the tank is supported), the following shall be taken into account:*
>
> > *a) for front attachments, the maximum design mass of the trailer shall be deemed to act where the coupling device is attached to the tank; and*
> >
> > *b) for rear attachments, the maximum design mass of the tank shall be deemed to act on the attachments of the tank to the running gear. The non-uniform mass distribution on the tank saddles shall be taken into account.*
>
> **Where the total gross capacity exceeds [45,000 litres] irrespective of whether the tank is supported or self-supported the shell cylindrical and conical sections, structural connecting and coupling device shall also be checked for longitudinal accelerations (2 g) where required in the regulation including potential of buckling load collapse and stresses in Table A.1.**
>
> Note: Current standard text in light grey, suggested added text in bold.

As part of this project this suggestion was discussed with the CEN/TC 296/WG2 expert group who advise on amendments to EN 13094. Some of the experts considered the current version of EN 13094, para 6.3.2 to include a requirement for the assessment of buckling already. If this interpretation is correct, then the suggestion above for amendment of EN 13094 would not be necessary, but instead amendments to clarify the current requirement may be beneficial.

3 Rollover

The objective of this part of the research was to use the outputs and understanding from the previous research[19]:

- To investigate the safety implications for extra-large tank vehicles in rollover, and
- If any concerns are identified, to determine how they may be best addressed in view of ADR requirements, referenced standards, and technical codes.

To meet this objective, the approach taken was to compare the energy absorption capability of current conventional tank vehicles, with those of hypothetical heavier extra-large tank vehicles of different designs. The premise was that the extra-large tank vehicles should be at least as safe in rollover as current conventional tankers. The approach was based on the findings of the previous research, which showed that, in order to maintain its integrity in a rollover, a semi-trailer tanker must absorb the topple impact energy, which is related to its weight, through deformation of its shell and strengthening element structure, without rupturing.

This section is structured as follows. Firstly, the relevant findings of the previous research are described. Next, the results of the research work are reported; namely the comparison of the energy absorption capability of current conventional tankers and potential hypothetical heavier extra-large ones. Following this, there is a discussion section, and finally, there is a section detailing the conclusions and suggested next steps.

[19] Edwards et al. 2023. '*Research on performance test procedures for petroleum road fuel tankers: Summary report*'. TRL Published Project Report PPR2027. https://www.trl.co.uk/uploads/trl/documents/PPR2027-Research-on-performance-test-procedures-for-road-fuel-tankers---summary-report--v-final1.1-041023.pdf

3.1 Background - findings of previous research

The following sub-sections give a summary of the findings of the previous research, details of which can be found in the referenced report[19]. The main outputs of the research were a collision analysis and the (partial) development of performance-based test methods for rollover, together with an understanding (from associated finite element modelling) of the test parameters relevant to current tanker designs, and a route to their future adoption in standards and regulation in the form of an outline technical code for rollover resilience.

3.1.1 Collision analysis

A collision analysis for flammable liquid (FL) tankers was performed by Apollo Vehicle Safety[20] using the GB national collision database (STATS19 2009 to 2018 data), supplemented with data available from other sources such as DfT ADR reports, the Road Accident In-Depth Study (RAIDS) database, and relevant collision case studies identified in the literature worldwide.

The analysis found that almost all significant releases of flammable liquids (FL) arise from traffic collision incidents involving a rollover or a collision with another heavy vehicle (Table 3-1), and noted that these release incidents represented only a small proportion of all traffic collision incidents involving FL vehicles.

Table 3-1: Summary of risk for main collision mechanisms (STATS19 data 2009-2018)

Collision mechanism	Proportion of data set that involved each collision mechanism			
	All GB HGVs (>3.5t) (2009-18 average)	All GB FL vehicles (2009-18 average)	Collisions involving high risk of a spill	Collisions involving a spill*
Rollover	3.7%	5.2%	38% – 69%	60% - 95%
Vehicles damaged at rear by another heavy vehicle	2.6%	3.4%	5% - 17%	0% - 12%
Vehicles damaged at front by another heavy vehicle	3.9%	2.0%	7% - 10%	8% - 9%
Vehicles damaged at side by another heavy vehicle	2.9%	3.6%	0%-9%	0% - 8%
Other			15% - 27%	Not documented, thought to be low

[20] Knight I and Dodd M (2019) ‘*Performance test procedures for petroleum road fuel tankers. DRAFT report for Part A – Review and analysis of accident data, impact conditions and regulations*’. London: Department for Transport

* Note: Tanks with emergency pressure relief valves are designed to allow some limited quantities of fuel to spill in an incident to prevent a pressure surge causing more damage to the tank. The literature and data sources used to arrive at these estimates did not consistently record the mechanisms by which goods were released in impacts, or detail the quantities released. Thus, this estimate includes all releases of hazardous goods, including those that may have involved minor leaks by design.

Further analysis of the type of the rollover for flammable liquid tankers revealed the following:

- 31% involved a simple on carriageway rollover.
- 13% involved a run-off road rollover.
- 56% involved a rollover and an impact, but in an unknown order.

It should be noted that the STATS19 flammable liquid tanker sample size was small.

Studies in the literature have reported that the simple on carriageway rollovers typically involved a 90 degree topple and slide (Figure 3-1). Without impact, there was generally no spillage, except if the topple was at high speed. However, when an object is impacted, spillage is frequently observed.

Figure 3-1: Offside view of damage to tank as a result of 90 degree roll and slide to rest (left) and close up of hole in tank showing signs of abrasion (right)

3.1.2 Development of outline technical code – rollover: topple and impact

The aim of the technical code was to detail 'performance based' test methods which could be used to demonstrate the resilience (resistance to rupture, abrasion, and penetration) in rollovers of novel metallic gravity discharge fuel tankers, which may deviate from ADR requirements. The acceptance requirements for the test methods, where defined, were set to demonstrate a performance level which was equivalent, as far as practical, to the performance of a conventional tanker which currently meets the ADR requirements.

The collision analysis described above identified that a common rollover scenario is firstly a 90° on-road rollover where the side of the tanker impacts the ground; and then secondly, a period of sliding along the ground. Before the tanker comes to rest it may strike an object and this could penetrate the tanker shell.

On this basis and because of practicalities, such as the need for a repeatable test method, it was decided that the three main distinct events in a rollover should be assessed in three separate tests in the code:

- Topple test[21];
- Abrasion test; and
- Penetration test.

For the topple test, two potential test methods were selected for inclusion in the technical code, namely a full scale topple test, and a sub-section drop test (Figure 3-2). Although previous research had shown that a full scale topple test was representative of a typical real-world rollover[22], its cost was high. It was clear that the cost of a sub-section drop test would be lower and hence more acceptable, so FE modelling was used to develop a sub-section drop test which was representative of a typical real-world rollover.

Topple (full tanker)

Drop (sub-section)

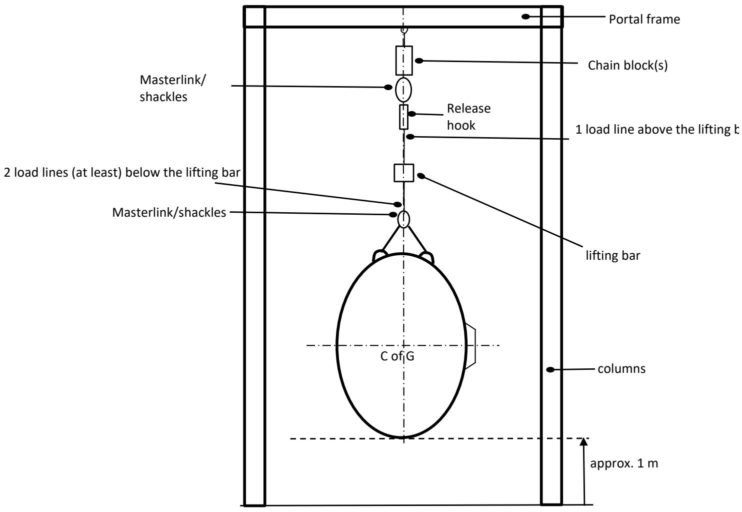


Figure 3-2: Full scale topple and sub-section drop tests selected for inclusion in the technical code

FE modelling using three metallic tankers representative of current designs with different shell to strengthening element joint types (2 banded, 1 stuffed – see Figure 3-3) was used to show that the deflections and likelihood of major loss of containment experienced by tankers

[21] The term 'topple' is used to describe the roll of a tanker through 90° and onto its side assuming there is no forward motion. The effect of forward motion is considered in the abrasion test which simulates the tanker sliding along the ground.

[22] Robinson B, Webb D, Hobbs J, London T (2015). Technical assessment of petroleum road fuel tankers – summary report all work packages. Available from: https://www.gov.uk/government/publications/petroleum-fuel-tankers-technical-assessment-november-2015 , note – download work package 3 zip file.

in real-world topple scenarios could be replicated in a suitably specified, two-compartment subsection drop-test (or a full-scale physical topple test).

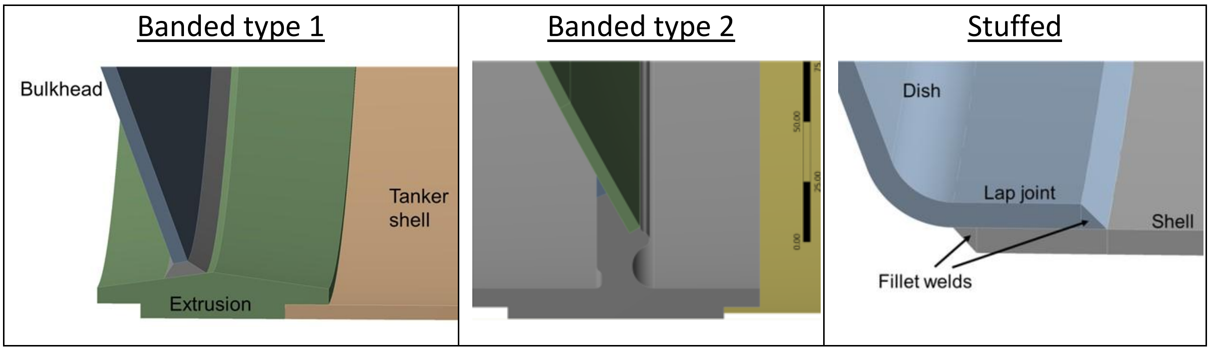


Figure 3-3: Different joint designs in the three FE models used to develop sub-section drop test method. Note that since its update in 2020, EN 13094 no longer permits tanks with a banded type 1 design because previous work[23] found it to be particularly susceptible to failure

The FE modelling performed included the following:

- Full tanker topples and drops to ascertain relationships between topple and drop. Key findings were:
    - Differences in fuel motion and its effect - there is much more fuel motion post topple compared to post drop, which means less energy for the structure to absorb in a topple and consequently deflections are less for equivalent impacts, i.e. impact speed matched at the point of impact.
    - Deflections vary along the length of the tanker – it was found that this was dependent on the energy distribution (i.e. weight distribution) with more deflection occurring at the rear compared to the front of the tanker. Relationships for this variation were established.
- Sub-section drop parametric studies which established relationships between impact energy and structural response parameters, mainly deflection. Quadratic relationships between energy and deflection were established with a high coefficient of determination, $R^2 > 0.9$ (Figure 3-4).

[23] London T (2016): TWI Report No. 25272/1/16: Department for Transport Technical Assessment of Petroleum Tankers: Assessment of BS EN 13094 Lap and Partition Joint Designs. Available from: https://assets.publishing.service.gov.uk/media/5fc65475d3bf7f7f575b4369/Technical-assessment-of-BS-EN-13094.pdf

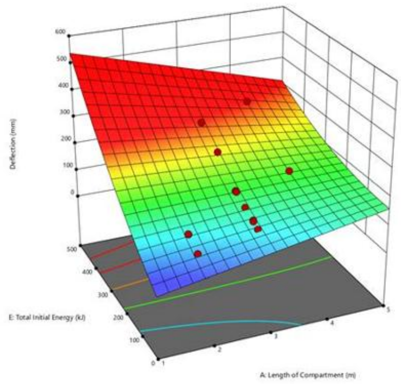


Figure 3-4: Quadratic relationship established between impact energy and deflection for drop tests

- The use of forming limit diagrams which is a widely established method in assessing if sheet metals that are strained in the forming process are likely to fail structurally. The forming limit can be expressed as a ratio of the major principal strain to the major limit strain known as the 'omega' value (ω) where anything above an omega value of 1 means that failure is likely; and anything below 1 means that failure is not likely (Figure 3-5).

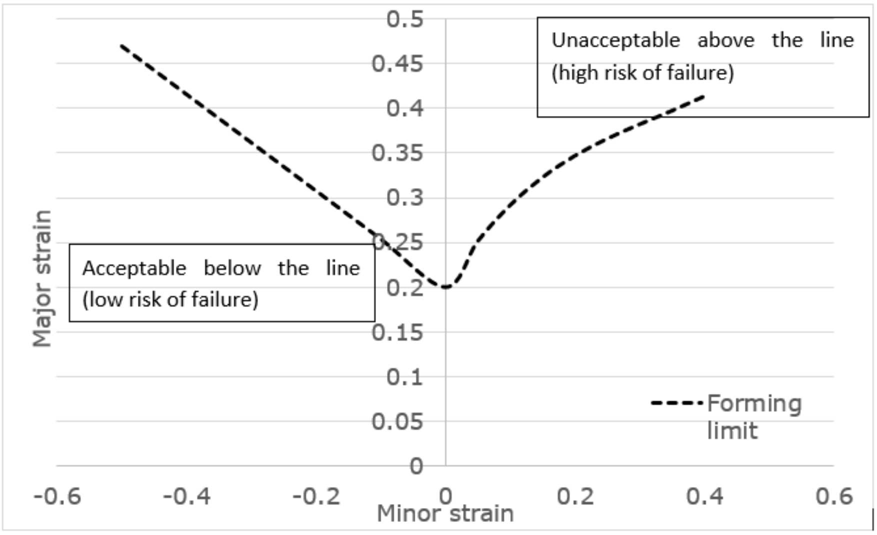


Figure 3-5: Forming limit diagram used to determine likelihood of tank failures

It was found that the tank stiffness and its ability to absorb energy before failure (indicated by omega value) were highly dependent on the type of joint design. Figure 3-6 shows that for an equivalent deflection the energy absorbed per partition (strengthening element) is about twice as much for a banded type 2 design compared with a stuffed design. This indicates the much higher stiffness of the tank with the banded type 2 design.

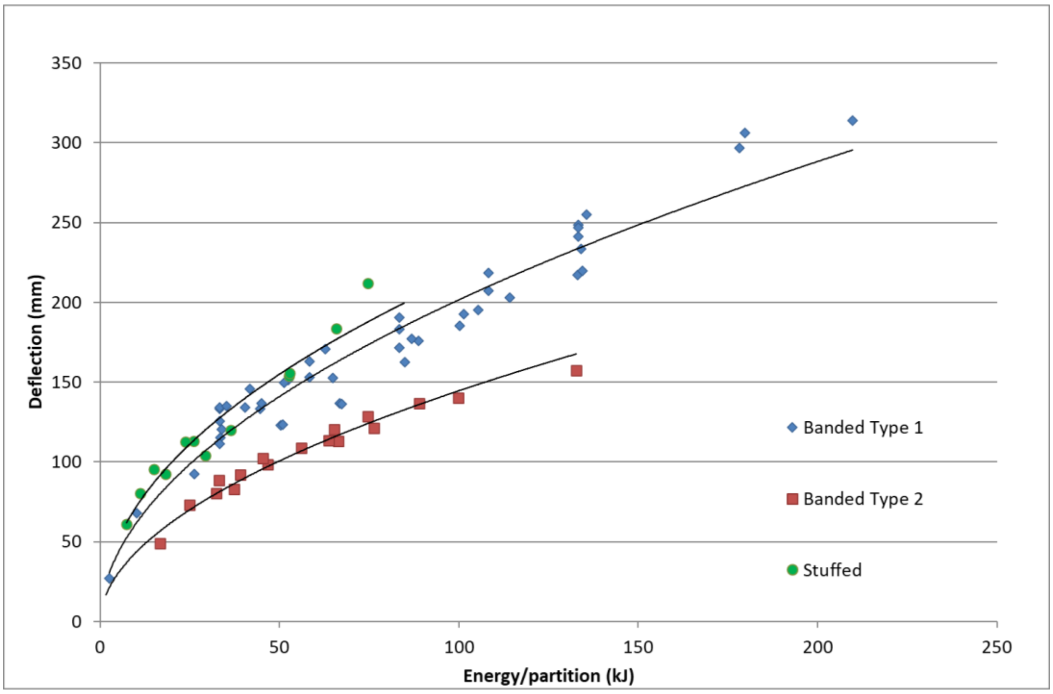


Figure 3-6: Comparison of tank sub-section deflection against impact energy absorbed per strengthening element for banded type 1, banded type 2 and stuffed type joint designs

Figure 3-7 shows that a tank with a stuffed type joint can deflect much further before omega values exceed 1, indicating a likelihood of failure. This is good because it needs to deflect more to absorb an equivalent amount of energy given that it is less stiff.

