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Research on Performance Test Procedures for Petroleum Road Fuel Tankers

Summary report

Mervyn Edwards (TRL), Brian Robinson (BJR
Solutions), Mike Stewart (HSE-SD), James Hobbs
(HSE-SD) and Tyler London (TWI)

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Executive summary

Introduction and research context

Several years ago, when a number of petroleum tankers operating in the UK were found not to be compliant with existing ADR regulations, the Department for Transport (DfT) sponsored research to understand the risks¹. This involved a detailed review of collision data related to articulated tank vehicles, the development of a topple test procedure and associated finite element modelling assessments to explore the potential effects of the defects. This ultimately resulted in the progressive removal of the relevant vehicles from service. This, and other work carried out since, has demonstrated that the finite element modelling techniques developed in the research could quantitatively assess the mechanical performance and susceptibility to failure of, in particular, structural circumferential joints of tanker trailers in rollover incidents.

Building on this previous research, the aim of the research reported here is to develop 'performance-based' finite element modelling approaches and appropriate physical test procedures to approve tankers with novel designs that otherwise would not satisfy the current 'design-based' ADR approval 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*.

It is anticipated the new test procedures would enable manufacturers to develop tank shells using design and construction methods not necessarily depicted in the existing standards, but which are nevertheless able to sustain rollover, frontal, rear and side impacts without having to use a series of more costly full-scale tests to demonstrate the suitability of their tank shells. In so doing, the new test procedures will reduce barriers to new tanker designs and construction technologies, and further improve the regulations and standards.

The potential is that any new test procedures could be included in a new or revised reference standard in ADR, or in a technical code that may be recognised by a competent authority of a Contracting Party to ADR alongside the following ADR related standards for petroleum road fuel tankers:

- BS EN13094 Tanks for the transport of dangerous goods – Metallic gravity-discharge tanks – Design and construction. (BSI, 2020)
- BS EN12972 Tanks for transport of dangerous goods – Testing, inspection and marking of metallic tanks. (BSI, 2018)

This report relates specifically to the methodologies and findings of the part of this new research referred to as work package "Part B", which was to conduct impact scenario modelling and develop appropriate physical testing parameters. It was led by TRL Ltd in conjunction with HSE's Science Division (HSE SD).

¹ 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.

The main outputs of this research are 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. A secondary output is the development of a better understanding of a frontal impact (tank rupture) scenario and the associated loading of the tank structure through the kingpin assembly / support structure.

The first step of this work was to understand the type and nature of the accidents experienced by petroleum road fuel tankers in the real-world and the associated risk of fuel spillage. This work (carried out by Apollo Vehicle Safety in “Part A”) also identified candidate test procedures that, potentially, could be adapted into performance-based methods.

It was found that the safety performance of the flammable liquid (FL) vehicle fleet in Great Britain is broadly comparable to that of the general population of heavy goods vehicles. Almost all significant releases of flammable liquids arise from traffic collision incidents involving rollover and/or collision with another heavy vehicle. These represent only a small proportion of all traffic collision incidents involving FL vehicles. Within this small group of high-risk collisions, rollover is by far the most likely to result in significant releases.

Where rollover does occur, simple on-carriageway rollover was found to be the most common type. Although not quantifiable in national data, the literature strongly suggested that this will typically involve a 90-degree roll onto the side of the vehicle. There was evidence to suggest that tanks typically survived this type of rollover without substantial releases of flammable liquid but there is some limited evidence to suggest this may not always be the case, with the risk increasing as initial travel speed increases. The literature suggests that incidents where the vehicle rolled and then collided with an object on or off the carriageway, such as a roadside barrier or tree, were among the most likely to result in significant releases of fuel.

Front to rear collisions were also found to present significant risks of substantial releases of flammable liquid, both when the tank of the FL vehicle is hit at the rear and damaged by a direct impact and, in the case of articulated vehicles, when the FL vehicle is involved in a frontal impact and collision forces are transmitted to the tank indirectly through the fifth wheel and king pin assembly / support structure.

Based on these Part A results, the focus for the Part B research (by TRL and HSE SD) was to develop cost-effective tank integrity assessment methods for the selected accident configurations, namely rollover and frontal impact (of an articulated petroleum road fuel tanker). Further technical and analytical support was provided by TWI Ltd in “Part C”.

Development of impact modelling and associated test parameters - rollover

The accident analysis work 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/road surface before the tanker comes to rest or impacts an object, which may penetrate the tanker shell. To ensure the development of a complete and robust assessment for rollover type accidents, the approach taken for the work was to develop methods to assess each of these three distinct events (topple, abrasion and penetration) individually, with most of the work focused on the initial topple and road-surface impact.

Initial topple and road-surface impact

Several potentially suitable test methods to simulate a rollover impact were considered but only two were found to be sufficiently worthy of further detailed assessment; topple testing and drop testing.

A topple test has the major advantage of being most directly representative of a real-world rollover impact scenario, albeit without any forward motion or subsequent abrasion as the tank slides along the ground. The tank can be filled with liquid, making the impact energies and resulting structural deformations also representative. The feasibility of a full-scale topple test was demonstrated in the previous DfT-sponsored research, which showed the method to be a controllable and repeatable test (see Figure ES-1). That research also demonstrated that the results of such a topple test could be predicted with a good degree of accuracy by finite element modelling.



Figure ES-1: Full-scale topple test

A drop test is likely to be a straightforward design and fabrication activity (particularly for tank subsections). Repeatability between different test houses would also likely be good as the design of the drop rig would not influence how much energy the tank structure would need to absorb – only the pad onto which the tank was dropped would have to be specified to a suitable level of rigidity. To generate realistic impact speeds, the drop height is only likely to be about 1 m and can be readily adjusted to vary the impact energy. Dropping from a fixed crossbar frame is considered likely to be the most suitable test rig set-up, as represented by Figure ES-2.

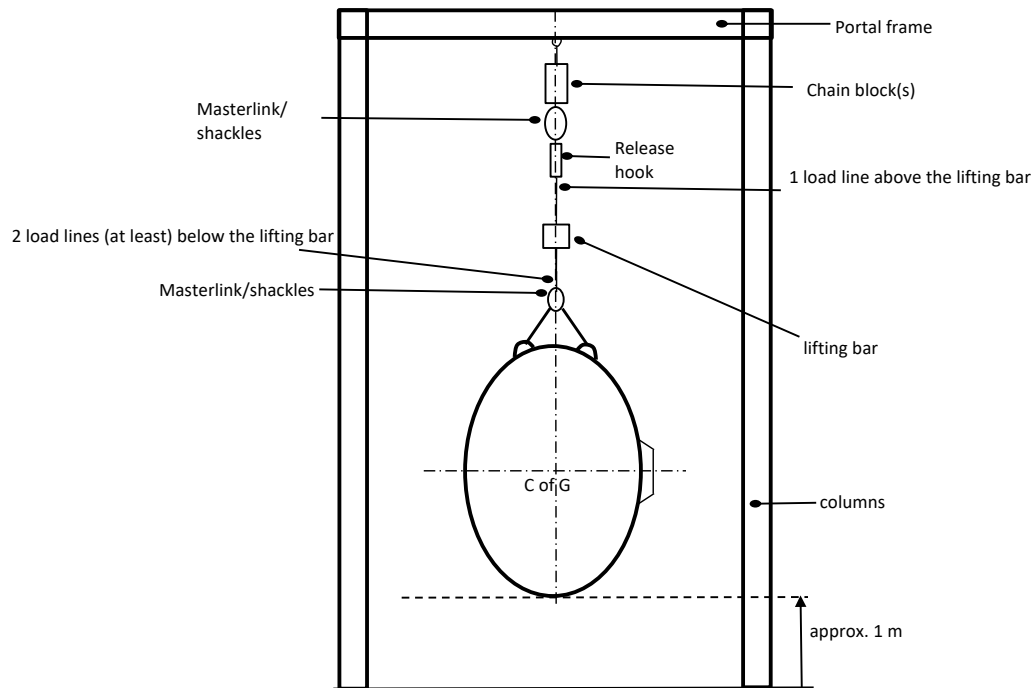


Figure ES-2: Potential drop-test rig set-up

The basic premise of this new research was that a full-scale tanker topple test was already validated and demonstrated to be a suitable test method. Should the manufacturer of a novel tank design (varying structurally from the relevant design-based sections of ADR) wish to demonstrate that their design had an equivalent (the same or better) level of safety in a topple scenario (to a minimally ADR-compliant design), the assumption is that one option for them would be to carry out such a full-scale test. This new research would thus focus on alternative methods that would likely be less costly to perform (e.g. by impacting only a number of subsections rather than the entire tank) but still provide a similar level of realism in simulating a full-scale topple event.

Additional topple test modelling confirmed the basic premise that a full-scale topple test would be a suitable test method, but also further emphasised the potential advantages of a subsection drop test – not least in its inherently more predictable and less variable liquid behaviour and its suitability for easily varying the impact speed and energy (to produce the desired levels of deformation and plastic strain). The research therefore turned to focus on the further development of modelling and assessment options for subsection drop tests.

The first step in assessing the full feasibility of a test method based on subsection drop testing was to simulate a drop test of a complete tank, to ascertain, for example, how structural deformations might differ from the topple test benchmark under otherwise similar conditions of impact speed and energy.

From these initial models, it can be concluded that a vertical drop results in higher levels of deformation than an equivalent topple, based on matching the (vertical) speed at the point of impact. With the topple case, the movement of the tanker and the liquid is about the pivot point, and so has a horizontal component, particularly further away from the pivot point. Some of this horizontal movement will still be present after the impact – meaning not all the kinetic energy is dissipated in the impact and absorbed by the tank structure and fluid

within. Conversely for the drop case, all movement is vertically downwards, and it comes to a near complete stop after impact – meaning almost all the energy has to be absorbed and, consequently, deflections are greater.

Initial modelling of the complete tanker in a topple test showed that the amount of tank deformation varied along the length of the tanker, with more deformation occurring at the rear than at the front. For a performance-based assessment based on a tanker subsection, it is important to be able to select the subsection that adequately replicates the performance of the most vulnerable part of the complete tanker design. Initial subsection drop-test modelling showed that if the front part of the tanker is chosen as the test section (to incorporate its different cross-sectional profiles) the fill level or drop height would need to be altered to achieve the higher levels of deflection that would typically occur at the rear of the tanker. However, this would result in higher deflections to the front bands than would be realistic in a real-world topple scenario. With this new knowledge, attention turned to modelling in much greater detail how the subsections for drop testing could/should be specified and if/how factors such as the number of compartments, their dimensions, the impact energy and different model outputs could be used to design a suitably robust regulatory test protocol. Given the detailed nature of the work and the models' relevance only to the structural behaviour of conventional metallic tank structures in representative rollover conditions, a decision was also taken at this point to exclude any potential non-metallic tank designs from further consideration.

A parametric study was instigated to:

- Conduct further subsection drop test modelling to establish the correlations between key test variables and assess whether these correlations result in a viable test method, and;
- If the drop test method is shown to be viable, to provide initial variables/conditions for test method design/development.

Three different tank configurations were used for this study, all based on designs that have been approved under ADR and all of aluminium construction. Two were two different types of banded designs constructed with circumferential welded bands joining the tank compartments (hereafter referred to as “banded type 1” and “banded type 2”) and one of a stuffed design constructed with partitions pressed and welded into the tank shell.

The use of forming limit diagrams is a widely established method in assessing if sheet metals that are strained in the forming process are likely to structurally fail. If the forming limit has been exceeded, then failure is likely (i.e. the shell or bulkheads have been bent beyond the point of breaking). 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. Failure becomes more likely the higher the omega value is above 1. Omega is thus a useful measure of how close or how far the strain conditions are to or beyond the forming limit.

Modelling of a banded type 1 tanker with one, two and three compartments in a single subsection showed that the important test parameter was the impact energy per partition, rather than just the impact energy. With energy per partition standardised, impact

responses followed the same trends for one, two and three compartment subsection models, and also for two complete banded type 1 tanker models with eight and ten bands.

There was good correlation between deflection and impact energy per partition for each of the three tanker designs assessed (two banded, one stuffed). All three designs showed a power-law relationship between the deflection and impact energy per partition with strong correlations ($R^2 > 0.9$).

For the metallic tankers assessed, this study has identified strong relationships (large coefficient of determination R^2) between impact energy and structural response parameters for tanker drop models. This shows that it should be possible to replicate structural responses observed in full-scale topple tests in a subsection drop test, with appropriate impact energy (which is directly related to compartment length, fill volume, liquid density and drop height).

For the banded designs, omega values were fairly constant within the range of 30 kJ to 70 kJ (per partition), which is the range of interest as this is comparable to the energy absorbed per partition in a full-scale topple test. However, there is a clear difference between the designs, with the banded type 1 design having a significantly higher value of omega, indicating a significantly higher risk of failure. These results suggest that, at least for these banded designs, a subsection drop test could differentiate between 'low risk' and 'high risk' designs across a wide range of impact energies based on a comparison of the predicted omega values for each design.

Omega values were lowest for the stuffed design and showed little change in value at impact energies of around 30 kJ to 70 kJ/partition (much like the banded tankers). These results agreed with previous separate research by TWI (confirming lower omega values for the banded type 2 and stuffed design).

To explain differences in deflections along the tankers' length witnessed in the full-scale topple tests carried out for the previous research, further work was commissioned to build a deeper understanding of how the energy of impact is distributed (unevenly) across the bands and how a performance-based test method could account for such differences.

While reasonable agreement between modelling outputs and physical tests results was found for the banded type 2 tanker, the actual test measurements of radial deflection were found to be substantially below the calculated values for the type 1 tankers tested.

However, the main purpose of the assessment of the experimental data was to observe the deflection response trends along the length of the tankers. In this respect, there was good agreement between measured and modelled deflections with two observations confirmed: highest deflections occur at the rear of the tanker; and deflections increase at the front of the tanker where the compartments are higher above the kingpin (although the deflections are still lower than at the rear). The ratio between the maximum deflection and the corresponding average deflections (across all bands/partitions) was found to be consistently in the range 1.15 – 1.24.

Abrasion and penetration

The approach taken to develop potential outline test methods for assessment of the resistance of a tank to penetration and abrasion in a rollover incident was to:

- Identify potential test method candidates from general review of test methods used for the assessment of penetration and abrasion, with focus on regulatory test procedures and testing of coupon type samples.
- Assess potential candidate test methods, based on their advantages and disadvantages in terms of technical feasibility, practicality, and cost, to produce a short list for further assessment and development.
- For each short-listed candidate outline potential work required to develop test method further.

Resistance to penetration requirements already exist in ADR via the referenced standard EN 13094 and can readily be adopted and/or adapted for approving novel designs.

Various test methods potentially relevant to abrasion testing of novel tanker designs were assessed but none was found to be directly applicable without further development. Concept test methods based on a grinding wheel and/or tyre durability testing were selected as most likely to be suitable for further development into an abrasion resistance test for tanker rollover scenarios.

Outline Technical Code for rollover resilience

An Outline Technical Code for assessing the rollover resilience of novel metallic tanker designs has been developed, incorporating the results of the research to-date on abrasion and penetration testing and those arising from the topple/drop-test modelling.

This outline technical code includes a suggested methodology for calculating the required impact energy for a two-compartment subsection drop test, and the application of this methodology has been verified by comparing modelled deflection and omega predictions with those from full-scale topple testing of the same design.

Summary of rollover findings and suggested next steps

The main conclusion of this research regarding the development of performance-based requirements for rollover safety is that:

The deflections and likelihood of major loss of containment experienced by tankers in real-world rollover scenarios can be replicated in a suitably specified, two-compartment subsection drop-test (or a full-scale physical topple test) supplemented by abrasion and penetration tests.

The research into rollover impacts and the development of an outline technical code for future performance-based requirements has, however, the following main limitations:

- The findings are based on somewhat simplified and idealised representations of tankers. They may not represent the full range of actual real-world designs.
- All three designs modelled represent tank structures that have been approved under ADR requirements. No “novel designs” have been assessed.
- Non-metallic structures have not been investigated.
- Subsection drop-tests are considered to be the most cost-effective option (instead of full-scale topple tests), but the likely savings have not been quantified.

It is reasonable to assume, based on some informal discussions, that a number of manufacturers may wish to make use of the greater flexibilities to innovate in tanker design that any revisions to ADR and its associated standards and technical codes arising from this research may provide. New designs to accommodate potential future increases in vehicle weight limits were identified as being of particular interest. To make use of a performance-based technical code, however, the cost and regulatory burdens associated with gaining approval via performance-based testing would need to be viable so as not to undermine the business case for such innovation. As the research progressed, it became increasingly evident that fully providing for greater flexibilities to innovate in tanker design will inevitably be a lengthy and complex undertaking, especially where a much larger number of “what-if” scenarios would need to be considered and any risks of adverse safety outcomes fully mitigated, across the fewer frontal, side and rear impacts as well as for the more frequent rollover impacts. Even limiting the scope of new performance-based tests to just metallic gravity-discharge tanks is likely to require much detailed study and careful validation to satisfy the relevant authorities.

Given the limitations, we suggest that any further work include an assessment of the market potential for novel petroleum fuel tanker designs (for example against an expected background of productivity concerns and declining petrol/diesel sales) and use the results to quantify the likely cost-effectiveness of any detailed further work.

If this assessment is positive, additional work to complete the development and validate/demonstrate the suitability of the Outline Technical Code (for rollover) could proceed, and frontal, side and rear impact scenarios considered in detail.

Development of impact modelling and associated test parameters – frontal impact

Front to rear collisions can present significant risks of substantial releases of flammable liquid. In the case of FL articulated vehicles with self-supporting trailers, 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. The (Part A) accident data work highlighted one collision in which the front of one fuel tanker struck the rear of another, and there was a substantial release of flammable liquid. On the basis that loading via the king pin assembly in frontal collisions (i.e. front-to-rear and front-to-front collisions) can present significant risks of substantial releases of flammable liquid, this type of impact loading scenario was chosen as one worthy of further investigation within this (Part B) research. The objective was to model this impact loading scenario (see Figure ES-3) to better understand damage and failure mechanisms and help set baselines for future potential test methods.

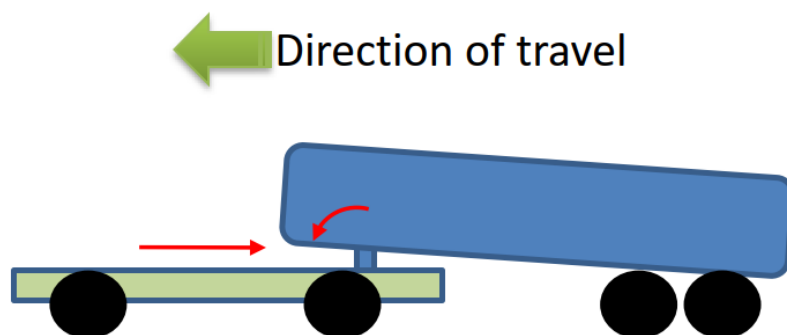


Figure ES-3: Modelling of king pin assembly loading during a frontal collision

Importance of kingpin support length

The baseline model used for this work was the same full tanker finite element model developed and validated as part of a previous study for DfT which investigated roll-over type impacts. The model was of a banded type 1 tanker design whereas the tanker involved in the above collision was a stuffed type design and had a shorter king pin support structure length. A shorter length of king pin structure will increase the forces required to resist the moment of loads applied through the kingpin assembly, which could lead to an increased risk of tank rupture. The effects of king pin structure length were thus a key focus for the research.

The modelling results show that much lower loads could be sustained without a buckling failure (and possible rupture) for the short king pin assembly compared to the long king pin assembly, on average a 30% higher load could be sustained for the long king pin unit.

Simple calculations using $F = ma$ show that if the majority of the load is via the king pin assembly in collisions for a 2 *g* deceleration with a semi-trailer mass of 37 t the king pin load will be around 725 kN and for a 3 *g* deceleration, 1090 kN. Comparison of these loads with the modelling results indicates that, for king pin crash loads between 2 and 3 *g*, there is a high likelihood that failures in the region of the king pin will occur for tankers with short length king pin assemblies and some likelihood for long length king pin assemblies if sufficiently high decelerations are experienced, i.e. around 3 *g* or more. These deceleration magnitudes are less than half of the 6.6 *g* minimum required in UNECE Regulations No. 67, 110, and 100 for the mechanical integrity of propulsion system components in crashes, such as the attachments of pressurised fuel tanks (R67 & 110) and battery mechanical integrity (R100).

Suggested next steps

The accident analysis only found one example of tank failure in the region of the king pin assembly in frontal impacts. Given that results from the modelling lead to an indication of a high likelihood of failure for short length king pin assemblies, the question arises as to why more examples were not found. There are a number of possible reasons for this which include:

- Collision type rare
- Few tankers in fleet prone to failure type
- FE models not fully representative of real-world tanker designs and/or associated loading not fully representative of real-world collision loads

The following work could be performed to gather information to further assess the performance of tanker trailers in frontal impacts, understand better the risk of tank failure and flammable liquid spillage in these collisions, and help determine whether, or not, this risk is similar for all types of design of tanker or if it is related to particular types of design.

- Accident analysis to:
 - Expand analysis of ADR collision reports to identify further relevant collisions.

-
- Compare national (Stats19) data and ADR reports to understand better reporting levels for incidents involving petroleum road fuel tankers and gain confidence that the reporting system is sufficiently robustly capturing such incidents to be a reliable guide as to frequency.
 - Estimate better king pin loadings in relevant collision(s).
 - Tanker fleet survey to:
 - Determine proportion of tankers in current fleet potentially prone to buckling failure, i.e. those self-supported (i.e. without chassis) in area behind king pin and/or with a short length king pin assembly.
 - FE modelling to:
 - Improve the tanker model so that it is representative of the tanker involved in the collision highlighted in the accident analysis.
 - Improve the loading method and the load magnitude so that it is more representative of the loading experienced in the collision highlighted in the accident analysis.
 - Compare and contrast performances of different tanker trailer designs.

This work should be able to identify if a noticeable real-world issue exists for frontal impacts in general or with specific makes / models of tanker trailer and how it may be best addressed in view of ADR requirements, referenced standards and technical codes.