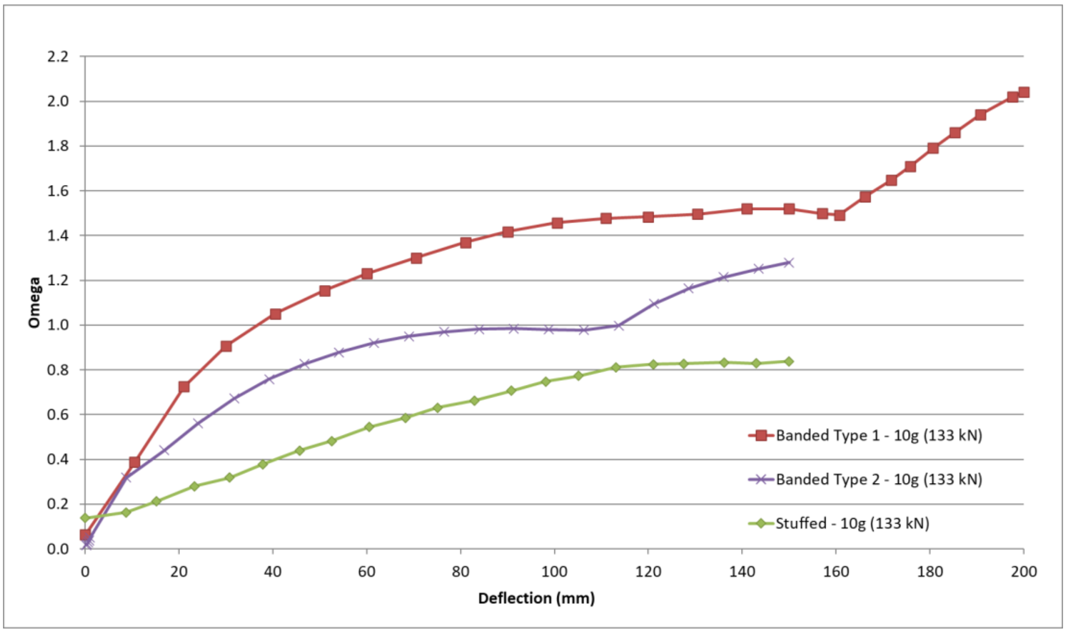


Figure 3-7: Comparison of tank sub-section deflection against omega for banded type 1, banded type 2 and stuffed type joint designs

Figure 3-8 shows that the omega values are below 1 for both the stuffed and banded type 2 joint designs up to impact energy absorbed per partition values of ~ 60 kJ, although the omega values for the banded type 2 design are much closer to 1. This indicates that both these designs should not fail in a topple at energies up to ~ 60 kJ, although a stuffed design has greater potential (i.e. omega less than 1) at energies above ~60 kJ.

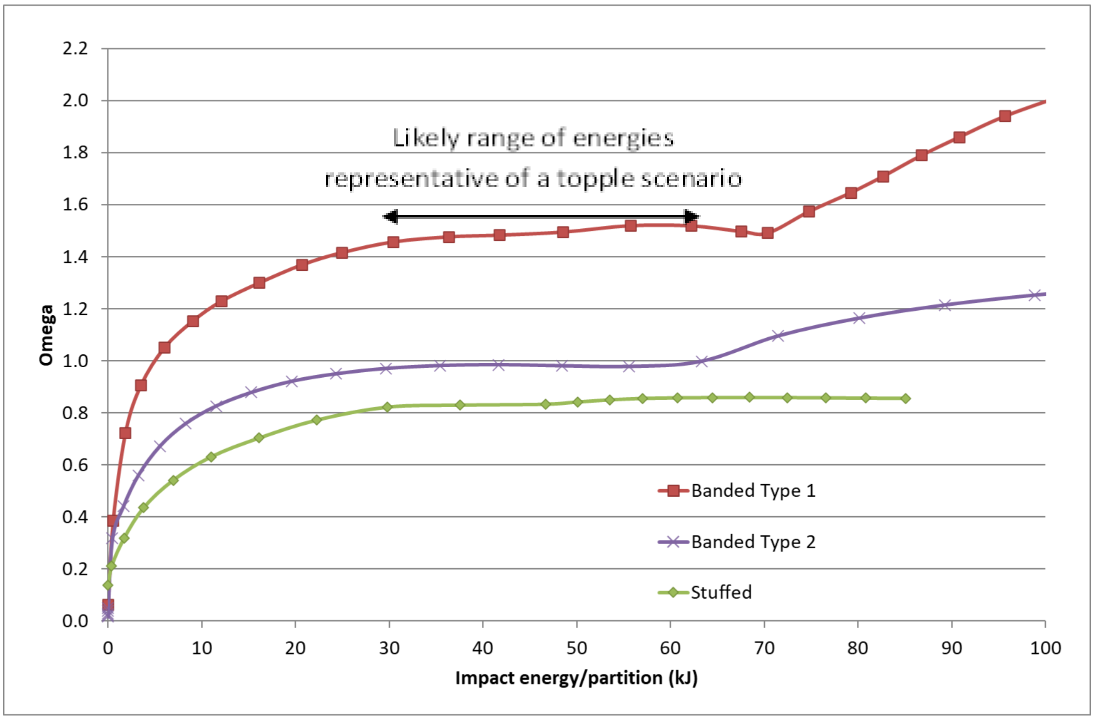


Figure 3-8: Comparison of omega against impact energy per partition for banded type 1, banded type 2 and stuffed type joint designs

Key learnings from this work which were used to develop the sub-section drop test method options included:

- Impact energy is mainly absorbed in the deformation of the dish part of the strengthening elements (partitions / baffles).
- Compared to a drop test, in a topple test, substantial energy (~ 25%) remains in fuel motion following ground impact.
- Partition / baffle to shell joint design has a major influence on the stiffness of the strengthening element (partition) and its failure.

Four test method options for assessment of topple and impact were included in the outline technical code, as described in Table 3-2. Apart from Option 1, which is a physical full-scale test, all the options contain a two-compartment physical sub-section drop test to either validate a FE model (option 2), or demonstrate the integrity of a worst-case sub-section (Options 3a, 3b, 4).

Table 3-2: Summary of technical code options to demonstrate topple impact performance

Option No.	Title	Description of Approach	Applicability Limitations	Basis of Acceptance
1	Physical full scale topple test (worked example described in the code)	A full-scale physical topple test of a tanker fully-loaded with water (no finite element analysis is required in this option).	None. Applicable for all tanker designs within the scope.	Integrity of the complete tanker in a physical topple test – no leaks of water to the environment.
2	FE model of full-scale topple test with validation using subsection model and physical drop test. (worked example described in the code)	A finite element model of a full-scale tanker topple which shall contain a method to predict failure. This full-scale model will be validated by 'induction/inference' and not by real-world topple data; i.e. the full-scale model will be used to build an appropriate two-compartment subsection model which will be used to model an appropriate drop test. This will be validated with a physical test.	None. Applicable for all tanker designs within the scope.	Integrity of tank predicted by full-scale FE model once model validation complete – no leaks to the environment predicted in the full-scale FE model, and no leaks to the environment in the subsection drop test.
3	Physical drop test of worst-case subsection 3a. Worst case subsection identified using FE model 3b. Worst case subsection identified using empirical assumptions. (worked example described in the code)	This option is based on identifying the tanker compartment that is at the highest risk of failing, i.e. the 'worst case' one. The integrity of this 'worst-case' compartment is assessed in a two-compartment sub-section drop test, the height of which is chosen to be representative of the loading conditions that the compartment would experience in a topple. Therefore, demonstration of integrity in the drop test provides evidence of the whole tanker integrity in a topple. The parameters for the drop test will be based on calculations for the Basic impact energy requirements and will be adjusted to allow for variations in deflection along the length of the tanker, which can be determined using: • A finite element model of a complete tanker (Option 3a) to find the worst-case compartment(s) as described above; or • a *standard length adjustment factor* which is described in the code.	Option 3a is currently not applicable for tankers where the joint design is not the same at each partition. Option 3b is not currently applicable for the following types of tanker designs: • tankers where the joint design is not the same at each partition • swept end designs • stuffed designs (because the supporting modelling development work did not consider these types of design)	Integrity of the two-compartment subsection in a physical drop test – no leaks to the environment.
4	Physical drop test of subsection for joint design only	This option is the same as option 3b except that the impact energy/partition is fixed, and the acceptance is only for the circumferential joint designs between band and shell, and the joint between partition and band/shell where relevant.	Applicable for assessing only the joint designs on all tankers in scope.	Integrity of the two-compartment subsection in a physical drop test (no leaks to the environment).

3.1.3 Findings relevant to extra-large tank vehicles

The specific findings from the research and parts of the technical code described above which are relevant to extra-large tank vehicles include:

- Research finding: The impact energy in a topple is mainly absorbed by deformation of the dish part of strengthening elements (partitions / baffles)
    - This finding is relevant because, given that extra-large tank vehicles will weigh more and thus have more impact energy to absorb in a topple than a conventional sized tanker, they will require more strengthening elements (or equivalent, e.g. stronger strengthening elements) to offer an equivalent level of safety and absorb the impact energy without tank failure and resulting fuel release.
- Technical code option 3b: This option includes a method to calculate the average and maximum impact energies that strengthening elements of a tanker must absorb to maintain its integrity in a topple. Based on these energy requirements, it determines the drop height for a 2 compartment sub-section test to demonstrate that the tanker has sufficient energy absorption capability to maintain its integrity in a topple.
    - This code option is relevant because the calculation can be used to estimate how much energy absorption capability a potential extra-large tank vehicle will require to offer an equivalent safety level to a current conventional tanker.

The calculation steps for technical code option 3b are as follows:

1. Calculate centre of gravity height change in topple, i.e. $\Delta h = h_t - h_i$. For a conventional tanker example, this was measured to be 2.342 m – 1.275 m = 1.067 m.

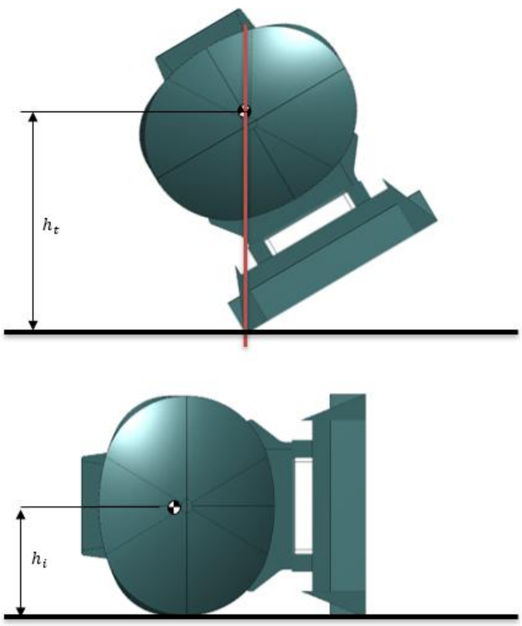


Figure 3-9: Change in tanker centre of gravity height, Δh, between point of topple, h_t, (top) and impact, h_i, (bottom)

2. Calculate potential energy (PE) of topple = m * g * Δh where m is mass of semi-trailer tanker
3. Calculate average energy absorbed per strengthening element in a drop test
   - Multiply by 0.73 to account for energy retained in fuel motion in topple test
   - Divide by number of strengthening elements (N)

   Average energy per strengthening element = (PE* 0.73)/N
4. Calculate two compartment sub-section vertical drop test energy
   - Calculate maximum energy per strengthening element, i.e. that absorbed by elements in worst-case sub-section
     - Multiply (average energy per element) by length factor squared (1.25^2) to account for variation in deflection along length and increased energy absorbed by worst case sub-section. Also, multiply by safety factor (1.1).
     - Max energy per strengthening element

       = (Average energy per strengthening element) * 1.25^2 * 1.1
   - Calculate sub-section vertical drop test energy
     - Multiply by number of strengthening elements, which is 3 for a two compartment sub-section.
     - Test energy

       = (Max energy per strengthening element) * 3

3.2 Comparison of energy absorption capability of current conventional tankers and hypothetical extra-large tank designs

The calculation for option 3b of the technical code, described in Section 3.1.3 above, was used to compare the energy absorption per strengthening element for current conventional tankers and two potential hypothetical 52,000 kg extra-large tank designs with an elliptical cross-section design. These hypothetical designs featured 10 and 13 strengthening elements, respectively (Table 3-3). The extra-large tanker designs were chosen on the basis of a 60,000 kg gross combination weight (tractor weight assumed to be 8,000 kg) practical type design that could meet the axle weight limits for GB and the requirements of the current EN 13094 standard.

It is seen that the extra-large tank design with 10 strengthening elements has a maximum energy absorption per strengthening element of about 68 kJ, which is similar to that of a conventional tanker with 7 strengthening elements and a stuffed joint design. In contrast, the design with 13 strengthening elements has a much lower average energy absorption of about 52 kJ, which is closer to that of conventional tankers with a larger number of strengthening elements and a banded type 2 joint design, in particular one with 10 elements which is known to perform well in a topple test.

For tanks with an elliptical cross-section, the main requirement related to the number of strengthening elements in the current EN 13094 standard is paragraph 6.8.2.2 (a) which states:

for shells with a circular and/or elliptical cross-section including combinations of those cross-sections having a maximum radius of curvature of 2 m, the shell is equipped with strengthening elements comprising partitions or surge plates, or external or internal rings, so placed that at least one of the following conditions shall be met:

1) the distance between two adjacent strengthening elements is less than or equal to 1.75 m;

2) the capacity contained between two partitions or surge plates is less than or equal to 7,500 l.

It should be noted that calculations were performed showing that both the potential hypothetical extra-large tank designs considered could be designed practically such that they met this requirement. However, whereas both could be designed easily to meet the option 1 requirement, it would be much easier to meet the second option with a 13 element type design, because the greater number of elements allows designs with more (and hence smaller) compartments and/or baffles.