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1 Introduction

1.1 Research background

The Dangerous Goods Division 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 and do not needlessly hinder trade. Goods vehicles used for the carriage of certain categories 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 2023)², which references European standards EN13094 ‘Design and Construction – gravity discharge’ tanks and EN12972 ‘Testing, inspection, and marking of metallic tanks’. In the UK, owners of these vehicles must apply to the Driver and Vehicle Standards Agency (DVSA) for ADR certification. 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 is always changing. New materials or manufacturing techniques may well enable a manufacturer to design an FL tank vehicle that that doesn’t meet ADR but offers some substantial new performance benefit such as:

- **Safety:** increased resistance to rupture, lower centre of gravity etc;
- **Productivity:** reduced unladen mass, increased payload, reduced cost etc;
- **Environment:** increased fuel efficiency, improved end of life recyclability etc.

To permit such a new vehicle, then design prescriptive regulation would need amendment. This can be a lengthy and uncertain process, which often has the effect of significantly stifling technical innovation.

Performance-based regulation aims to control the desired outcome and not how that outcome is achieved. This requires performance tests and assessments that properly represent real-world collisions with a high risk of leakage of the product being carried. The vehicle will pass if no leak occurs, regardless of how it is designed. Such approaches allow innovation but can add to the burden of approval. Tests and assessments for performance-based requirements are usually more complex and expensive to undertake (e.g. a crash test) than the checks of design requirements (e.g. measuring material thickness). The key is identifying a balance that ensures at least an equivalent level of safety while providing

² UNECE Agreement concerning the International Carriage of Dangerous Goods by Road (ADR 2023): <https://unece.org/transport/standards/transport/dangerous-goods/adr-2023-agreement-concerning-international-carriage>

manufacturers with the freedom to innovate, ensuring the complexity and cost of proving compliance does not become a new barrier to innovation.

Several years ago, when a number of petroleum tankers operating in the UK were found not to be compliant with existing ADR regulations, the Department for Transport sponsored research to understand the risks³. This involved a detailed review of collision data related to articulated tank vehicles, the development of a topple test procedure and associated finite element modelling assessments to explore the potential effects of the defects. This ultimately resulted in the progressive removal of the relevant vehicles from service.

Following this work, further research was performed which used the finite element modelling techniques developed to investigate the risk of failure of ‘partition and end to shell wall joints’ contained in EN 13094 Annex D (informative) examples of welding details⁴. The modelling method used a forming limit diagram approach, and the results showed a high risk of failure for informative joint design D.14(a) in EN 13094:2015 in a rollover type accident. Therefore, its removal from the standard was recommended and subsequently implemented.

As this work had demonstrated that the finite element modelling techniques developed could quantitatively assess the mechanical performance and susceptibility to failure of, in particular, structural circumferential joints of tanker trailers in rollover incidents, it was recommended that the method should be included in EN 13094 for assessment and approval of novel joint designs, i.e. those not contained in informative Annex A. To date this recommendation has not been implemented. However, it provides an example of how performance-based methods could be introduced into the ADR-relevant standards and help meet the objectives of this project.

Building on this previous research, the aim of the research reported here is to develop ‘performance-based’ finite element modelling approaches and appropriate physical test procedures to approve tankers with novel designs that otherwise would not satisfy the current ‘design-based’ approval 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.*

1.2 Research structure

A further set of linked research projects was thus established to assess opportunities for performance-based safety test procedures for petroleum fuel tankers that may be used as an alternative to the constraints of the existing design and construction rules.

³ 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.

⁴ TWI (2016). Department of transport technical assessment of petroleum tankers: Assessment of BS EN 13094 lap and partition joint designs. Available from: https://assets.publishing.service.gov.uk/government/uploads/system/uploads/attachment_data/file/940245/Technical-assessment-of-BS-EN-13094.pdf

It is anticipated the new test procedures would enable manufacturers to develop tank shells using design and construction methods not necessarily depicted in the existing standards, but which are nevertheless able to sustain rollover, frontal, rear and side impacts without having to use a series of more costly full-scale tests to demonstrate the suitability of their tank shells. In so doing, the new test procedures will reduce barriers to new tanker designs and construction technologies, and further improve the regulations and standards.

The potential is that any new test procedures could be included in a new or revised reference standard in ADR, or in a technical code that may be recognised by a competent authority of a Contracting Party to ADR alongside the following ADR related standards for petroleum road fuel tankers:

- BS EN13094 Tanks for the transport of dangerous goods – Metallic gravity-discharge tanks – Design and construction. (BSI, 2022)
- BS EN12972 Tanks for transport of dangerous goods – Testing, inspection and marking of metallic tanks. (BSI, 2018)

1.2.1 *Part B: Development of impact modelling and associated test parameters*

The main body of this report relates specifically to the methodologies and findings of the part of this new research referred to as work package “Part B”, which was to conduct impact scenario modelling and develop appropriate physical testing parameters. It was led by TRL Ltd in conjunction with HSE’s Science Division (HSE SD).

The main outputs of this research are 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. A secondary output was the development of a better understanding of a frontal impact (tank rupture) scenario and the associated loading of the tank structure through the kingpin assembly / support structure. This work has revealed a potential shortcoming with the current ADR standards and is also described in detail in this report.

1.2.2 *Parts A and C*

Two further work package research projects were commissioned by DfT:

- A review and analysis of accident data, impact conditions and regulations (“Part A”). The results from this research (led by Apollo Vehicle Safety Ltd) helped provide real-world accident context for Part B.
- To provide for detailed Engineering Critical Assessments (ECAs) of the impact conditions (“Part C”). This activity was led by TWI Ltd and ran closely alongside Part B and supported TRL and HSE SD with additional specialist expertise to guide their research design and assist in the detailed interpretation of modelling results.

1.3 Organisations involved

Parts A, B and C of the research were each competitively tendered. The appointed contractors were Apollo Vehicle Safety Ltd (Part A), TRL Ltd & HSE SD (Part B) and TWI Ltd (Part C).

1.3.1 *Peer review and project management arrangements*

As well as leading their own Work Package, each consortium member had extensive involvement in peer reviewing the techniques deployed and emerging findings from the other individual Parts. This peer review function was further aided by the involvement of a fifth organisation; Cambridge University Engineering Department (CUED, Prof Michael Sutcliffe).

A sixth organisation, BJR Solutions Ltd and its Director, Brian Robinson, also provided peer review support and was contracted by DfT to project manage the procurement and delivery of Parts A, B and C. In the latter stages of the research, he was contracted directly by TRL to support the drafting of this report.

1.4 Structure of this report

Section 2 provides further details of the background context for Part B, both in terms of the research and regulatory environment and via a summary of the key findings of the real-world accident study carried out by Apollo Vehicle Safety (Part A).

Section 3 describes the detailed modelling and associated development of physical test parameters for the most important tanker impact scenario highlighted by Part A; rollover. It describes the potential test methods and presents the results from extensive impact modelling of complete tanks and subsections for the initial (topple and ground impact) phase of a rollover accident as well as discussing how performance-based approaches might also consider the subsequent sliding along the ground phase through penetration and abrasion testing. It also describes the key features of an outline draft technical code for the performance-based approval of novel metallic tank designs.

Section 4 describes some preliminary modelling of a specific tanker frontal impact collision scenario, also identified in Part A, and discusses its potential implications for ADR design requirements.

Section 5 rounds off the main body of this report by drawing some conclusions, discussing the remaining key knowledge gaps and limitations of the research to-date and making suggestions for further research to support DfT's objectives.

2 Research background and road safety context

The Agreement concerning the International Carriage of Dangerous Goods by Road (ADR) was done at Geneva on 30 September 1957 under the auspices of the United Nations Economic Commission for Europe (UNECE), and it entered into force on 29 January 1968.

The Agreement itself is short and simple. The key article is the second, which says that apart from some excessively dangerous goods, other dangerous goods may be carried internationally in road vehicles subject to compliance with:

- “the conditions laid down in Annex A for the goods in question, in particular as regards their packaging and labelling; and
- the conditions laid down in Annex B, in particular as regards the construction, equipment and operation of the vehicle carrying the goods in question.”

Among other things, to address issues arising with vehicles in service and to adapt to technological progress, Annexes A and B have been regularly amended and updated since the entry into force of the ADR. Within Annex A, Chapter 6.8 of the agreement contains requirements for tanks, including petroleum tanks which transport flammable liquids by road. Most of these are mandatory design type requirements, some of which are contained in standards referenced in Chapter 6.8 of ADR such as EN 13094. The regular amendments have resulted in the ADR reflecting designs which have evolved over time and been shown to be safe in service.

However, prescriptive type requirements can stifle technical innovation and new designs because they inevitably only allow conventional, already-proven approaches, even if new technological advances might render them sub-optimal. The objective of the project is thus to develop “performance-based’ regulation to approve tankers with novel designs that would not meet current ‘design-based’ approval requirements, *i.e. to provide an alternative means of approval that gives more freedom to innovate while maintaining an equivalent (the same or better) level of safety.*

To provide an alternative means of approval using performance-based methods will require ways to assess that the integrity of the tank (in terms of containment of its contents) is maintained in the full range of real-world accidents (rollover, frontal, rear and side) at least to the level that a current ADR tank achieves, *i.e.* current safety levels should not be compromised. Potentially, this could be achieved through the introduction of a series of full-scale tests representative of the accident configurations seen in the real-world. However, such an approach would inevitably not be cost effective because of the large cost of full-scale tests, especially if design iteration was needed to meet approval requirements, which could lead to a need to repeat tests and an even larger associated cost.

Therefore, research was required to develop cost effective methods to assess a tank’s integrity. The first step of this work was to understand the type and nature of the accidents experienced by petroleum road fuel tankers in the real-world and the associated risk of fuel spillage. This work (carried out by Apollo Vehicle Safety under their Part A contract) also identified candidate test procedures that, potentially, could be adapted into performance-based methods. Its main results are summarised below.

2.1 Summary of outcomes of accident analysis work (Part A)

To better understand the safety of the GB fleet certified to carry flammable liquids (FL vehicles) and to provide the foundation to develop performance-based cost-effective methods to assess a tank's integrity, Apollo Vehicle Safety analysed recent collision data and updated previous literature reviews on the causes of releases of liquids from tank vehicles⁵. Apollo also reviewed regulations and test procedures intended to assess vehicle safety performance in the types of collisions most frequently experienced by tank vehicles.

The work involved a combination of analysis of the GB national collision database, the Road Accident In-Depth Study (RAIDS) database, identification of relevant collision case studies worldwide and a comprehensive review of existing literature and standards.

2.1.1 Review of accident data

It was found that the safety performance of the FL vehicle fleet in Great Britain is broadly comparable to that of the general population of heavy goods vehicles. Some specific differences in the type of collisions suffered and the rate of collisions per vehicle were found but these can mainly be attributed to the fact that FL vehicles are more likely to be large articulated vehicles and less likely to be small rigid vehicles than is the case for the general HGV fleet.

Almost all significant releases of flammable liquids arise from traffic collision incidents involving rollover and/or collision with another heavy vehicle. These represent only a small proportion of all traffic collision incidents involving FL vehicles.

Within this small group of high-risk collisions, rollover is by far the most likely to result in significant releases. There is evidence that incidents involving significant releases do occur in impacts to each of the front, rear and side of a tank vehicle. However, variations between different small samples of data in different countries, creates uncertainty around exact proportions. It is considered appropriate in this case to treat front, rear, and side impact as having a probability of release equal to each other, though much less than that of rollover.

Three sub-groups of rollover were identified:

- Simple on-carriageway, rollover and slide to rest
- Rollover and run-off road
- Rollover and collision

Overall, a tank vehicle rolling over is a very rare event in GB. Where it does occur, simple on-carriageway rollover was found to be the most common type. Although not quantifiable in national data, the literature strongly suggested that this will typically involve a 90-degree roll onto the side of the vehicle. There was evidence to suggest that tanks typically survived this type of rollover without substantial releases of flammable liquid but some limited evidence to suggest this may not always be the case with high initial travel speeds.

⁵ Knight and Dodd (2020). Performance test procedures for petroleum road fuel tankers – Review and analysis of accident data impact conditions and regulations. Apollo Vehicle Safety. Report for Part A of DfT contract P4/030/0013c

Incidents where the vehicle both rolls over and leaves the carriageway (in whichever order) but does not suffer collision with an object are less common. However, it was concluded that the likelihood of a substantial release in these incidents would be much larger because the literature suggested that in these incidents it is more likely that the vehicle will roll through more than 90 degrees, longitudinal loads on the tank could be increased, and that point loading is more likely.

A scenario involving a rollover and collision is most frequent in GB but will include an unknown proportion of cases where a minor collision (e.g. with a light vehicle) was causative of a rollover but it was the damage from the rollover that presented the risk to the tank. The risk in these cases is similar to situations where the rollover is caused by cornering forces, that is, similar to one of the previous two categories, depending on whether the vehicle left the carriageway or not. However, the literature suggests that incidents where the vehicle rolled and then collided with an object on or off the carriageway, such as a roadside barrier or tree, were among the most likely to result in significant releases of fuel.

Front to rear collisions were also found to present significant risks of substantial releases of flammable liquid, both when the tank of the FL vehicle is hit at the rear and damaged by a direct impact and, in the case of articulated vehicles, when the FL vehicle is involved in a frontal impact and collision forces are transmitted to the tank indirectly through the fifth wheel and king pin assembly / support structure.

Minimal information was identified about the characteristics of collisions between another heavy vehicle and the side of a tank vehicle.

2.1.2 Existing performance-based test procedures of relevance

Candidate tests procedures, which have potential to form the basis of a test for the tank vehicle, were identified in regulatory research for the accident configurations as follows:

Rollover:

- UNECE Regulation No. 66 – strength of the superstructure of buses in rollovers
- US FMVSS No. 220 – school bus rollover protection
- UNECE Regulation No. 29 – commercial vehicle cab strength
- SAE J2422 – heavy truck quasi-static cab roof strength evaluation

Front / rear impact:

- UNECE Regulation No. 29 – commercial vehicle cab strength
- SAE J2420 – heavy truck dynamic frontal strength evaluation

2.2 Implications for complementary research (Parts B and C)

On the basis of the Part A results described above, the focus for the Part B research (by TRL and HSE SD) was to develop cost-effective tank integrity assessment methods for the selected accident configurations, namely rollover and frontal impact (of an articulated petroleum road fuel tanker).

3 Part B: Development of impact modelling and associated test parameters -rollover

The accident analysis work described in Section 2.1, 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/road surface before the tanker comes to rest or impacts an object, which may penetrate the tanker shell. However, the sequence of events of a rollover accident can vary. For example, the tanker can topple onto an object (e.g. a road barrier or street furniture) that could penetrate the tanker, which is then followed by a period of sliding. From this, the following three distinct events potentially involved in rollover accidents and which can lead to the loss of the tank integrity were identified:

- Simple topple and impact
- Abrasion related to sliding, initially with high normal forces for a short period during the impact phase and subsequently with lower normal forces for a longer period during the sliding phase.
- Penetration related to impact

To ensure the development of a complete and robust assessment for rollover type accidents, the approach taken for the work was to develop methods to assess each of these three distinct events individually, with most of the work focused on the simple topple and impact. A performance-based test regime for any one of these events could lead to the unintended consequence that safety levels in respect of one or both of the other two are compromised, e.g. strong joints leading to good performance in the topple phase test would not guarantee a sufficiently well-constructed tank outer shell to withstand abrasion loading when that tank slides along a road surface. It is therefore important that any performance-based approach considers all possible failure mechanisms.

As mentioned above, the overall aim of the research was to develop methods to approve tankers with novel designs that would not meet current 'design-based' approval requirements while maintaining an equivalent level of safety. On this basis the approach was taken that the acceptance requirements for the test methods, where defined, were set to demonstrate a performance level which is equivalent, as far as practical, to the performance of a tanker which currently meets the ADR requirements.

This research is described thoroughly in the following sections and a route to its future adoption in international standards/regulation is presented in the form of an outline technical code for rollover resilience.

Three different tank configurations were modelled for this study, all based on designs that have been approved under current or earlier versions of ADR and all of aluminium construction, two being banded-type designs and one a stuffed design. Further details are provided in the following sections.

3.1 Topple and impact – assessment of potentially representative tests

Six potentially suitable test methods to simulate a rollover impact were considered. The following sections describe, first and briefly, those methods rejected from further consideration and the key reasons for rejection and second, the two methods considered sufficiently suitable for further detailed assessment.

Each method was assessed initially on a range of criteria including whether such a method was already in use for international standards purposes, if adaption to cater for tanker rollover impacts was feasible, how practical such a test would be to implement (in each case with either a complete tank or subsections thereof), how repeatable its results would be, how representative of a real-world rollover impact it would be, etc. Appendix A presents the full details of each method's assessment – here we present only those most pertinent.

3.1.1 *Test methods rejected from further consideration*

3.1.1.1 *Pendulum impact testing*

While such methods are already in use, most notably for UN ECE Regulation 29 (HGV cab strength requirements), their feasibility in a tanker rollover scenario was considered to be low. The impact energy imparted by the pendulum would likely be quite low (without a very large and heavy test rig) and thus to achieve realistic levels of structural deformation, the tank would probably need to be empty. If a liquid load was used, its dynamic post-impact response would not be the same as in a real-world topple. There would also be considerable scope for variability in results from otherwise like-for-like tests due to minor variations in test rig design and set up affecting exactly how much energy is absorbed by the tank (and how much by the test rig and other support structures).

3.1.1.2 *Horizontal impact testing*

Specialist vehicle horizontal impact sleds are in widespread use, e.g. for passenger car pole side impact testing but adapting those facilities for much bigger and heavier tank vehicles may be problematic and would incur considerable risks of non-availability of test facilities for tank manufacturers. Test results would probably be less reproducible if different test houses designed different rigs (e.g. to suit their particular circumstances and existing layouts). Impact energy requirements could be reduced if using an empty tank but, as with the pendulum impact method, how representative of real-world rollover impacts that would be, with fluid-structure interactions and different post-impact dynamics, is very difficult to predict.

3.1.1.3 *Drop testing onto angled ramp (and subsequent sliding)*

This test would involve dropping a tank onto an inclined ramp and it sliding down the ramp after impact. This has the key advantage of meaning both the main phases in a real-world rollover impact could be reproduced in one test – initial ground impact and subsequent sliding (abrasion) as the tank moves across the ground. However, the repeatability challenges due to friction could be significant as would ensuring test-to-test repeatability (each test would tend to smooth out whatever abrasive surface was on the inclined ramp).

Dropping a large, heavy structure onto an incline may also pose significant containment issues for the test house, as would ensuring minimal energy is absorbed by the ramp and its supporting structure.

3.1.1.4 *Inclined ramp to impact with perpendicular block testing*

Designing a rail trolley to attach under the tank to carry out a ramp impact test is the fourth and final test method considered but subsequently rejected. The basic set-up (at the beginning of the test) is depicted in Figure 3-1.

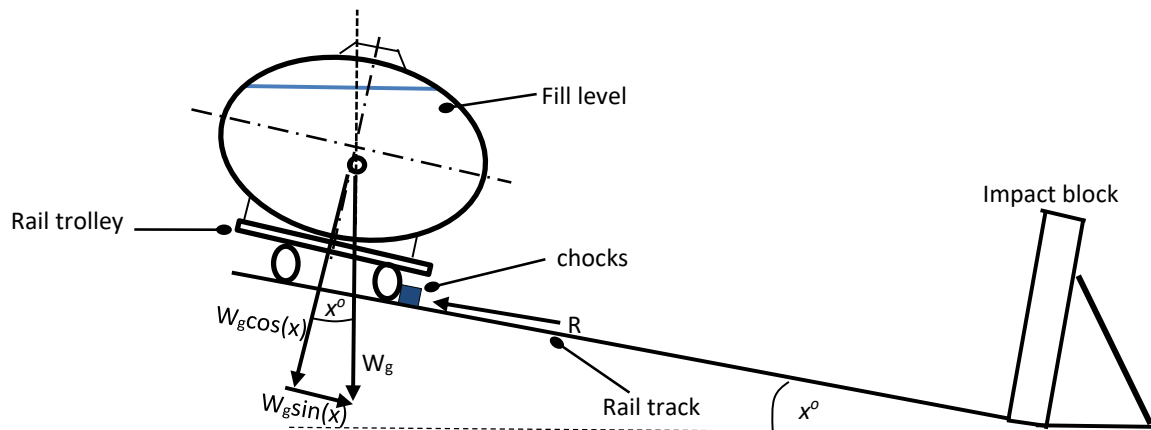


Figure 3-1: Guided ramp impact test (tank in position at the beginning of the test)

This method has some advantages over those described above, including that the precise impact point on the tank can be readily controlled and, potentially, the same basic rig (with appropriate modifications) could be used for other impact scenarios, e.g. frontal or rear impacts. It would also potentially allow for a more representative liquid angle at the moment of impact.

For ramp angles of more than about 25°, the trolley wheels would need to be restrained to prevent the tank rolling, however, and once built, modifying the rig to accommodate higher impact speeds/energy than originally envisaged could be extremely difficult.

Further assessment was made of the specific case of a 45° ramp angle test. This work, using detailed fluid and finite element modelling, found that the deformation of the tank in such a test scenario would be highly sensitive to liquid position. Slight variations in angle at impact, which would be inevitable in such a test, would cause much larger variations in the kinetic energy that would have to be absorbed by the tank structure. Primarily for this reason, this test method was also excluded from further consideration.

3.1.2 *Test methods suitable for further detailed consideration*

3.1.2.1 *Topple testing*

A topple test has the major advantage of being most directly representative of a real-world rollover impact scenario, albeit without any forward motion or subsequent abrasion as the

tank slides along the ground. The tank can be filled with liquid, making the impact energies and resulting structural deformations also representative.

The feasibility of a full-scale topple test was demonstrated in the 2015 research, which showed the method to be a controllable and repeatable test (Figure 3-2). That research also demonstrated that the results of such a topple test could be predicted with a good degree of accuracy by finite element modelling.