Table 3-3: Comparison of maximum energy absorbed per strengthening element for current typical UK fleet tankers and potential hypothetical extra-large tanker designs

Technical code option 3 – drop test energy calculation Step and formula	**Typical UK fleet (7 strengthening elements, stuffed type joint design)**	**Typical UK fleet (8 strengthening elements, banded type joint design)**	**Typical UK fleet (9 strengthening elements, banded type joint design)**	**Typical UK fleet (10 strengthening elements, banded type joint design)** Shown to perform well in topple test	**Extra-large 1 (10 strengthening elements, banded type joint design)**	**Extra-large 2 (13 strengthening elements, banded type joint design)**
1. Centre of gravity change of height in topple = 1.067 (assume all tankers similar)	1.067 m	1.067 m	1.067 m	1.067 m	1.067 m	1.067 m
2. Potential energy in topple (PE) = mgh	m = 36,000 kg PE = 377 kJ Vol ~ 42,000 l	m = 36,000 kg PE = 377 kJ Vol ~ 42,000 l	m = 36,000 kg PE = 377 kJ Vol ~ 42,000 l	m = 36,000 kg PE = 377 kJ Vol ~ 42,000 l	m = 52,000 kg PE = 544 kJ Vol ~ 55,000 l	m = 52,000 kg PE = 544 kJ Vol ~ 55,000 l
3. Average energy absorbed per strengthening element in drop test = (PE* 0.73)/N	PE*0.73 = 275 kJ N=7 Average energy=39.3 kJ	PE*0.73 = 275 kJ N=8 Average energy=34.4 kJ	PE*0.73 = 275 kJ N = 9 Average energy=31 kJ	PE*0.73 = 275 kJ N=10 Average energy = 28 kJ	PE*0.73 = 397 kJ N=10 Average energy=39.7kJ	PE*0.73 = 397 kJ N=13 Average energy = 30.5kJ

Technical code option 3 – drop test energy calculation Step and formula	**Typical UK fleet (7 strengthening elements, stuffed type joint design)**	**Typical UK fleet (8 strengthening elements, banded type joint design)**	**Typical UK fleet (9 strengthening elements, banded type joint design)**	**Typical UK fleet (10 strengthening elements, banded type joint design)** Shown to perform well in topple test	**Extra-large 1 (10 strengthening elements, banded type joint design)**	**Extra-large 2 (13 strengthening elements, banded type joint design)**
4. Maximum energy absorbed per element and two-compartment section vertical drop test energy						
Max energy per element = (average energy per element)*1.25^2 * 1.1	Max energy = 68 kJ	Max energy = 59 kJ	Max energy = 53 kJ	Max energy = 48 kJ	Max energy = 68 kJ	Max energy = 52 kJ
Test energy = (max energy per element) * 3	Test energy = 203 kJ	Test energy =177 kJ	Test energy = 160 kJ	Test energy = 144 kJ	Test energy = 204 kJ	Test energy = 158 kJ

3.3 Discussion

As mentioned in Section 3.1.2 above, and illustrated in Figure 3-6 and Figure 3-8, previous work found that although a stuffed type joint is generally less stiff than a banded type joint, it has greater energy absorption capability before failure. This was demonstrated using a forming limit diagram and the omega criterion[24].

Figure 3-10 shows a plot of omega against energy absorbed per strengthening element (partition) for tankers with banded type 1, banded type 2, and stuffed type joint designs. It should be noted that since its update in 2020, EN 13094 no longer permits tanks with a banded type 1 design [Annex D, D.14(a)] because previous work[25] found it to be particularly susceptible to failure and hence it was removed. Therefore, tankers with banded type 1 designs are not considered in the current work, but it is interesting to note the high omega values and hence high risk of failure associated with the design. It should be noted that the forming limit curve, to calculate omega values shown in Figure 3-10, was chosen in a conservative manner, so any values less than 1.0 indicate an acceptable design (low risk of failure), and indeed values of 1.0 and slightly above could possibly indicate an acceptable design. This means that the omega values for the banded type 2 design just below 1.0 are acceptable, but the even lower omega values for the stuffed design are more desirable because they offer an even larger safety margin. The red dashed line shows that the omega value for the banded type 2 design rises above 1.0 at an energy of about 60 kJ.

Figure 3-10 is marked with green dashed lines to show the maximum energies of 68 kJ and 52 kJ for the hypothetical extra-large tankers designs with 10 and 13 elements, respectively. It is seen that both these energies are below an omega value of 1.0 for a stuffed type joint design, indicating that they are acceptable. However, for a banded type 2 joint design, the omega value for an extra-large tanker with 10 elements (energy 68 kJ) is slightly above 1.0 indicating that it is on the borderline of acceptability.

It should also be noted that the 7 strengthening element conventional tanker that absorbs an equivalent maximum energy to the 10 element extra-large one, has a stuffed type joint design (see Table 3-3), and hence has a greater safety margin than a potential extra-large design with a banded type 2 joint design.

[24] The omega criterion is based on a Forming Limit Diagram (FLD) and indicates the risk of failure with values below 1 indicating that failure is not likely and values above 1 indicating that failure is likely. Omega is defined as the ratio of the major principal strain to the major limit strain at the given minor strain – see Figure 3-5 in Section 3.1.2 above.

[25] TWI Report No. 25272/1/16: Department for Transport Technical Assessment of Petroleum Tankers: Assessment of BS EN 13094 Lap and Partition Joint Designs (2016): https://assets.publishing.service.gov.uk/media/5fc65475d3bf7f7f575b4369/Technical-assessment-of-BS-EN-13094.pdf

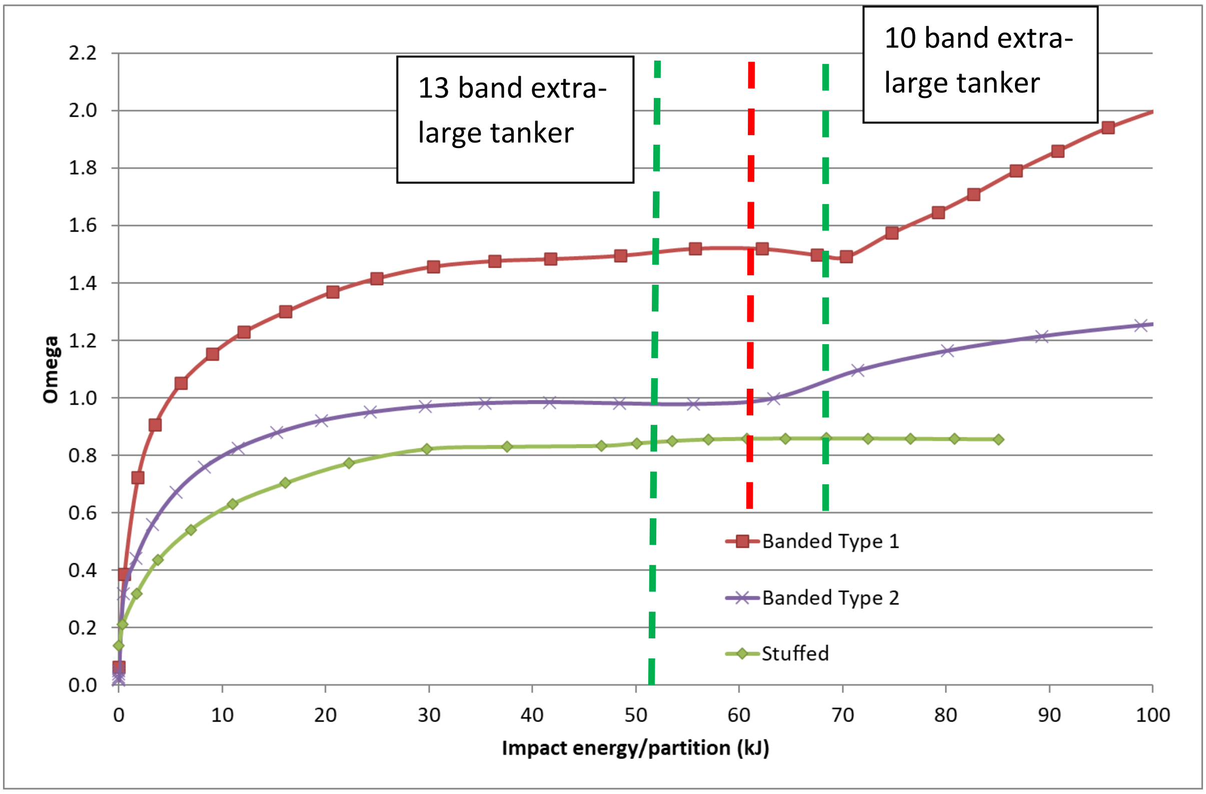


Figure 3-10: Plot of omega against energy absorbed per strengthening element (partition) for banded and stuffed joint designs with green dashed lines added to show maximum energy absorbed per element for 10 and 13 band extra-large tankers and a red dashed line to show where the omega value for banded type 2 design exceeds 1.0. Note that since its update in 2020, EN 13094 no longer permits tanks with a banded type 1 design because previous work found it to be particularly susceptible to failure

The previous work found that the impact energy in a topple is mainly absorbed by deformation of the dish part of strengthening elements (partitions / baffles). Extra-large tank vehicles will weigh more and thus have more impact energy to absorb in a topple than a conventional sized tanker. Therefore, they will require more strengthening elements (or equivalent, e.g. stronger strengthening elements) to offer an equivalent level of safety and absorb the impact energy without tank failure and fuel release.

The previous work also investigated the influence of the joint design type on the energy absorbed in a topple impact using a forming limit diagram and omega approach to indicate likelihood of failure and acceptability of designs. It found that a stuffed type joint design, although less stiff, should offer greater energy absorption capability before failure than a banded type joint design.

A comparison of the maximum energy absorbed per strengthening element values for two hypothetical potential extra-large tank vehicle designs weighing 52,000 kg with 10 and 13 strengthening elements, with those of current conventional tankers, found that the values for the extra-large vehicle designs lay within the range of those for current conventional tankers indicating that they should offer adequate levels of safety. However, it was noted that, whereas the design with 13 elements had a maximum energy absorbed value close to a current tanker with a banded joint design known to perform well in a topple, the design with

10 elements had a value comparable to a current tanker with a stuffed joint design which offers greater impact energy absorption capability than a banded joint design. Therefore, consideration of the influence of joint design on energy absorption is necessary to determine confidently that an extra-large tanker with a banded joint design offers adequate levels of safety.

Application of the forming limit diagram and omega from the previous work to the hypothetical extra-large tank designs showed that although the 13-element design should offer adequate energy absorption capability with either stuffed or banded type joints, the 10-element design may not offer adequate energy absorption with a banded type joint, i.e. the omega value is slightly above 1.0 indicating borderline acceptability.

It should be noted that whereas there is reasonable confidence in the forming limit diagram / omega approach and curves used overall, they are not precise. Therefore, they should not be used to predict the acceptability of borderline cases, i.e. the 10-element extra-large tank design. To give more confidence in the curves physical drop tests are needed to validate them. Also, to assure the range of their applicability further modelling work with a range of tanker cross-section shapes is needed.

In summary:

- Previous work showed that:
    - In typical simple rollovers, consisting of simple 90 degree topple and slide, without object impact, there is generally no spillage of flammable liquid, except from those that occur at high speeds.
    - Strengthening element (partition) to shell joint design has substantial effect on its energy absorption capability in a topple event, with stuffed design being less stiff, but having greater energy absorption before failure.
- The study performed found that:
    - The current EN 13094 standard permits the construction of a hypothetical 52,000 kg extra-large tank vehicle with 10 strengthening elements with either a stuffed or banded design.
    - Application of the forming limit diagram and omega from the previous work to this 10-element tanker design showed that whilst a stuffed design should offer sufficient energy absorption capability in a topple, a banded design may not, without incorporating additional strengthening elements or equivalent, e.g. stronger strengthening elements.

3.4 Conclusions and suggested next steps

The investigation into the safety implications of extra-large tank vehicles in rollover concluded that it may be necessary to consider the effect of joint design to assure that they have adequate energy absorption capability in a topple. However, confidence in the forming limit diagram / omega approach and curves used to derive this conclusion, although reasonable, was not high enough to support changes to the ADR requirements.

This conclusion was derived from the following findings:

- The current EN 13094 standard permits the construction of a potential extra-large tank vehicle which requires strengthening elements with a maximum energy absorption capability at the upper end of that seen for current conventional tankers, specifically one with a stuffed design.
- A banded design, which has a lower energy absorption potential before failure compared with a stuffed design, would also be permitted by the current EN 13094 standard for a potential extra-large tank vehicle. However, curves of omega against energy absorption potential per strengthening element for different joint designs, derived as part of the previous work, show that a banded design may not have sufficient energy absorption potential. However, it should be noted that although confidence in the derived curves was reasonable, it was not high because work to validate them was limited, for example no drop test work was performed.

Regarding next steps, should consideration be given to allowing the use of extra-large tank vehicles, it is suggested that tanker stakeholders are informed of the importance of consideration of the additional energy absorption required in topple impact for these vehicles due to their additional weight, and particularly the influence of joint design. Planned project dissemination activities should help achieve this objective.

It is also suggested that further work is performed to help validate the technical code developed in the previous project and thus provide more confidence in the forming limit diagram / omega approach and curves used to derive the conclusion. A first part of this work could be the verification of the difference in behaviour for tanks with banded and stuffed type joint designs predicted by the modelling and the forming limit diagram / omega approach. At an impact energy per partition of 85 kJ, the approach predicts omega values of 1.2 and 0.82 for tanks with banded and stuffed joints, respectively (Figure 3-10). This indicates a low risk of rupture for the stuffed joint tank and in contrast a high risk of rupture for the banded joint tank. This prediction could be verified by performing two sub-section drop tests with similar tank sections but with different joint designs and showing that the tank with the stuffed design maintains its integrity and in contrast the tank with the banded design ruptures. Further FE modelling and sub-section drop tests could be performed to verify other aspects of the technical code such as its robustness for different tanker cross-sections and the magnitude of the safety factor recommended.

4 Acknowledgements

The authors would very much like to thank the many individuals and organisations who have contributed to and assisted in this research.

In particular, we owe a debt of gratitude to the stakeholders from the UK fuel tanker industry, including manufacturers and Appointed Inspection Bodies, that have so diligently contributed their time, knowledge and experience to the research.

Finally, we must acknowledge the significant contribution made by our client, the Department for Transport's Dangerous Goods Unit and, especially, the tireless and invaluable support provided by Steve Gillingham, and, following his retirement, by his successor David Adams.

Appendix A Finite Element (FE) modelling – further information

A.1 Background information for collision analysis

The background information gathered consisted of:

- Crash loads required by various regulations for frontal impact collisions regulatory requirements
- Load Cell Wall force data for HGV tests

This information is shown below:

A.1.1 Regulatory crash loads

- UNECE R67 'LPG vehicles' - M3 and N3 category vehicles
    - Container attachment - withstand 6.6.g without damage occurring
- UNECE R80 'Strength of seats on buses' - M2 and M3 category vehicles
    - Dynamic tests of seats with crash pulse peak between 8g and 12g, average 6.5g to 8g
- UNECE R100 'Electrical safety' – mechanical shock of rechargeable energy storage system – M3 and N3 category vehicles
    - Dynamic test with crash pulse peak between 6.6g and 12g
- UNECE R110 'Specific components for CNG and/or LNG' - M3 and N3 category vehicles
    - Container attachments – withstand 6.6.g without damage occurring
- UNECE R134 'Hydrogen and fuel cell vehicles' - M3 and N3 category vehicles
    - Container attachments – withstand 6.6.g and tank remain attached at minimum of one point

The crash pulse corridors for UNECE Regulation No. 80 and Regulation No. 100 are shown below.