Figure 3-2: Topple test carried out for 2015 research

A modified version of the topple test, involving rotation of more than ninety degrees, was also considered. UN ECE Regulation 66 includes a topple test (for buses/coaches) whereby the vehicle is tipped from a raised platform onto a rigid horizontal surface 800 mm below. This would represent a real-world rollover from the road into a roadside ditch and is somewhat more onerous test for a tanker (than the original topple test) as the impact velocity would be higher and the loading to the structure more concentrated. This modification to the topple method was, however, rejected primarily because the accident data (from Part A) indicated such high rotation impacts to be much less common than the standard ninety-degree rotation scenario. It also indicated that releases of fluid were more common in this type of collision such that requiring zero release in this type of test would represent a level of safety in excess of the current approval level.

3.1.2.2 Drop testing

A drop test is likely to be a straightforward design and fabrication activity (particularly for tank subsections), which means it should be reasonable to expect other test houses to develop their own drop test methods. Repeatability between different test houses would also likely be good as the design of the drop rig would not influence how much energy the

tank structure would need to absorb – only the pad onto which the tank was dropped would have to be specified to a suitable level of rigidity (e.g. a 50 mm thick steel plate bolted to a concrete slab). Preliminary modelling showed it would be likely to give deflection results suitably close to those modelled in 2015 and used as the benchmark for this research.

To generate realistic impact speeds, the drop height is only likely to be about 1 m and can be readily adjusted to vary the impact energy. The tank should contain water to represent the fuel load and ensure that impact energy is also realistic.

Several variations of drop test were assessed, e.g. using different initial hold and release systems and flat or raised impact pads (see Appendix A). The conclusion was that dropping from a fixed crossbar frame was likely to be the most suitable test rig set-up, as represented in Figure 3-3.

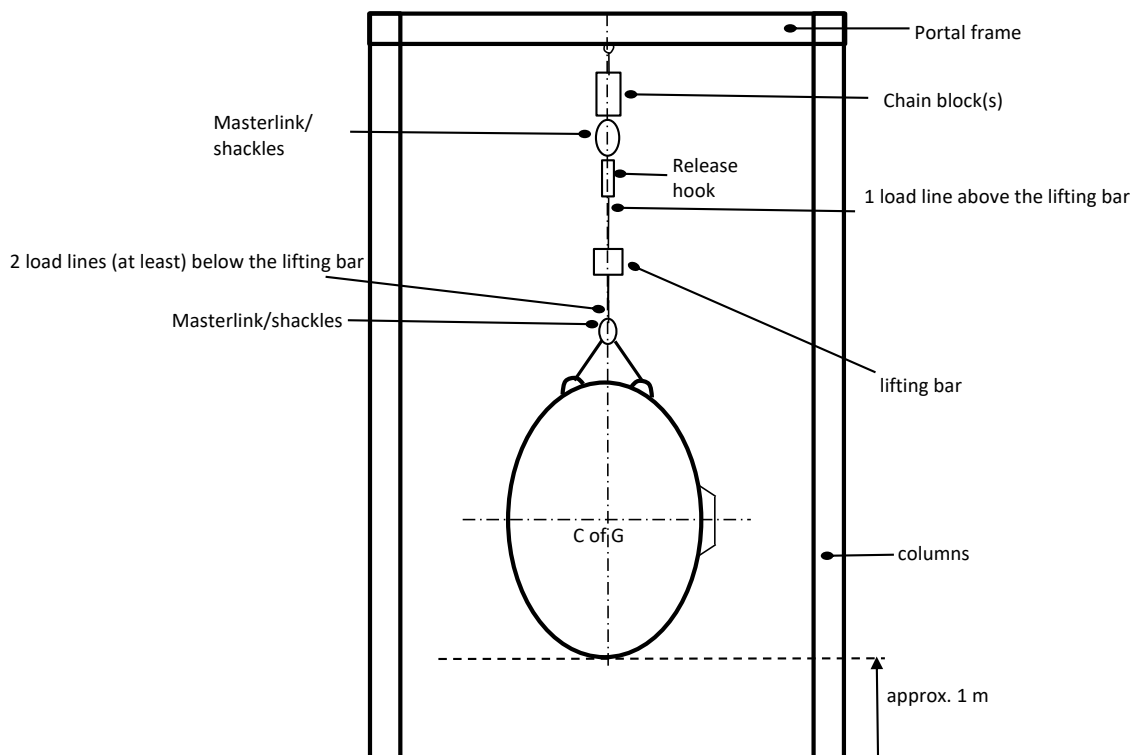


Figure 3-3: Drop test from a cross bar with one release point

3.2 General approach to modelling

Finite element methods for solving time dependent problems can be split into two main types: explicit and implicit. Explicit methods calculate the state of the system at a later time (Δt) based on the state of the system at the current time only. Because the time step must be small (the order of microseconds), to limit computer model runtime these methods are typically used for short impact type events, such as car crashes. Implicit methods find a solution by solving an equation involving both the current state and the later one. Much larger time steps can be taken using this method, so this method is typically used for longer timescale events. However, solving the equation can require much computer resource and sometimes it can be difficult to get it to converge to a solution. For modelling of the rollover

scenario (topple or drop), an impact type event, the Ansys Autodyn explicit solver has been used.

While it is important to remember that no model can ever capture the physical event it is representing with perfect accuracy, it is important that the model captures enough of the detail of the event to be useful. There are many ways in which models can fail to replicate reality, such as a failure of the model to capture the physics of the event, boundary condition inaccuracies, and errors made in the set up. Some may be easy to spot, and others may be more subtle.

For the rollover impact scenario modelling described in the following sections, HSE's approach to model validation was as follows:

- Start with the validated model of the topple test using water from the previous work.⁶ The model was validated via direct comparison of key parameters (deflections, flat lengths etc) measured in full-scale tanker topple physical tests.
- Intermediate models have not been validated against physical tests but have been produced in an incremental way to enable the sources of differences to be identified and evaluated. For example, changing the liquid in the topple model from water to petrol resulted in an increase in deformation despite an equivalent mass of liquid being used. This was investigated and found to be due to an increase in kinetic energy and angular momentum compared to water. This was because the petrol's larger volume and tank fill increased the height of the centre of gravity of the tanker and hence the length of the pivot radius in the topple, compared to water.

3.3 Further modelling of topple test

The basic premise of this new research was that a full-scale tanker topple test was already validated and demonstrated to be a suitable test method. Should the manufacturer of a novel tank design (varying structurally from the relevant design-based sections of ADR) wish to demonstrate that their design had at least equivalent (same or better) levels of safety in a topple scenario (to a minimally ADR-compliant design), the assumption is that one option for them would be to carry out such a full-scale test. This new research would thus focus on alternative methods that might be cost effective to carry out (e.g. by impacting only a number of subsections rather than the entire tank) but still provide a similar level of realism in simulating a full-scale topple event.

⁶ Assessing the safety of petroleum road fuel tankers (2015) Work Package 1 'Full scale testing and associated modelling' reports:

- ES-14-39-00: Full scale testing and associated modelling – overall summary
- ES-14-39-04: Tanker topple test methods and results
- ES-14-39-05: Modelling to provide load case data for rollover – approach and initial development
- ES-14-39-06: Modelling to provide load case data for rollover – validation and application

Available from <https://www.gov.uk/government/publications/petroleum-road-fuel-tankers-technical-assessment> accessed August 2023. Note download Work Package 1 reports.

In modelling the inclined ramp test method, however (described in the rejected test methods section above), a concern was raised that the exact liquid angle at the point of impact may be a key parameter and the resulting structural deformations may be very sensitive to quite small variations in this angle. Further modelling of the full-scale topple test scenario was carried out to investigate this concern and inform the research team on whether the topple test method shows similar sensitivity to liquid position and motion as the slide method.

3.3.1 *Assessment of effect of liquid and its motion in topple test*

A two-step approach was used. First, the liquid (petrol) behaviour was modelled (for a single compartment) in detail to estimate the liquid angle and liquid motion at impact. The second step used a structural response model (of two compartments) to establish the sensitivity of its results to small variations in this liquid behaviour.

To model the liquid behaviour in the structural response model, an approximation method was developed that gave sufficiently similar liquid behaviour to that predicted by the dedicated liquid behaviour model, but still allowed for reasonable run times of the structural response model. Trying to model both liquid behaviour and structural responses to a high degree of accuracy at the same time was not feasible due to processor/run time constraints. This “translation and rotation” simplification is described fully in Appendix B.

The detailed liquid model indicated an angle at impact of approximately 24°. The structural response model was run using angles of 20° and 30° to assess how much difference in structural response was likely if the angle was varied slightly about that central estimate.

Band deflections predicted by the structural response model showed very little variation (of < 1%) across these two angles. Plastic strains in the bulkheads were also predicted by the structural model and showed very little variation between the two angles (of < 4%).

The overall conclusion was thus that the tank’s structural response in a topple test scenario is not sensitive to liquid angle or precise movement of the liquid.

Further modelling of water as the liquid (rather than petrol) showed that the liquid behaviour was similar between the two (the water model also predicted an angle of impact of approximately 24°). It also demonstrated, however, that the deformations predicted for a two-compartment subsection topple test were both lower than those of a full-scale tanker topple test (petrol or water, as modelled and/or as tested in 2015) and consistently lower for a petrol load than if a water load of equivalent volume (and thus higher mass) was used. Plastic strains showed a similar pattern – higher for water load than for petrol load but still lower than for the full-scale topple test benchmark.

This additional topple test modelling, while confirming the basic premise that a full-scale topple test would be a suitable test method, also further emphasised the potential advantages of a subsection drop test – not least in its inherently more predictable and less variable liquid behaviour and its suitability for easily varying the impact speed and energy (to produce the desired levels of deformation and plastic strain). The research therefore turned at this stage to focus on the further development of modelling and assessment options for subsection drop tests.

3.4 Initial drop test modelling of complete tank and subsections

The first step in assessing the full feasibility of a test method based on subsection drop testing was to simulate a drop test of a complete tank, to ascertain, for example, how structural deformations might differ from the topple test benchmark under otherwise similar conditions of impact speed and energy. For this drop test modelling, the complete (8-banded) tanker model developed as part of the 2015 research was stripped of the fifth wheel (front support), the suspension, axles and wheels. Drops were modelled from a point just before impact with a petrol load and with the vertical velocity set to the appropriate value. The deflections of bands A (front), D (band after conical section) and H (rear) were compared for the drop models and the petrol topple case from the 2015 research.

The initial drop test model of the complete tank containing petrol used an impact velocity matching the topple case (4.5 m/s) but resulted in significantly higher deflections at the three measured band locations compared to the topple case, of up to 30% higher.

In the second model run, again of the complete tank with a petrol load, the impact velocity was adjusted down to 4.0 m/s in an attempt to match the deflections observed in the topple model. This was successful to the extent that while all three band deflections were still higher than the topple benchmark, they were within a smaller margin of error, of less than 10%.

Comparison was also made to the full-scale topple model results for a water load of equivalent mass to the petrol load, generated as part of the 2015 research. Despite having the same mass, the deflections of the bands in the model containing petrol were significantly higher than those containing water for the topple case. The likely explanation is that for the topple case, the tanker and its contents rotate about a pivot point which is along the outer edge of the wheels. Due to the assumed position of water/petrol just before impact, the additional volume of liquid (in the petrol case) is at a greater distance from the pivot point, therefore increasing its kinetic energy at the moment of impact. For the water filled case, the kinetic energy before impact was 310 kJ, compared to 375 kJ for the petrol filled case.

Subsections of a tanker were also modelled at this stage, consisting of the front three compartments only (Figure 3-4, including the conical section and consisting of bands A-E).

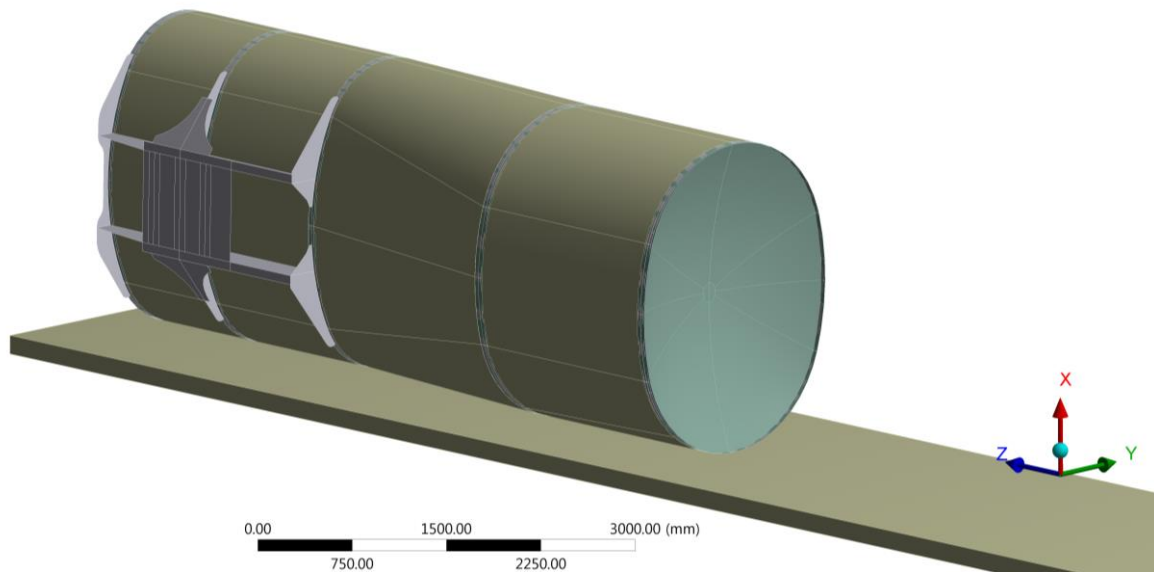


Figure 3-4: Subsection model of tanker consisting of front three compartments

As the rear compartment was not modelled in this first round of tanker subsection models, the deflection of the band at the rear of compartment three (Band E) was recorded. This location was also recorded for the complete tanker drop test model with an impact velocity of 4.0 m/s for comparison. In the configuration used here, this bulkhead bows into the last compartment modelled, so is just loaded on the convex side. In the full tanker configuration, with all the compartments filled, the only bulkheads loaded just on one side would be the end bulkheads (both loaded on the concave side). However, in operation, empty compartments are likely which would also result in bulkheads being loaded on the convex side only.

The same two impact velocities were modelled (4.5 m/s and 4.0 m/s) and as well as a petrol load, a water load was also assessed, making four subsection model runs overall. The lower velocity impacts produced deflections close to the predictions from the benchmark model, although with slightly lower deflections at Band A and higher at Band D.

For these subsection drop cases, there was very little difference between the petrol and water models and the predicted deflections were somewhat lower than the equivalent complete tanker models. In a vertical drop, the velocity of the full volume at impact is the same, and the location of the additional volume of petrol is irrelevant to the load's overall kinetic energy if the masses are the same in each case, so one would expect little or no difference in deflections between the petrol and water filled cases.

Results from the various model runs of complete tanks and subsections described above are presented in Figure 3-5.

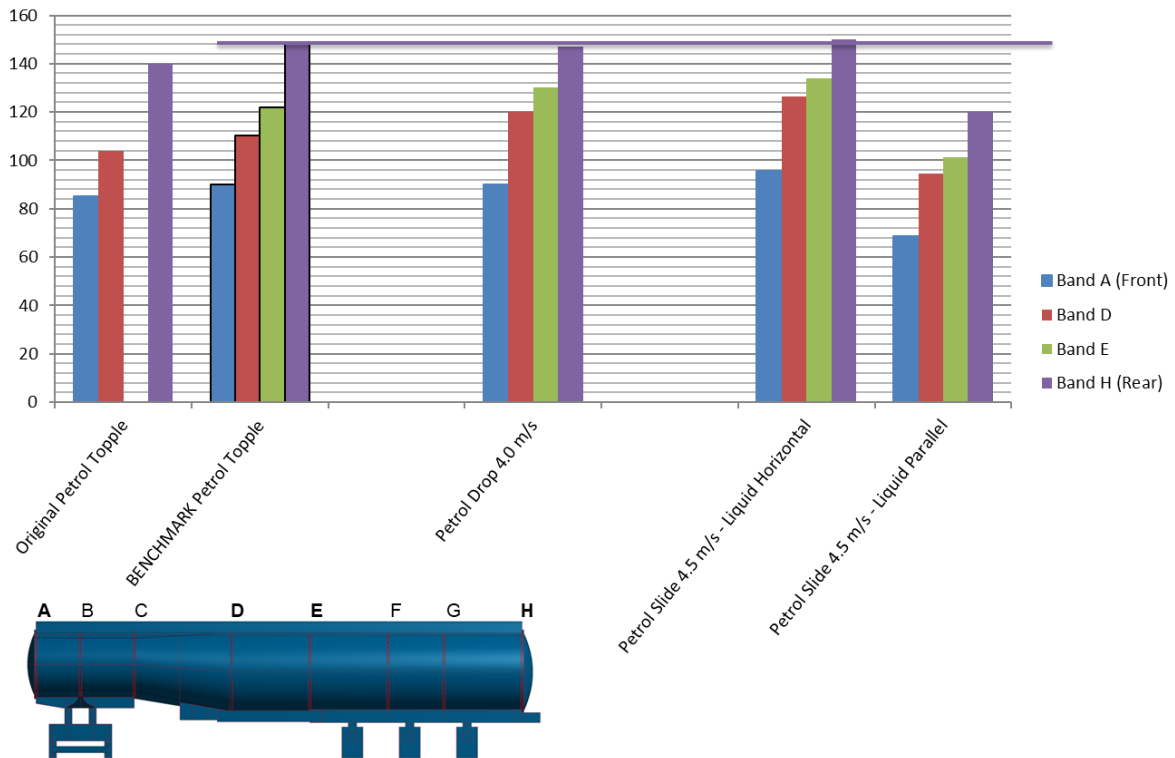


Figure 3-5: Summary of model results for deflections (mm)

Note that modelling of the benchmark petrol topple case here was performed with some parameters changed from the original model to make the model less specific to the previous test conditions. The specific changes to the model were:

- Fillet welds on all extrusions.
- Ground and tanker straightened.
- Slight increase in mass.
- Ground/tanker friction added.

3.4.1 *Lessons learnt and their Implications for further research*

From these initial models, it can be concluded that a vertical drop results in higher levels of deformation than an equivalent topple based on matching the (vertical) speed at the point of impact. With the topple case, the movement of the tanker and the liquid is about the pivot point, and so has a horizontal component, particularly further away from the pivot point. Some of this horizontal movement will still be present after the impact – meaning not all of the kinetic energy is dissipated in the impact and absorbed by the tank structure and fluid within. Conversely, for the drop case, all movement is vertically downwards, and it comes to a near complete stop after impact – meaning almost all the energy has to be absorbed and, consequently, deflections are greater.

The initial subsection modelling showed that if the front part of the tanker is chosen as the test section to incorporate its different cross-sectional profiles, the fill level or drop height would need to be altered to achieve the higher levels of deflection that would typically

occur at the rear of the tanker. However, this would result in higher deflections to the front bands than would be realistic in a real-world topple scenario.

With this new knowledge, attention turned to modelling in much greater detail how the subsections for drop testing could/should be specified and if/how factors such as the number of compartments, their dimensions, the impact energy and different model outputs could be used to design a suitably robust regulatory test protocol with associated pass/fail criteria.

Given the detailed nature of the proposed further work and the models' relevance only to the structural behaviour of conventional metallic tank structures in representative rollover conditions, a decision was taken at this point to exclude any potential non-metallic tank designs from further consideration in this research.

3.5 Further drop test modelling of tank subsection(s)

A parametric study was instigated to:

- Conduct further subsection drop test modelling to establish the correlations between key test variables and assess whether these correlations result in a viable test method, and;
- If the drop test method is shown to be viable, to provide initial variables/conditions for test method design/development.

The study involved multiple runs of HSE-SD's model, with key parameters varied for each run. Regression analyses were performed on the resulting data to determine if correlations exist and, if so, to develop mathematical expressions to characterise those relationships.

TWI performed regression analyses on HSE's model results to predict a wider range of responses beyond the discrete data points that HSE SD obtained. The regression analyses generated relationships (response surface functions) that link each of these data points together on a common plane. Two test parameters and one response parameter can be considered in one graph to generate a three-dimensional response surface. These results are essential to decide on suitable-sized subsections.

HSE SD and TWI also carried out forming limit calculations to predict if the tanker was likely to structurally fail and release content from compartment-to-compartment, or from compartment to the outside. In the 2015 research, modelling and physical tests on ADR-compliant designs, supported by accident data, had firmly indicated that major loss of containment was highly unlikely in a simple topple scenario, so this "no loss of containment" criteria was used as a suitable pass/fail test.

Full details of the study and its results are included in Appendix C. A summary is presented in the following sections.

3.5.1 Tanker designs

Three different tank configurations were used for this study, all based on designs that have been approved under ADR and all of aluminium construction. Two were two different types of banded designs constructed with circumferential welded bands joining the tank

compartments (referred to as “banded type 1” and “banded type 2”) and one of a stuffed design constructed with partitions pressed and welded into the tank shell.

The difference between banded and stuffed designs is primarily due to the way that the tanker is assembled. In banded tankers, the cylindrical sections of the tanker (known as the shell) are joined to a band at the ends of the cylinder. The band is like a reinforcing ring between the cylindrical sections (normally made from extruded aluminium). Banded tankers are typically 8-banded to 10-banded. The partitions that separate compartments within the tanker are also connected to the band, although some of these partitions may have a central hole or other openings in them, which means it is the same compartment on each side. This type of division is referred to as a ‘surge plate’. A surge plate increases the strength of the tanker and reduces liquid sloshing within the tank.

For the stuffed tanker design, there are no bands as the shell is continuous and the partitions are welded to the inside of the shell. As the stuffed design has no bands, they are generally less stiff than a banded design.

The principal structure of a banded tanker consists of three primary components: the tanker shell; extruded bands (the ‘extrusion’); and dish-shaped partitions which are often referred to as bulkheads (Figure 3-6).