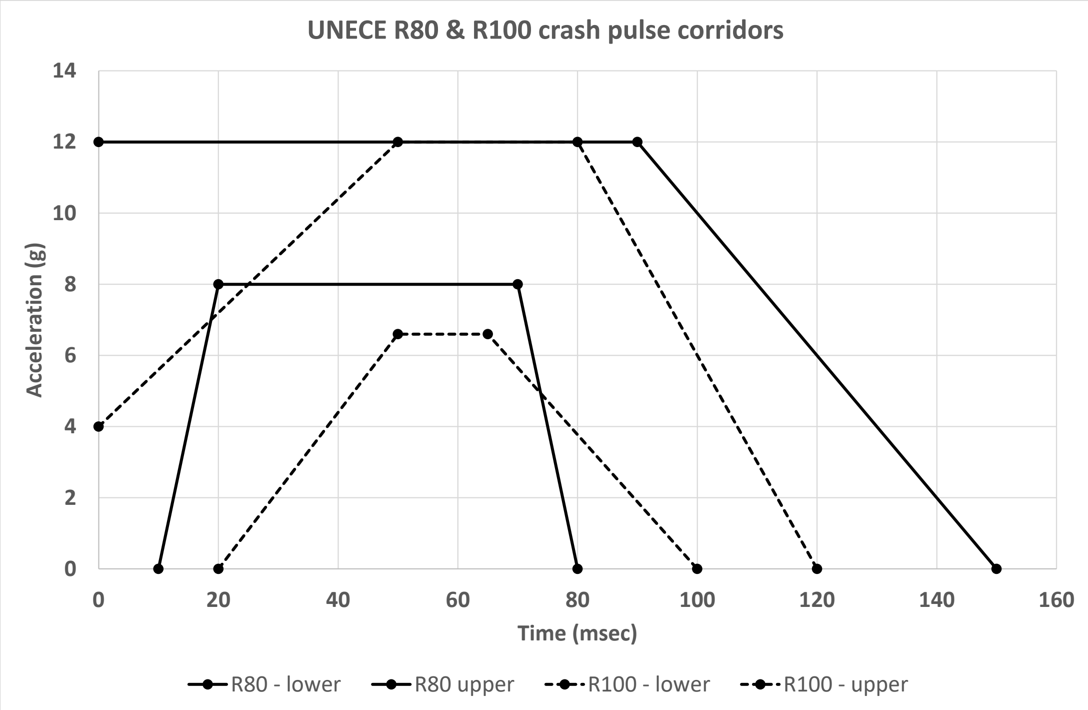
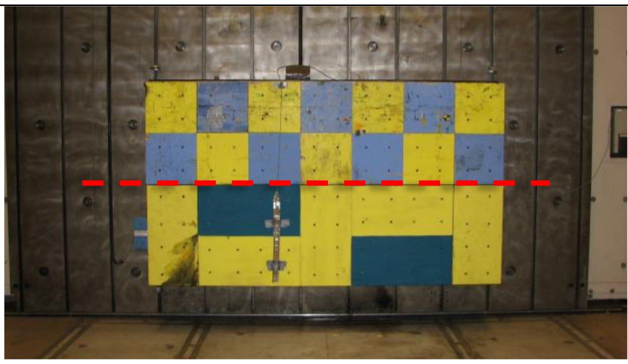


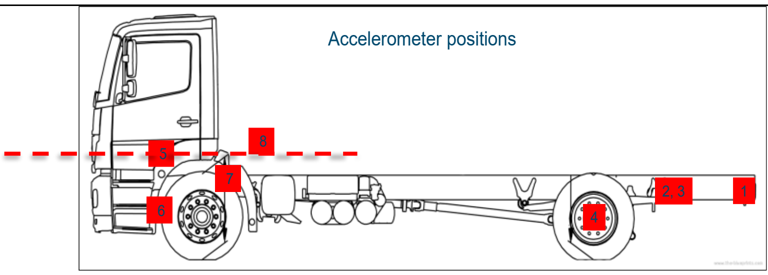
Figure A.1: UNECE Regulation No. 80 and Regulation No. 100 crash pulse corridors

A.1.2 LCW test force data

LCW force data from tests at 48 km/h, 64 km/h and 80 km/h, with a 18 tonne, 2 axle rigid body truck (test weight 7.5 tonnes) are shown below.



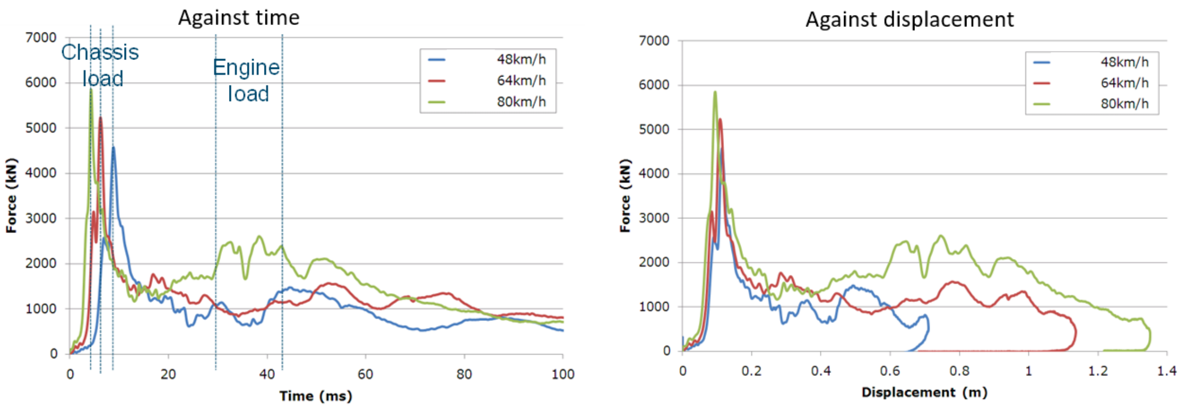
Rigid LCW



Instrumentation:

- Rate sensor (1) rear of vehicle
- Accelerometers: (2), (3) chassis rails, (40 rear axle, (5), (6), (7) engine and gearbox, (8) load

Figure A.2: LCW test setup and vehicle instrumentation



- Peak load result of chassis rail load which increases slightly with speed
- Chassis rail load timing changes due to change in speed

- Plot against displacement of rear of vehicle shows that chassis rail loading occurs at same vehicle displacement independent of speed

Figure A.3: LCW measured force (total) against time (left) and displacement (right)

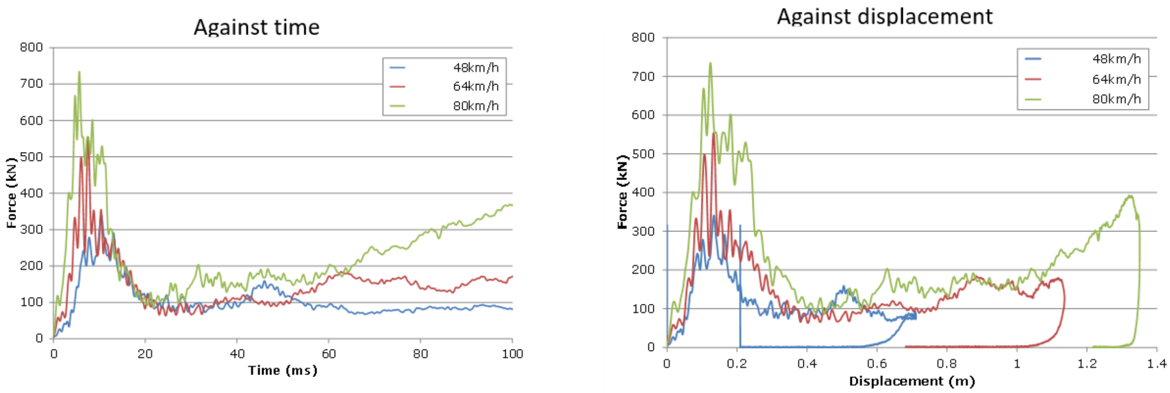


Figure A.4: LCW measured force (upper wall in alignment with cab) against time (left) and displacement (right); note cab crush loads estimated ~ 100 kN

Appendix B Additional data from LS-Dyna3D crash simulations

B.1 Parameter sweep 1 – lower initial velocities

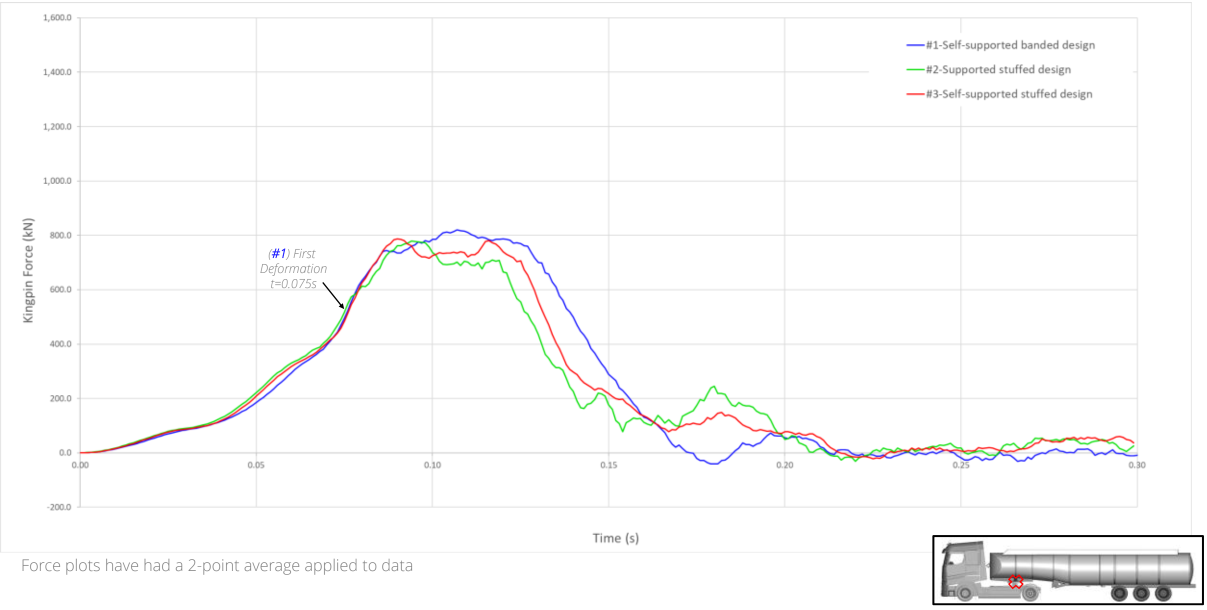


Figure B.1: Comparison of semi-trailer king pin force in longitudinal (x) direction for striking tanker at 12 km/h

Figure B.2: Comparison of semi-trailer king pin force in longitudinal (x) direction for striking tanker at 18 km/h

Figure B.3: Comparison of semi-trailer king pin force in longitudinal (x) direction for striking tanker at 24 km/h

B.2 Parameter sweep 2 - fixed king pin deceleration

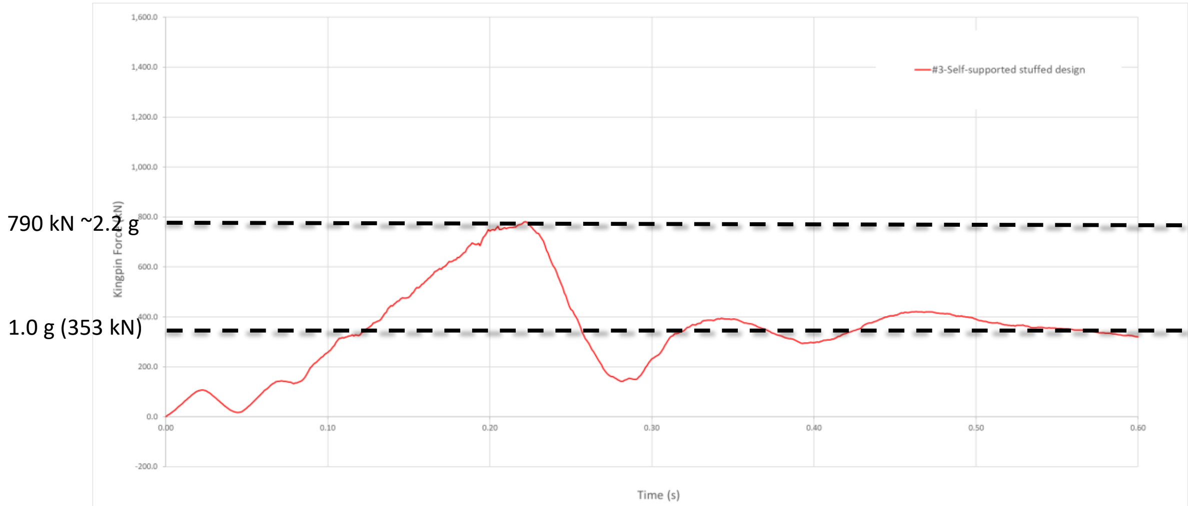


Figure B.4: King pin longitudinal forces for 1.0 g fixed king pin deceleration boundary condition parameter sweep – semi-trailer #3 self-supported stuffed design

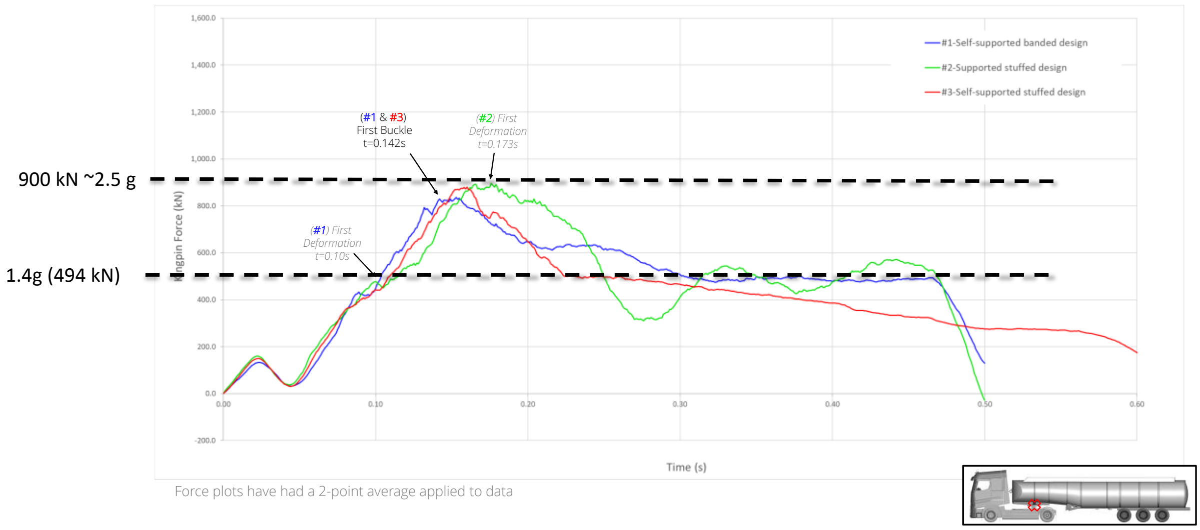


Figure B.5: Comparison of king pin longitudinal forces for 1.4 g fixed king pin deceleration boundary condition parameter sweep

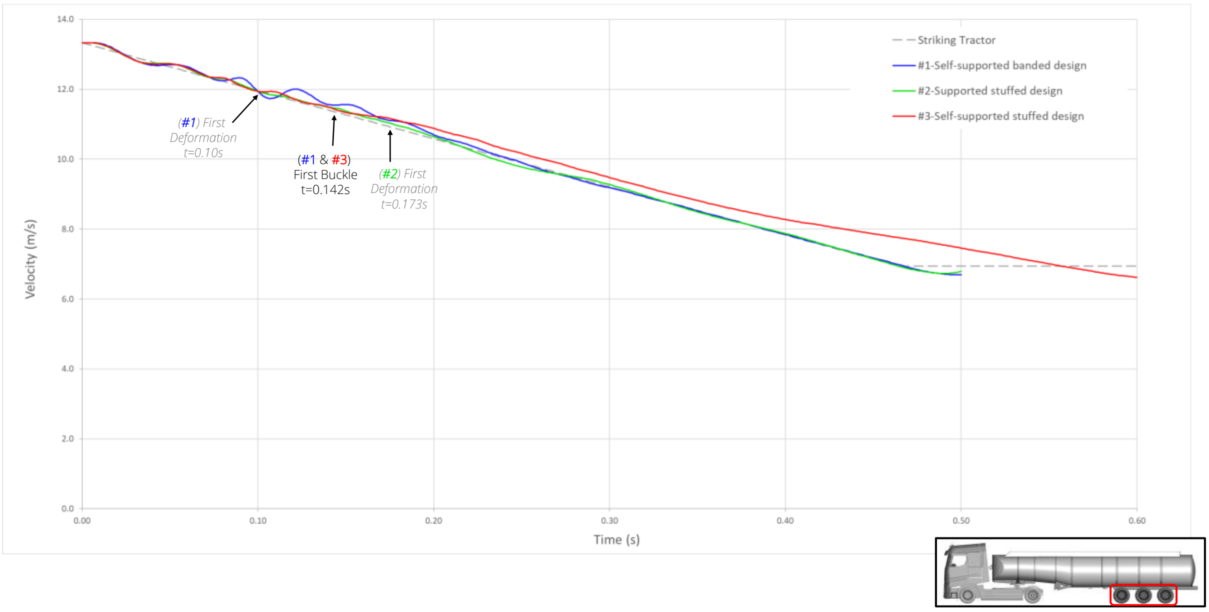


Figure B.6: Comparison of semi-trailer rear (axle) velocities for 1.4 g fixed king pin deceleration parameter sweep. Note that dashed grey line represents the deceleration boundary condition constraint applied to the king pin