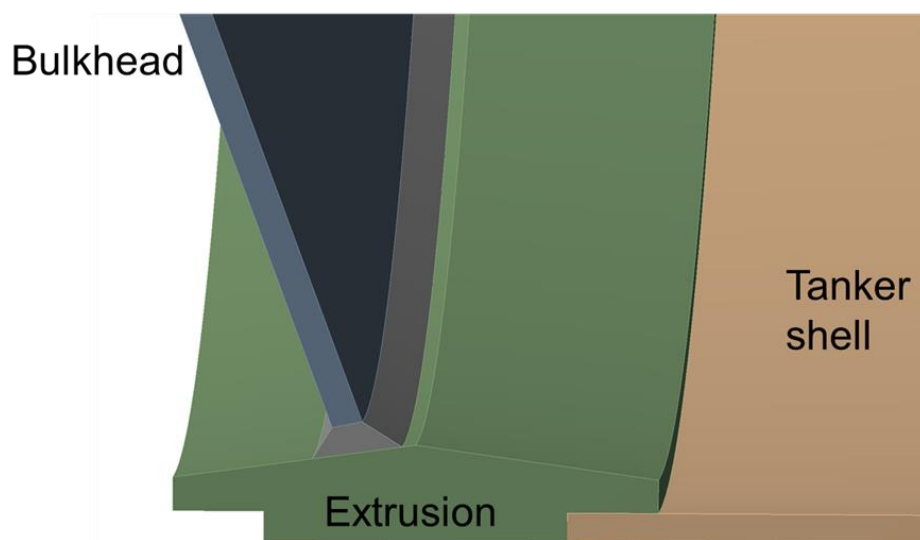


Figure 3-6: Main components of a typical banded tanker

The tanker shell is welded to the extrusion using an external butt weld with the addition of internal fillet welds in some designs. The connection between the extrusion and the partitions/bulkheads varies with the type of tanker design.

The main difference between the two banded designs assessed is that the type 1 design has the partition dish welded directly to the extrusion band. In the type 2 design modelled, there is an upstand on the extrusion band onto which the partition dish is attached.

The end partition dish of the modelled stuffed design is shown diagrammatically in Figure 3-7.

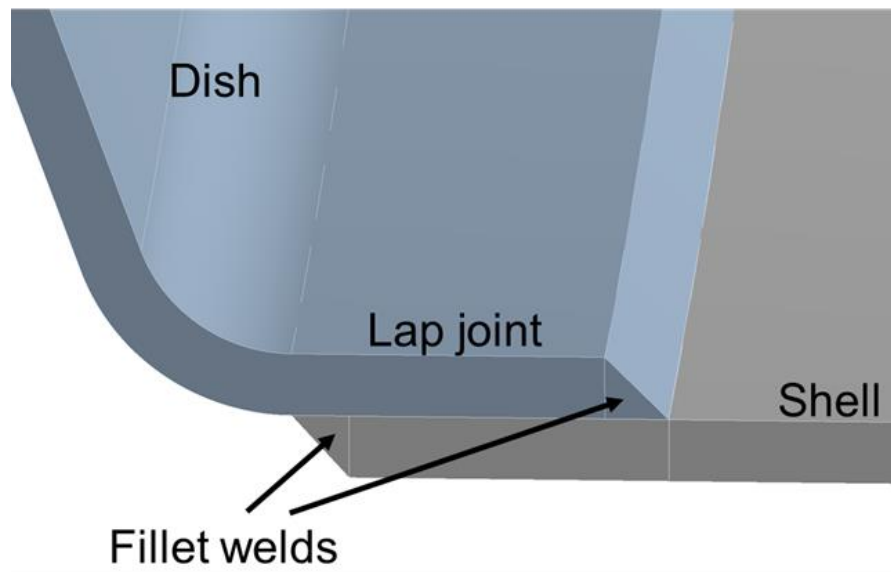


Figure 3-7: Joint detail as modelled – stuffed design

3.5.2 Modelling methods

HSE SD carried out the modelling using the Ansys Autodyn software. To keep computer runtimes for modelling reasonable an approach was deployed which essentially used two types of models. The two model types addressed the two key elements of the problem, namely the fluid structure interaction (FSI) and the tanker structure detailed behaviour. The FSI type model used a Eulerian domain to represent the fluid, and Lagrangian elements (2D shells) to represent the tanker structure. This type of model was used to predict the overall behaviour of the tanker structure (e.g. band deflections, energy absorbed) from a series of input test parameters (impact velocity, fill level etc.) taking into account precisely how the fluid loaded the tanker structure. It was also used to understand the sensitivity of tanker structure behaviour to changes in precisely how the fluid loaded the tanker structure. However, this type of model could not predict plastic strains to the accuracy required to predict the likelihood of tanker structure failure. Therefore, a detailed tanker structure type model was developed and used to determine relationships between tanker structure damage factors predicted by the FSI type model (e.g. band deflections, energy absorbed) and likelihood of tanker structure failure (e.g. plastic strains, omega – see Section 3.5.2.1). This type of model used 3D solid elements to ensure accurate prediction of plastic strains. Fluid structure loading was approximated by applying loads using a boundary condition because computer runtimes precluded the option to include the fluid in the model to apply loads as in the FSI type model. The results of the study to understand the sensitivity of tanker structure behaviour to changes in precisely how the fluid loaded the tanker structure mentioned above showed that this approximation was acceptable.

3.5.2.1 Forming limit diagrams and the use of omega values

The use of forming limit diagrams is a widely established method in assessing if sheet metals that are strained in the forming process are likely to structurally fail. If the forming limit has been exceeded, then failure is likely (i.e. the shell or bulkheads have been bent beyond the

point of breaking). The forming limit can be expressed as a ratio of the major principal strain to the major limit strain known as the ‘omega’ value. The variable ‘omega’ (ω) is defined as the ratio of the current major principal strain to the major limit strain determined from the forming limit curve (FLC) and evaluated at the current value of the minor principal strain. Anything above an omega value of 1 means that failure is likely; and anything below 1 means that failure is not likely. Failure becomes more likely the higher the omega value is above 1. Omega is thus a useful measure of how close or how far the strain conditions are to or beyond the forming limit (see Figure 3-8).

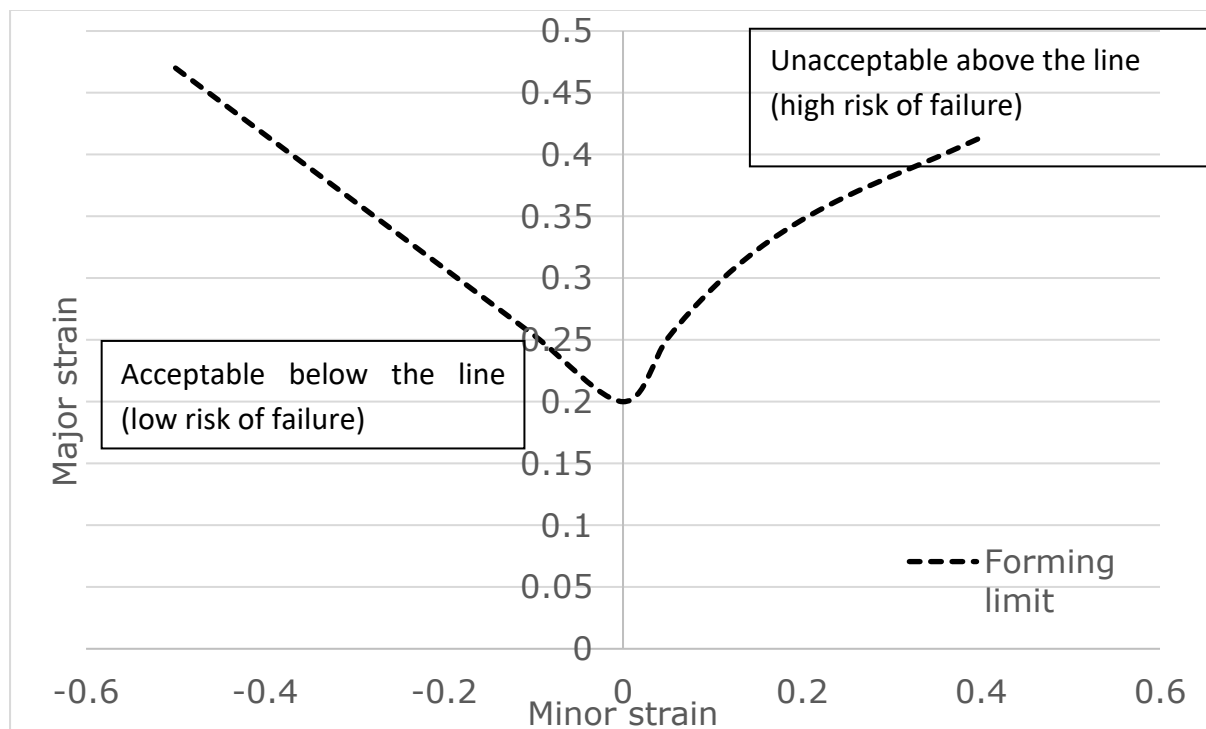


Figure 3-8: Forming limit curve ($\omega = 1$) used for current study

The omega values in this work were calculated from the forming limit diagram used in TWI report 25272/1/16 “Department for Transport Technical Assessment of Petroleum Tankers: Assessment of BS EN 13094 Lap and Partition Joint Designs”⁷ and represent a lower bound estimate for EN AW-5182 aluminium.

The forming limit diagram will also depend on temperature, thickness and strain-rate. Therefore, a lower bound representation was chosen amongst many published experimental findings. This means that the interpretation of the results is slightly conservative. If the response predictions are below the forming limit curve ($\omega = 1$), so it is not predicted to fail, then this would most likely be the case in reality (high probability). If

⁷ TWI (2016): Report 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/government/uploads/system/uploads/attachment_data/file/940245/Technical-assessment-of-BS-EN-13094.pdf accessed August 2023.

the prediction slightly exceeds the forming limit curve, then failure could be a potential outcome. However, due to the conservative nature of the approach, it is still possible that an omega value slightly above 1 may not lead to tanker failure in practice.

3.5.3 Parametric study design

Three parts to a tanker drop test are considered in the modelling:

- the ‘action’ (impact energy due to the drop height, size and mass of the tanker);
- the ‘reaction’ (forces developed by the ground reaction and pressure in the fluid due to the impact); and
- the structural response to the reaction forces (e.g. energy absorbed, deflection etc. of the components).

To predict how the subsections perform across a range of drop tests, six different test parameters were considered, as follows:

1.	compartment length	(numeric factor)
2.	fill level	(numeric factor)
3.	liquid density	(numeric factor)
4.	impact velocity	(numeric factor)
5.	number of compartments	(categoric factor)
6.	design – two banded designs, one stuffed	(categoric factor)

The first four test parameters above are referred to as numeric factors as they could take an infinite number of values. The fifth and sixth parameters are “categoric” in that they can only have a limited number of integer values or categories.

The study involved four steps, with each step using a different combination of the two categoric factors.

Step 1 – model a 2 band/1 compartment banded type 1 subsection.

Step 2 – model a 3 band/2 compartment banded type 1 subsection.

Step 3 – model a 1 compartment banded type 2 subsection.

Step 4 – model 1 compartment stuffed and 3 compartment banded type 1 subsections.

From previous work in this project, it was found that, in a drop test, the impact energy that is absorbed and dissipated is mainly due to plastic deformation of the dishes. However, there is a different ratio of dishes to compartments between the subsections and full-size tankers. A complete tanker has 8 to 10 dishes with 6 compartments (dish to compartment ratios of 1.33:1 to 1.66:1), whereas a single compartment subsection, for example, has 2 dishes per compartment (dish to compartment ratio of 2:1).