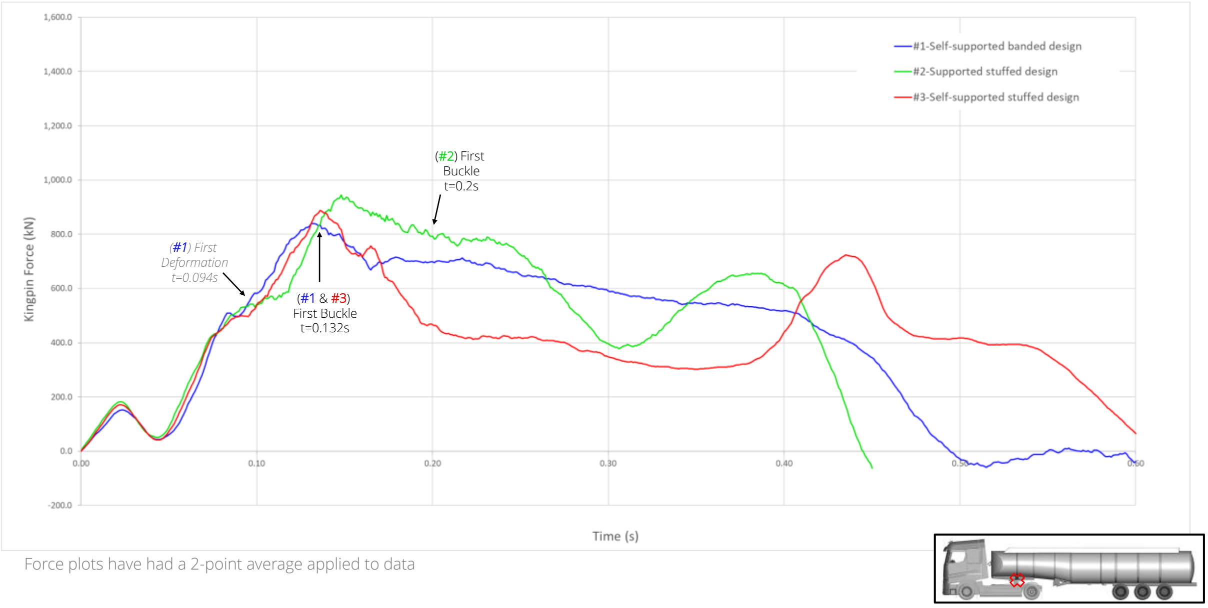


Figure B.7: Comparison of king pin longitudinal forces for 1.6 g fixed king pin deceleration boundary condition parameter sweep

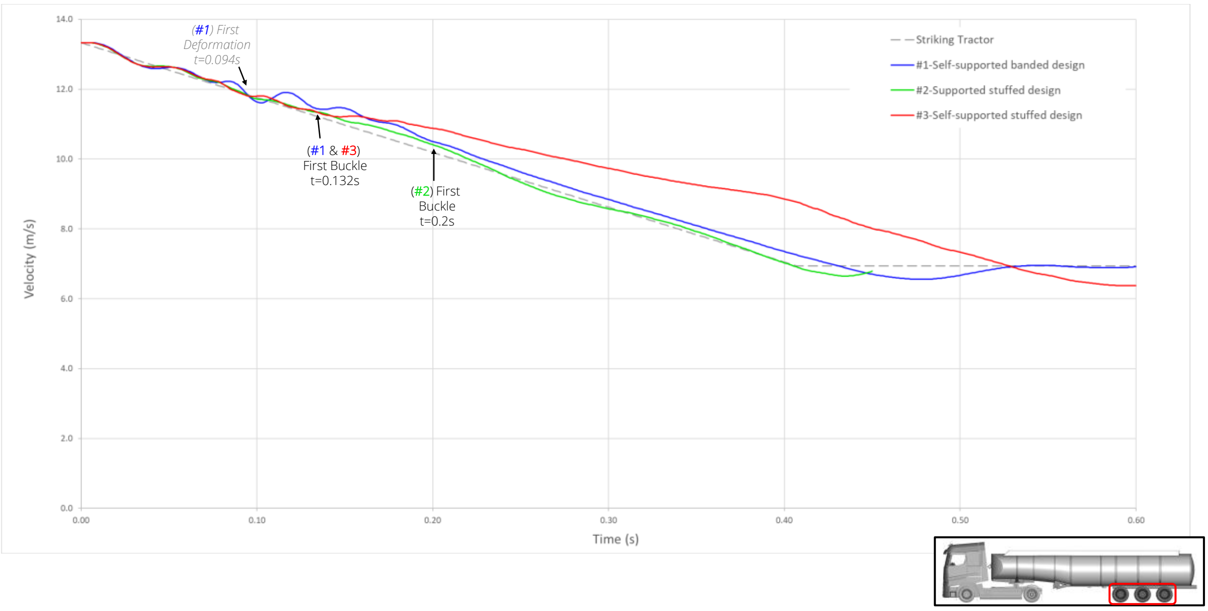


Figure B.8: Comparison of semi-trailer rear (axle) velocities for 1.6 g fixed king pin deceleration parameter sweep. Note that dashed grey line represents the deceleration boundary condition constraint applied to the king pin

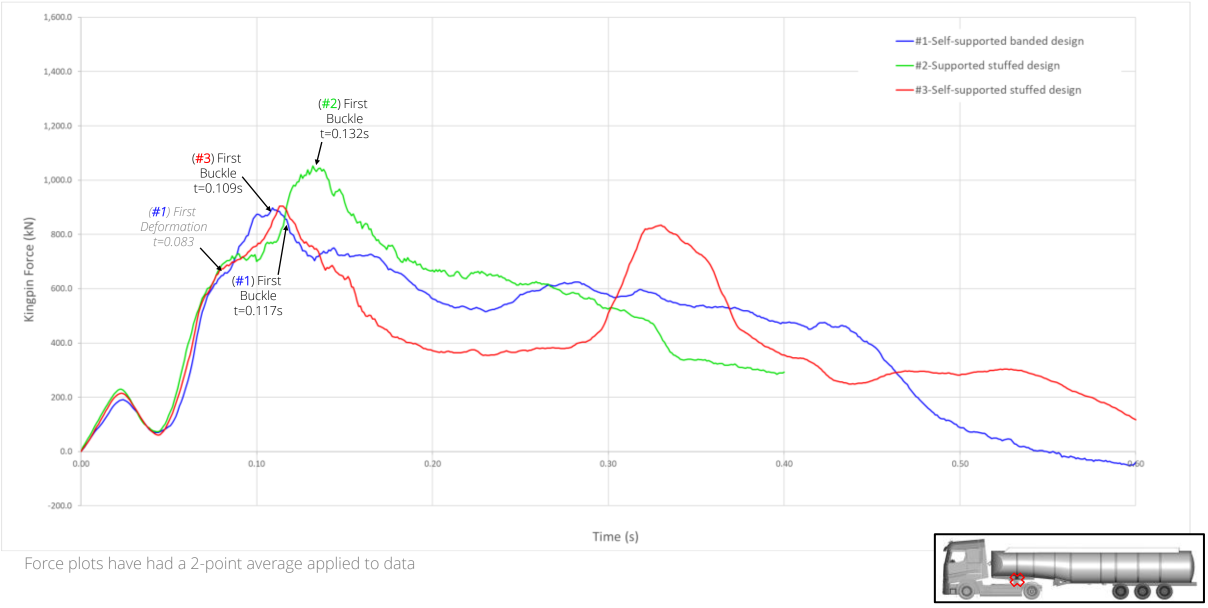


Figure B.9: Comparison of king pin longitudinal forces for 2.0 g fixed king pin deceleration boundary condition parameter sweep

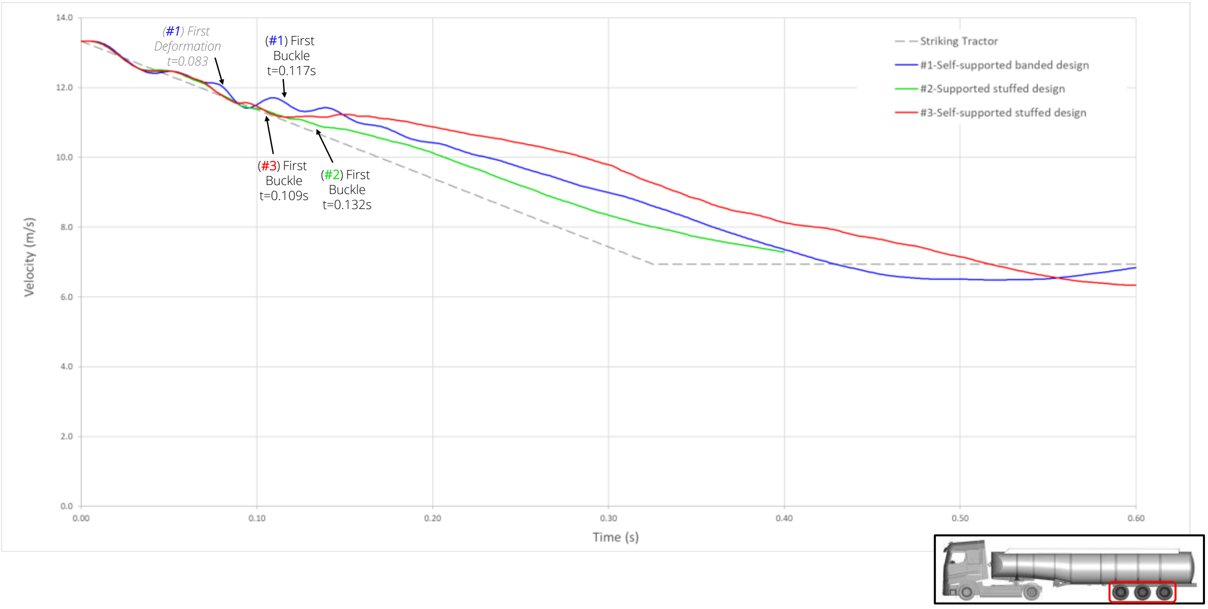


Figure B.10: Comparison of semi-trailer rear (axle) velocities for 2.0 g fixed king pin deceleration parameter sweep. Note that dashed grey line represents the deceleration boundary condition constraint applied to the king pin

Appendix C Investigation of relationship between global resilience and the energy absorbed in a topple impact

C.1 Introduction

The EN 13094 standard defines:

Global resilience as: *the ability of a shell with reinforcement(s) to withstand a sideways impact with a beam*

The test method is illustrated below (Figure C.1).

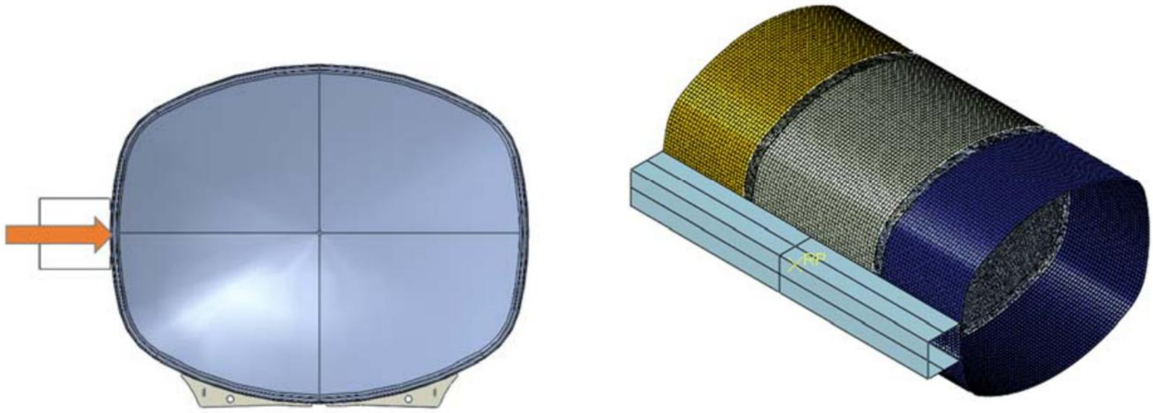


Figure C.1: Illustration of global resilience test. 4 m long, 430 mm high beam pushed into weakest 4 m side of tank until displaced 250 mm

Specific resilience as: *the integral of the applied force and the measured deflection of a test piece up to the point at which the test bar punctures the test piece, as indicated by the point of maximum force*

The test method uses a sample of tank shell material of size 500 mm by 500 mm. The sample is clamped and bolted into a test machine with a clamping ring with a nominal diameter of 445 mm and which contains twenty 13 mm diameter bolts which clamp through the test sample. The test involves the machine pressing a bar with a nominal diameter of 150 mm and 6 mm radiused edges into the test sample at a steady speed between 2 and 4 mm/s and measuring the force and displacement.

For tanker shells which do not have a circular or elliptical cross-section EN 13094 requires that:

- Global resilience (see paragraph 6.8.2.2 (i) and Annex B.6): The weakest 4 m shell segment has a global resilience (i.e. energy absorption capacity) of at least 100 kNm for a beam displacement of 250 mm without shell rupture.
- Global resilience: (see paragraph 6.8.2.2 (j) and Annex B.7): The energy absorbed during overturning is at least equal to that of a shell with a circular or elliptical cross-

section with the similar defined parameters. The energy absorbed shall be evaluated according to the global resilience method.

- Specific resilience: (see paragraph 6.8.2.2 (c) and Annex B): The shell shall have a specific resilience (i.e. resistance to penetration), as determined in accordance with Annex B (i.e. a specific resilience test), at least equal to that of a shell constructed in reference steel (mild steel):
    - Of a thickness of 5 mm for tank shells with a diameter not exceeding 1.8 m and
    - Of a thickness of 6 mm for tank shells with a diameter exceeding 1.8 m
- Specific resilience (see paragraph 6.8.2.2 (j) 4 and Annex B.7): The energy absorbed during an impact on lateral side and end at least equal to that of a shell with a circular or elliptical cross-section with the similar defined parameters. The energy absorbed shall be evaluated in accordance with Annex B.7, i.e. a specific resilience test.

On the basis that the EN 13094 standard uses 'global resilience' to compare the energy absorbed by two tanks in overturning (i.e. a topple impact), it was decided to perform FE modelling work to understand better the relationship (energy absorption) between global resilience and topple impact. For example, how does the energy absorbed in a global resilience test compare with that absorbed in a topple?

The work performed compared the energy absorbed in a global resilience test and a topple impact, for one tanker with a non-elliptical cross-section design with two different types of strengthening element. No work was performed for tankers with standard circular or elliptical cross-section designs because of budget constraints. Therefore, further work would be needed before it would be possible to answer questions such as 'is the energy relationship between global resilience and topple impact dependent on the tanker cross-section?'

The model build, simulation results and conclusions are described in the following sections.

C.2 Build of FE Model

<u>Geometry</u>

The geometry was based on that of a box shaped tanker with a non-elliptical cross-section shape with an extruded banded type joint design. A half symmetry model of a 4 m long section of the tanker with two strengthening elements (partitions), appropriate for a global resilience test for approval, i.e. weakest 4 m section, was built (Figure C.2).

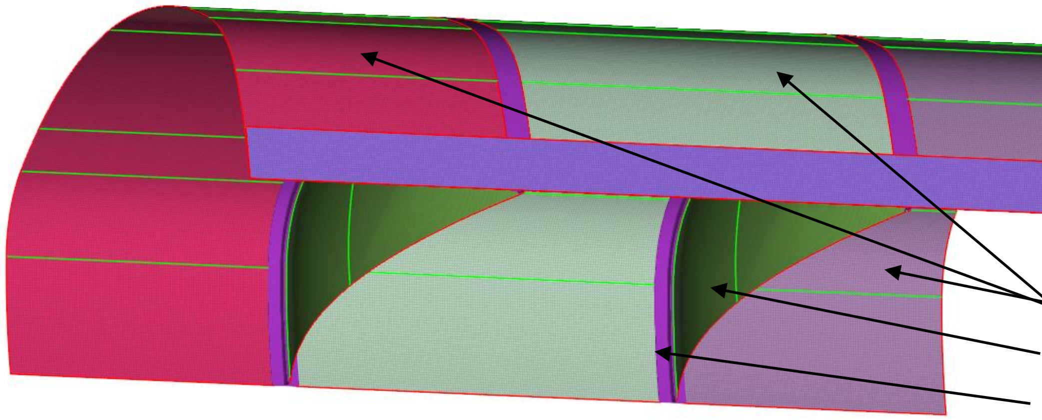


Figure C.2: FE model – 4 m long section of tanker with two partitions with half-symmetry, so top half of tank only shown

The tanker shell and partition dish elements were meshed using first order shell elements and the extruded banded sections using a mixture of first order solid, hexahedral and pentahedral elements

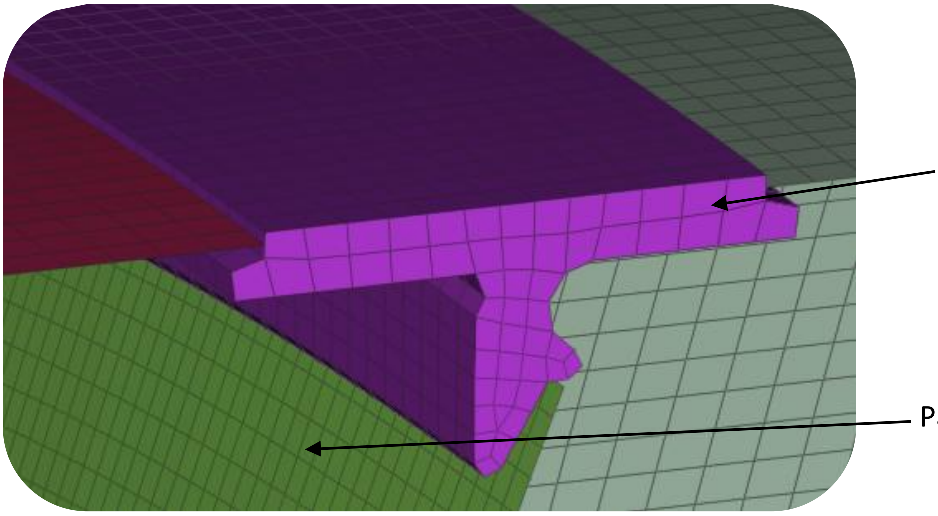


Figure C.3: Model close up showing tanker shell and dish meshed with solid elements and extruded band section meshed with solid elements

<u>Materials</u>

For materials a bilinear representation of the stress-strain curve was used with two sets of material properties for the different parts as shown in Table C.1.

Table C.1: Material properties

Part	Young's Modulus (GPa)	Yield (MPa)	UTS (MPa)	Elongation Limit	Poisson's Ratio
Tanker shell and dish	70	125	275	26%	0.3
Banded section	70	290	385	11%	0.3

Boundary conditions

Symmetry

For the half symmetry nodes as shown in Figure C.4 were constrained as follows:

- Nodes in turquoise were fixed in Y translation and X, Z rotation.
- Nodes in magenta were fixed in X translation and Y, Z rotation.

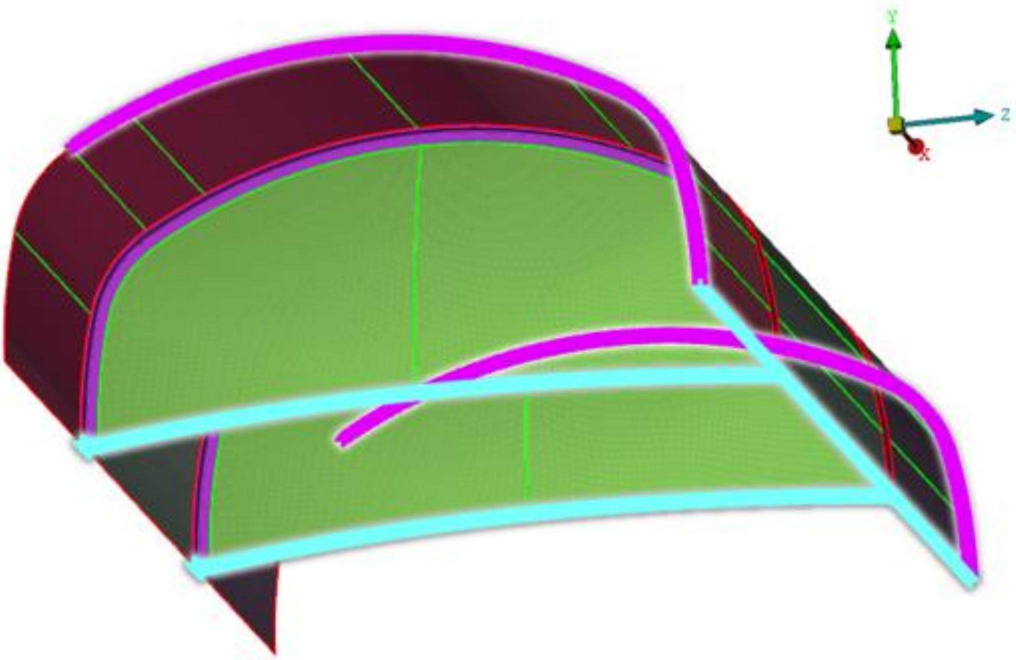


Figure C.4: Nodes constrained for half symmetry condition

Load application

Two 4 m long rigid ‘impactor’ planes were used to apply the load for the global resilience and topple simulations, respectively (Figure C.5). For the global resilience the impactor plane was 215 mm high to replicate the 430 mm high beam used for the global resilience test – note symmetry constraint 430 mm / 2 = 215 mm.   Both rigid planes were displaced 250mm into the tanker along the Z axis

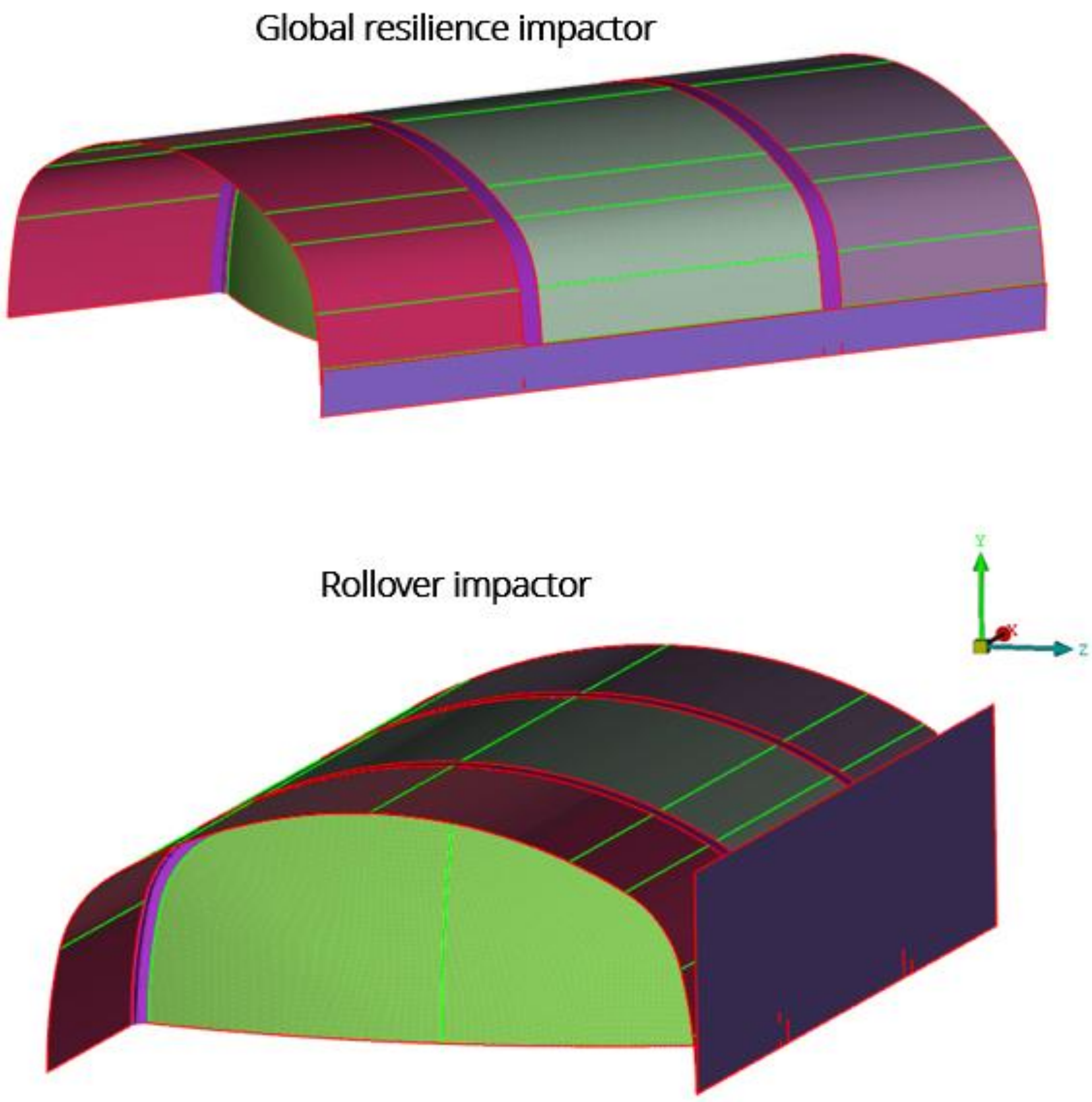


Figure C.5: 4 m long rigid planes with heights of 215 mm and 430 mm to apply loads for global resilience and topple simulations, respectively

Boundary conditions

The nodes shown in red area in Figure C.6 were constrained in translation and rotation to fix the model. The position and size of this area were chosen such that it gave a balance between constraining the model adequately and allowing the tank to deform in a realistic way when the rigid impactor plane loads were applied, i.e. deformation could occur in an unconstrained manner in the half of the tank where the loads were applied but did not occur around where the nodes were constrained.

Note that this area is different to that usually constrained for a global resilience test or simulation where the bottom and sometimes the top of the tank are constrained. However, this constraint would not allow the tank to deform in a realistic manner, so it was not used.

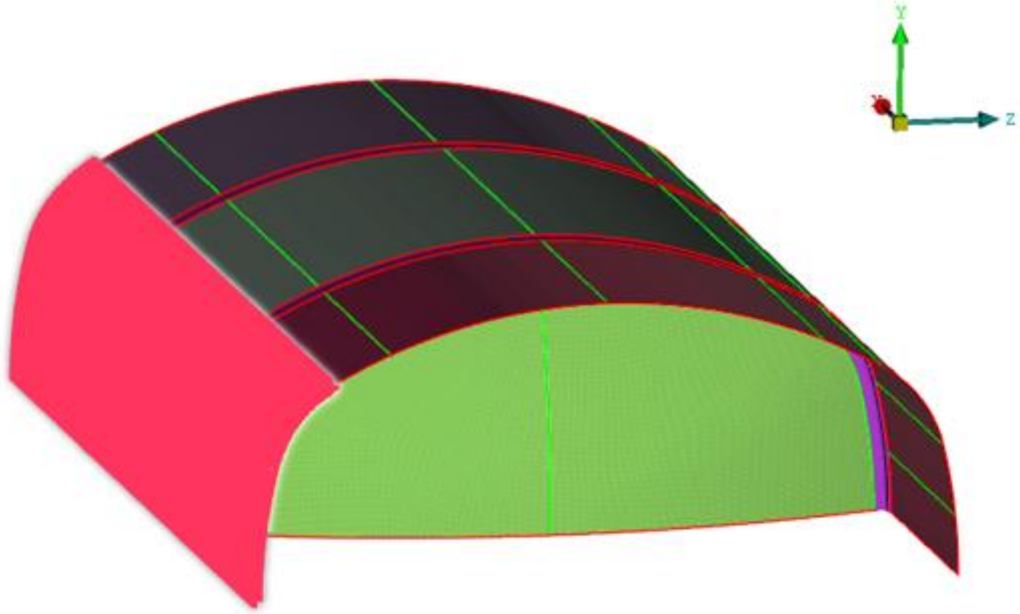


Figure C.6: Nodes in red area constrained in translation and rotation to fix model

The Abaqus explicit type solver was used to perform the simulations. This was because the model would not run consistently with the implicit solver, which generally takes less computer time to run, because of solution convergent type issues.

C.3 Results and discussion

Simulations performed

Four simulations were performed for the two impact types representing the global resilience (beam) and a topple (plane) and two designs, TD207 with a 5.28 mm thick dish and TD217 with a 7.5 mm thick dish, as shown in the matrix below

Table C.2: Simulation matrix

Plane impactor	Tank design	
	TD 207 (5.28 mm thick dish)	TD 217 (7.5 mm thick dish)
Global resilience (beam)	✓	✓
Topple (plane)	✓	✓

Also, to investigate the effect of the change of the boundary constraint from the tank bottom / top, usually used for global resilience tests, to the side of the tank to allow the topple simulation, one additional simulation with the TD 217 tank design and beam impactor (global resilience) was performed with the boundary constraint area on the tank top and bottom as shown (Figure C.7).

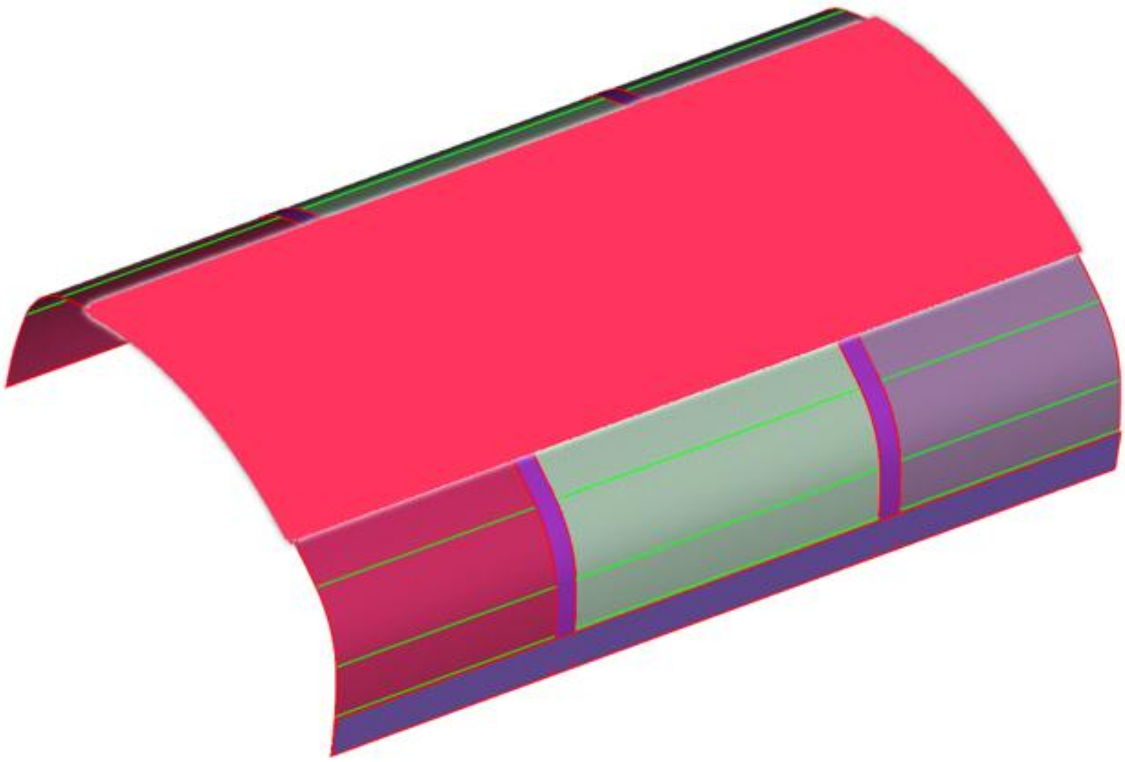


Figure C.7: Nodes in red area constrained in translation and rotation to fix model

Results and Discussion

The difference in the deformation shape of the tank for the global resilience (beam) and topple (plane) impactors are shown in Figure C.8. As expected, it is seen that whereas the global resilience beam just pushes in the middle of the tank which deforms around this area, the topple plane pushes over a much larger area to give a much more even deformation.

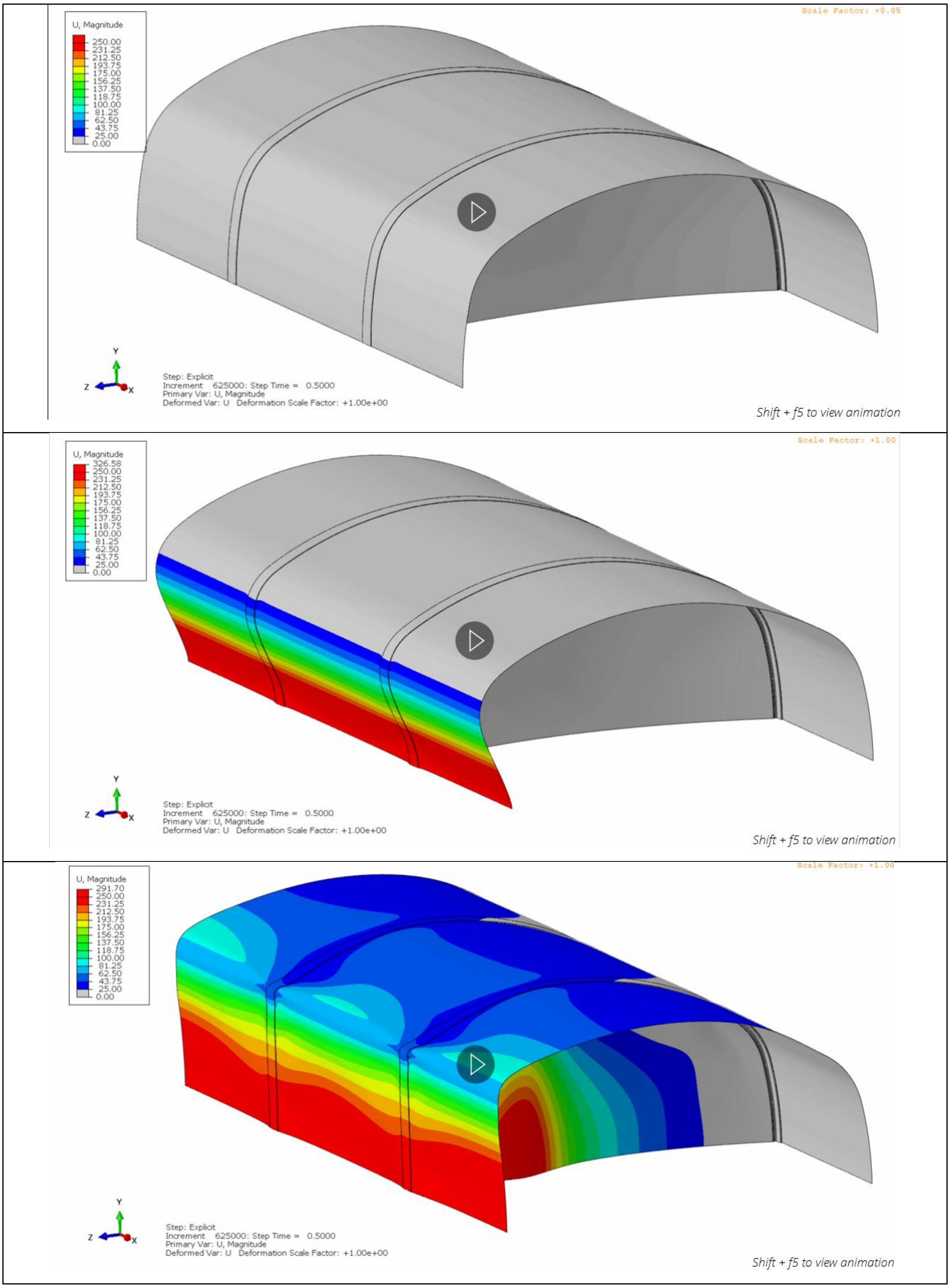


Figure C.8: Difference in deformation of tank (TD207 design) for the different impactors; top undeformed, middle global resilience (beam) impactor, bottom topple (planar) impactor

Figure C.9 shows a work displacement plot for the four main simulations.

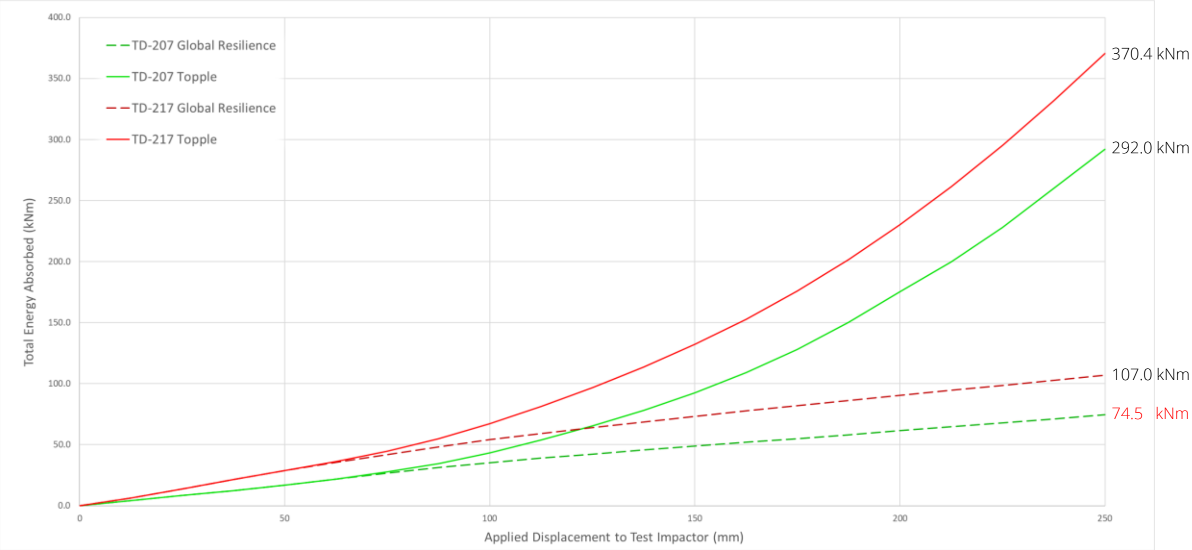


Figure C.9: Work displacement plot for four main simulations

It is seen that for a comparable displacement, up to a displacement of about 50 mm a similar amount of energy is absorbed in both the topple (planar) and global resilience simulations, but above this displacement a larger amount of energy is absorbed in the topple simulations, with 3.46 and 3.92 times the amount of energy absorbed at a displacement of 250 mm for the TD217 and TD207 tank designs, respectively. The reason for this behaviour is that up to a displacement of 50 mm the area of contact for the two impactors with the tank and hence deformation of the tank is the same, whereas above this displacement the area of contact with the tank for the topple impactor is greater which causes greater deformation and hence energy absorption. At all displacements the TD217 design absorbs more energy than the TD207 design because the strengthening elements have a thicker dish, which makes them stiffer, and they thus absorb more energy.

Figure C.10 shows a force displacement plot for the four main simulations.

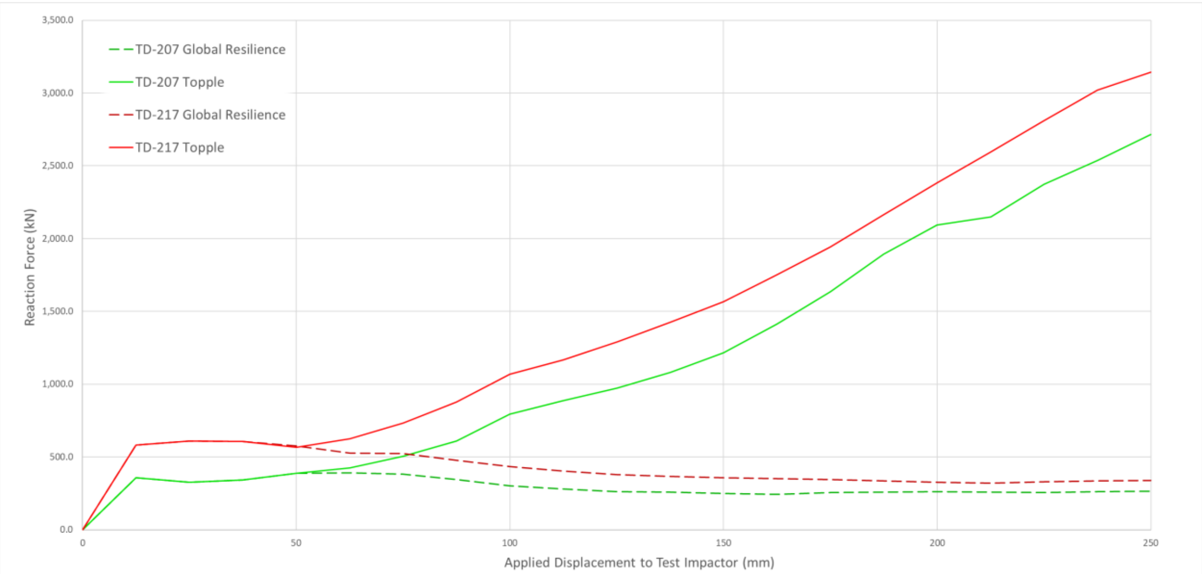


Figure C.10: Force displacement plot for four main simulations

As expected, it is seen that up to a displacement of about 50 mm, the forces are the same for the topple (planar) and global resilience (beam) simulations. However, at greater displacements the forces continue to increase for the topple (planar) simulations whereas they decrease slightly for the global resilience (beam) simulations. This is because for the topple simulations the plane continues to engage more of the side of the tank thus increasing its resistance whereas for the global resilience simulations no more of the tank side is engaged and plastic hinges are formed in the tank shell and extruded band as the beam pushes into it.

Figure C.11 shows a maximum omega - displacement plot for the four main simulations.

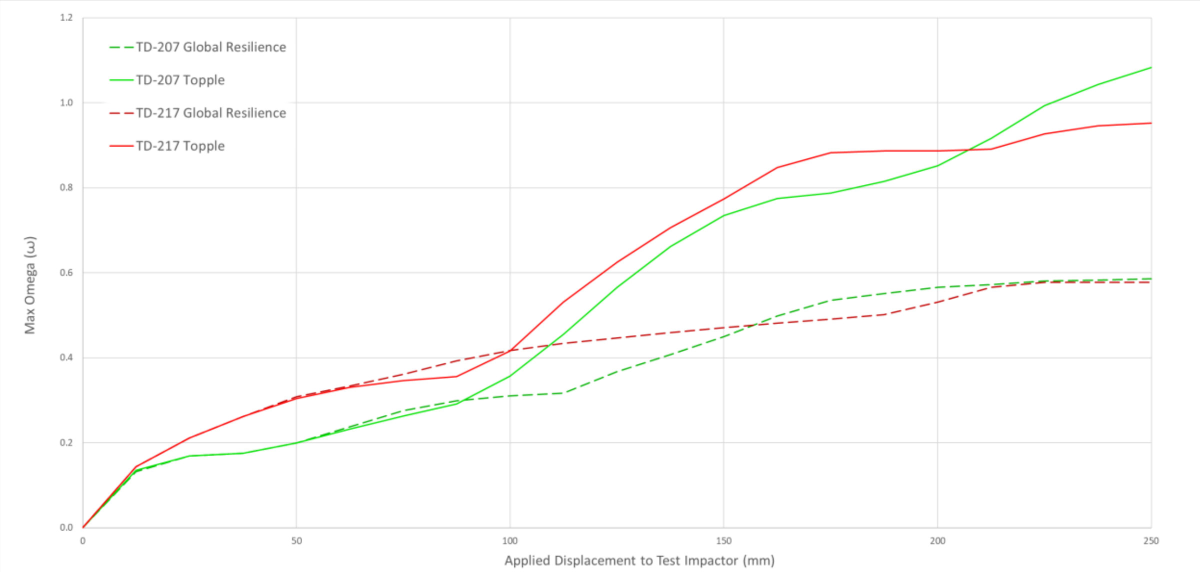


Figure C.11: Maximum omega - displacement plot for four main simulations

It is seen that at above about 100 mm of displacement the maximum omega values increase much more for the topple simulations than the global resilience ones. This is because for the topple simulation deformations are more localised and hence larger thus resulting in higher maximum omega values (see Figure C.12). However, as mentioned above it should be noted that much more energy is absorbed for a comparable displacement in the topple simulations.

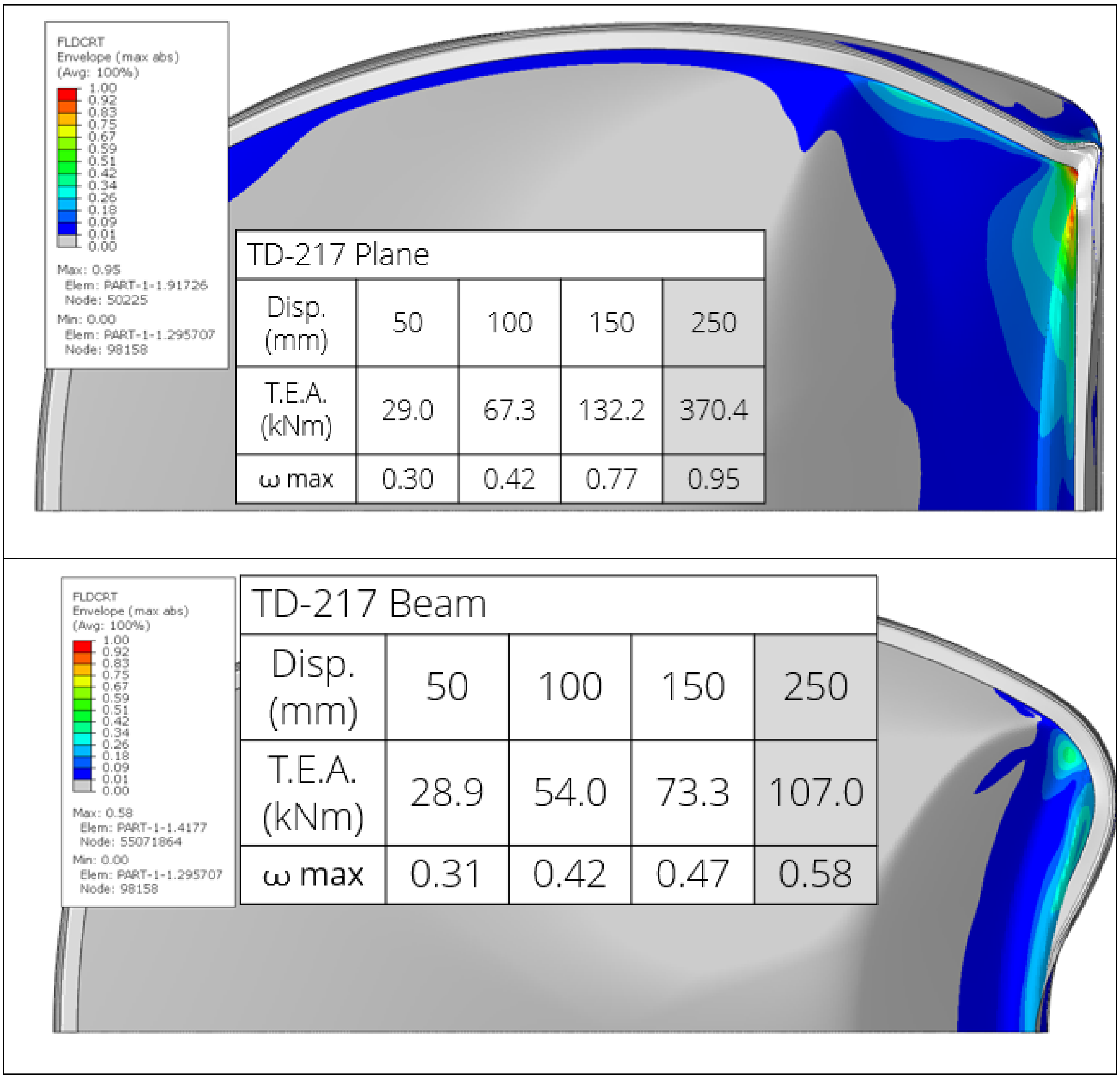


Figure C.12: End view of tank showing comparison of deformation and omega values at 250 mm displacement for topple (plane) (see top) and global resilience (beam) (see bottom). Note T.E.A. is Total Energy Absorbed

To help understand how the maximum omega (i.e. risk of tank rupture) varies by energy absorbed between topple impacts and the global resilience test, Figure C.13 shows a plot of total energy absorbed against maximum omega. It is seen that for comparable quantities of energy absorbed, the maximum omega values are approximately similar for topple impacts and the global resilience test, over the range for which data are available.

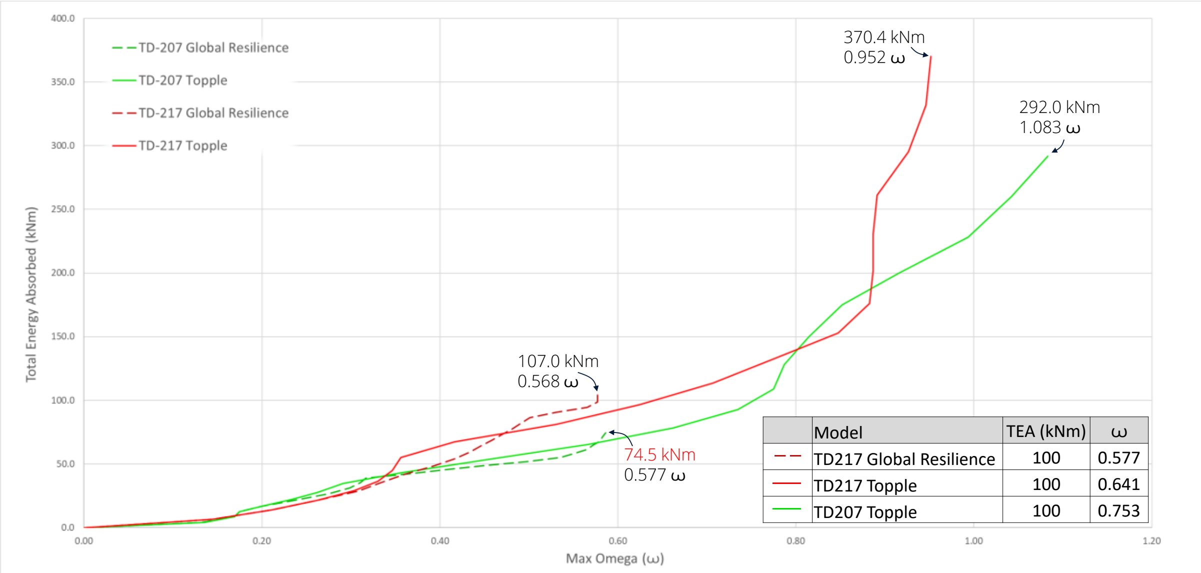


Figure C.13: Plot of total energy absorbed against maximum omega

To absorb an equivalent amount of energy the model with the less stiff partition (5.28 mm thick dish) has to deform more than the model with the stiffer partition (7.5 mm thick dish). This results slightly higher omega values as illustrated in Figure C.14.

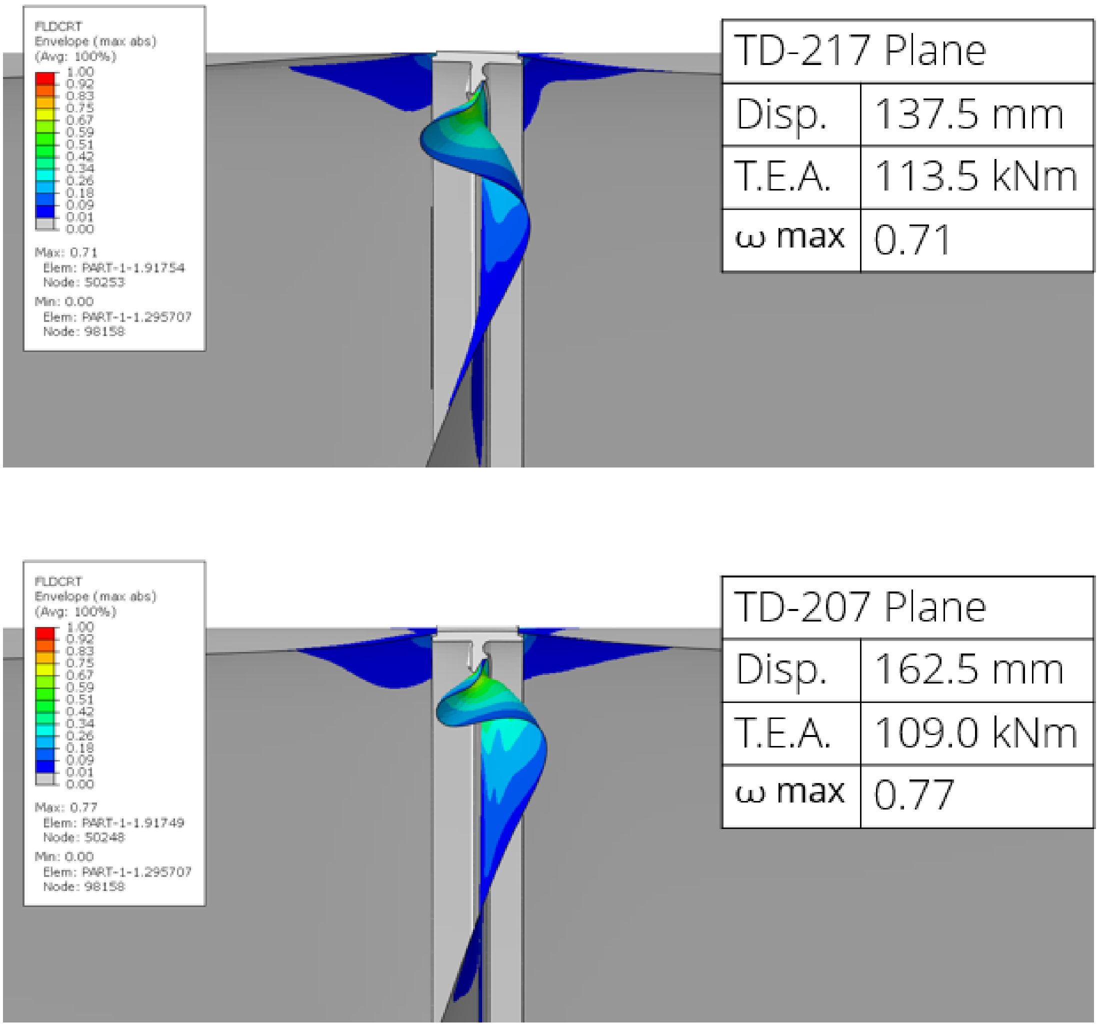


Figure C.14: View of model symmetry plane showing deformation of dish for topple impacts for 7.5 mm thick dish (TD217 top) and 5.28 mm thick dish (TD207 bottom) at point when just over 100 kNm total energy absorbed (T.E.A.)

To investigate the effect of the boundary condition on the global resilience test results an additional simulation was performed with the boundary constraint area on the tank top and bottom as described above and shown in Figure C.7.

Figure C.15 shows that, for a comparable displacement, more energy was absorbed for the boundary constraint at the bottom and top of the tank (BC No.2) than for the boundary constraint at the side of the tank.

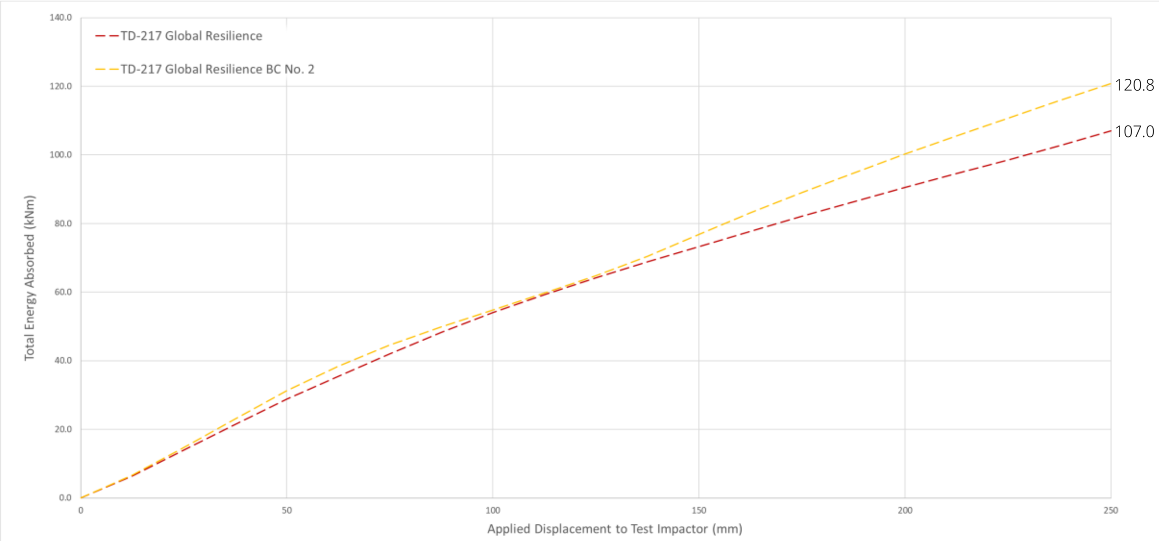


Figure C.15: Energy absorbed against displacement plot for global resilience simulations and the TD217 tank design (7.5 mm thick dish) with different boundary conditions

This was because the constraint at the bottom and top of the tank prevented deformation of the tank closer to the displacing beam than the boundary constraint on the far side, which caused larger local deformations and greater energy absorption.

This larger local deformation was reflected in higher omega values for the boundary constraint at the bottom and top of the tank (BC No.2) as shown in Figure C.16.

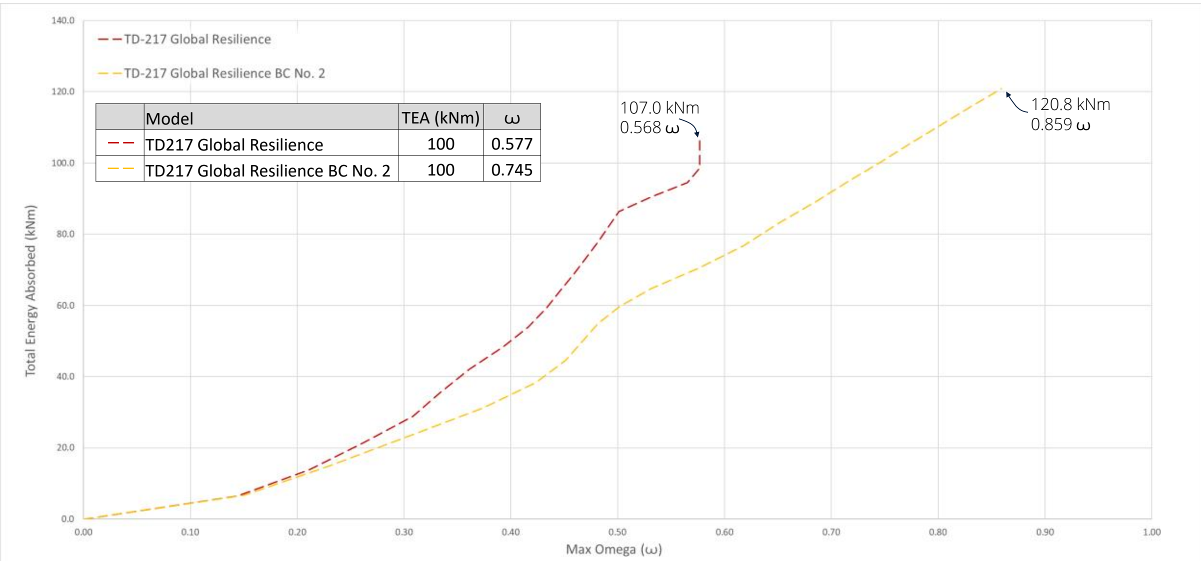


Figure C.16: Energy absorbed against maximum omega plot for global resilience simulations and the TD217 tank design (7.5 mm thick dish) with different boundary conditions

C.4 Conclusions

- For comparable displacements, the energy absorbed in a topple impact was greater or equal to that absorbed in a global resilience test.
    - Up to a displacement of ~ 50 mm, energy absorbed was similar because engagement of tank with beam and plane was similar and hence deformation of tank was similar.
    - Above a displacement of ~ 50 mm, energy absorbed in topple was greater because engagement of tank with plane was greater which leads to greater deformation of tank and hence energy absorbed, a factor of 3.92 at 250 mm displacement.
- For comparable energies absorbed, the risk of tank rupture, as indicated by omega, was approximately similar for a topple impact and a global resilience test for energies for which data available, i.e. up to circa 100 kNm.
    - For energies absorbed of up to 100 kNm, omega values were substantially below 1, the highest being 0.75, the lowest 0.58.
- The partitions with the thinner dishes (TD207), which are less stiff, must deform further compared to those with the thicker dishes (TD217) to absorb an equivalent amount of energy. This results in higher omega values indicating a higher risk of rupture.
- The boundary conditions used to constrain the model can affect the energy absorbed predicted by a simulation. Conditions which constrain the model bottom and top, usually used for global resilience simulations resulted in greater predicted energy absorption values (by about 10 %) than if model constrained on side.

Further assessment of petroleum road fuel tankers in frontal impact and rollover

The research reported built on previous work which investigated the performance of petroleum road fuel tankers in rollovers and frontal impacts. It focused on frontal impact collisions, mainly answering a question arising from previous work about a discrepancy between the results of the collision and FE modelling analyses, namely why not many tank failures were found in the collision analysis whereas the modelling analysis predicted they should occur often. It also considered the appropriateness of the current regulations for potential extra-large tank vehicles, both in terms of frontal impact collisions and rollover.

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