If a full-size tanker and a subsection tanker were dropped from the same height (with the same fill level giving the same impact energy per unit length), then the higher dish-to-compartment ratio of the subsection means that the energy absorbed per dish of the

subsection will be lower. Therefore, the number of compartments influences the energy absorbed per dish.

3.5.3.1 *Modelling matrix*

The values used for each test parameter (numeric factor) are shown below:

- Compartment length – 1.85, 2.93 and 4.00 m
- Liquid density – 0.75, 0.88 and 1.00 kg/l (to represent petrol, diesel and water)
- Liquid fill level – by volume – 0.70, 0.83 and 0.95 of compartment volume (for equivalent mass)
- Impact velocity – 4, 6 and 8 m/s (to achieve realistic energy absorption per dish)

These were combined into a series of up to 25 model-runs in each step, each with a unique combination of the four test parameters above and covering both mid-range and various combinations of upper and lower limits across the four parameters. It was anticipated that the number of required model runs would fall as work progressed through the four steps, reflecting emerging findings about the specific regression strengths and response surface functions indicating the individual significance of each parameter.

3.5.3.2 *Structural response parameters*

The impact of the tanker on the ground during a drop test creates two main reactions: a 'ground reaction force'; and forces arising from pressure in the liquid. The pressure in the liquid will depend on the liquid depth and the deceleration on impact. This pressure will act to force the bulkhead outwards and is referred to hereafter as the pressure force.

If these two reactions are similar between models of the same tanker design but with different numbers of compartments, then the structural response (and likelihood of failure) should also be similar.

The four structural response parameters outputted from each run of the model were:

- deflection (amount of crush)
- plastic work done (energy absorbed)
- plastic strain
- forming limit factor (the 'omega value')

3.5.4 *Results of parametric study*

3.5.4.1 *Step 1 modelling results*

For the deflection results, TWI obtained excellent correlation in the regression analysis by finding quadratic equations that generated the response surfaces. The analysis showed that the only numeric factors that were needed were length and drop height; and impact energy could be controlled by adjusting the drop height. This meant that the other numeric factors (fill level and fill volume) could be ignored as they were found to be variations of impact

energy alone (i.e. they had no additional effect on the tanker response). The response surface for deflection as a function of the numeric factors of impact energy and compartment length had a coefficient of determination (R^2) of 0.999, indicating almost perfect correlation.

For plastic work in the bulkhead, a very strong correlation against length and impact energy was also found in the regression analysis with an R^2 of 0.989.

Use of 2D shell elements was found to limit the ability of the fluid structure interaction (FSI) type model to properly predict plastic strains and omega values. This was a consequence of the strain singularity that was developing at the area of high strain in the dish. The local stresses (or more importantly strains) from the 2D shell model were not valid as they did not represent the true three-dimensional strain state.

To overcome this limitation, a different type of model which used 3D solid elements was deployed. Two model runs, using coupling (pressure) forces varied by 30%, showed negligible difference in omega values, indicating that omega is not sensitive to variations in pressure force. In both runs omega reached a value of 1 (indicating the forming limit had been reached) at a deflection of around 40 mm.

To assess if the response from this single-compartment subsection model could demonstrate an equivalent level of damage to the benchmark topple model (also based on the banded type 1 design), the results from the subsection predictions were compared with the benchmark topple predictions.

Benchmark petrol topple values from the eight-banded type 1 tanker were as follows (for the rear-end bulkhead Band H):

- Deflection: 148 mm
- Plastic work: 16.2 kJ
- Coupling (pressure) force: 156 kN

These values were set as targets in the regression software to find the best numeric factors (length and impact energy) for this single compartment model to generate the closest match.

Matching just the deflection and the plastic work responses could be achieved with a range of different lengths and impact energies but to match the pressure force, too, only a compartment length of 3.8 m with an impact energy of 100 kJ was found to be suitable.

For a water load and medium fill level (0.83), this would dictate a single 3.8 m long compartment (of the banded type 1 design modelled) weighing 14 tonnes be dropped from a height of 730 mm. Shorter, lighter subsections could be dropped from a greater height for the same deflection and plastic work. Pressure forces are likely to be slightly higher for a shorter compartment or a higher drop height, but this would be unlikely to have a significant effect on omega within the range currently modelled.

3.5.4.2 *Step 2 modelling results*

Modelling of the two compartment banded type 1 subsection was performed at single fixed values of liquid density (1.00 kg/l) and fill level (0.95), reflecting the step 1 finding that these

parameters had no effect not already captured by varying impact energy alone. Compartment lengths were varied from 1.4 – 3 m (with both compartments always of equal length) and impact energies from 100 – 400 kJ, across 13 separate model runs.

Good correlation was found between deflection and impact energy, and also plastic work and impact energy. One factor correlation (just considering impact velocity) gave a coefficient of determination of $R^2 > 0.97$; and two factor correlation (adding the length factor) improved the fit ($R^2 > 0.99$), indicating very strong correlation.

The pressure force was found to depend mainly on impact velocity with only a small effect caused by variation in the compartment length.

Benchmark petrol topple values from the eight-banded type 1 tanker were as follows (for the rear-end bulkhead Band H):

- Deflection: 148 mm
- Plastic work: 16.2 kJ
- Coupling (pressure) force: 156 kN

These values were set as targets in the regression software to find the best numeric factors (length and impact energy) for this two-compartment model to generate the closest match.

The closest match to all three parameters was obtained with a combination of a 2.5 m compartment length (5 m overall length), an impact velocity of 3.8 m/s (giving an impact energy of 150 kJ). Comparing these results with a single compartment, this gives a longer overall length (3.8 m increased to 5 m); higher mass (14 tonnes increased to 21 tonnes) and the same drop height (730 mm).

The best-match impact energy per partition (~50 kJ/partition) was very similar for both one and two-compartment models. The compartments for both models are longer than the longest compartment in the full banded type 1 tanker, and the impact velocity is slightly lower than in the full-scale topple test.

3.5.4.3 *Step 3 modelling results*

A one compartment subsection model of a banded type 2 tanker, utilising a different joint design to the type 1 tanker, was run at fixed values of liquid density (1.00 kg/l) and fill level (0.83). Compartment length was varied from 1.85 – 4.00 m and impact velocity from 3.0 – 6.0 m/s across 17 separate model runs.

Strong correlations were again found between deflection and impact energy ($R^2 > 0.97$), and plastic work and impact energy ($R^2 > 0.96$), both with little effect from compartment length variations. Both impact energy and length were found to effect pressure force.

3.5.4.4 *Comparisons of banded type 1 and type 2 designs*

Comparing across the results from steps 1-2 (banded type 1 design) and step 3 (banded type 2), the modelling firmly pointed to the type 2 joint design being stiffer than the type 1 joint – requiring more strain energy (work done) to achieve the same deflection. For a given value of energy per partition, deflections from the type 2 model were lower than for type 1

(typically by about 30%) and plastic work done in the end dish was also lower (by about 20%). Pressure forces for the type 2 design were higher than for type 1, again reflecting its higher joint stiffness.

Omega values for the banded type 2 design were lower than the type 1 design, by some 30%, at given values of deflection or impact energy per partition. In both cases, omega values were found to plateau at values of energy per partition likely to be broadly representative of a real-world topple impact (30 – 70 kJ/partition) – at just above or below 1.0 for the type 2 design and at about 1.5 – 1.6 for type 1, depending on the pressure force.

These model predictions were shown to agree well with measurements taken from full-scale topple tests of each tanker design in 2015.

3.5.4.5 Step 4 modelling results

This step involved two separate versions of the model, one a three-compartment subsection model of the banded type 1 design, the other assessing a stuffed design. Six runs of a three compartment banded type 1 model were completed, with two compartment lengths (all compartments either 1.85 or 2.93 m long) and impact velocities varied from 1.0 – 4.5 m/s.

At given values of energy per partition, deflection values predicted from this three-compartment subsection model fitted very well to those from both the one and two compartment models (and those from the complete tanker models). Combining all the data points from all versions of the banded type 1 tank design gave a good correlation between deflection and energy per partition ($R^2 = 0.94$). Achieving an R^2 score greater than 0.9 when considering only a single variable (impact energy) highlights the importance of impact energy as the key factor determining deflection.

A further 12 runs were completed on a model of a one compartment stuffed tanker subsection design. Fluid density and fill volume were again fixed (at 1 kg/l and 0.83 respectively). Compartment length was varied from 1.77 – 4.00 m and impact velocity was varied from 2.0 – 6.0 m/s.

Good correlation was once again found between deflection and energy per partition ($R^2 = 0.95$). Deflections predicted for this stuffed design were also consistently higher than those from either of the banded designs modelled in steps 1-3 at given values of energy per partition – by typically about 50% compared to the type 2 and about 10% when compared to type 1.

Conversely, omega values for the stuffed design were consistently lower than even the lowest banded design (type 2) at any given values of deflection or energy per partition. At around 30 kJ per partition, for example, omega for the stuffed design was predicted to be about 0.8, compared to 0.95 for banded type 2 and 1.45 for banded type 1, at the same pressure force.

3.5.5 Overall conclusions from parametric study

The need to use two types of models (fluid structure interaction (FSI) and detailed structural with 3D solid elements) in the modelling demonstrates clearly that construction of FE

models capable of predicting whether failure is likely to occur in a topple or drop type event is quite a difficult and involved process which requires a high level of expertise.

Modelling of a banded type 1 tanker with one, two and three compartments in a single subsection showed that the important test parameter was the impact energy per partition, rather than just the impact energy. With energy per partition standardised, impact responses followed the same trends for one, two and three compartment subsection models, and also for two complete banded type 1 tanker models with eight and ten bands.

There was good correlation between deflection and impact energy per partition for each of the three tanker designs assessed (two banded, one stuffed). The response of the stuffed joint design followed a similar trend to the banded designs. All three designs showed a power-law relationship between the deflection and impact energy per partition with strong correlations ($R^2 > 0.9$).

For the metallic tankers assessed, this study has identified strong relationships (large coefficient of determination R^2) between impact energy and structural response parameters for tanker drop models. This shows that it should be possible to replicate structural responses observed in full-scale topple tests in a subsection drop test, with appropriate impact energy (which is directly related to compartment length, fill volume, liquid density and drop height).

The deflection of the banded type 2 design was lower than for the banded type 1 design whilst the deflection of the stuffed design was slightly higher than the banded type 1 design, for a given impact energy per partition.

For the banded designs, omega values were fairly constant within the range of 30 kJ to 70 kJ (per partition), which is the range of interest as this is comparable to the energy absorbed per partition in a full-scale topple test. However, there is a clear difference between the designs, with the banded type 1 design having a significantly higher value of omega, indicating a significantly higher risk of failure. These results suggest that, at least for these banded designs, a subsection drop test could differentiate between 'low risk' and 'high risk' designs across a wide range of impact energies based on a comparison of the predicted omega values for each design.

Omega values were lowest for the stuffed design and showed little change in value at impact energies of around 30 kJ to 70 kJ/partition (much like the banded tankers). These results agreed with previous separate research by TWI (confirming lower omega values for the banded type 2 and stuffed design).

3.6 Detailed comparisons of modelling outputs and physical test results

To fully detail the steps that are needed to deliver a subsection test method for rollover, further analysis of existing data was required. A key aim of this was to develop a simple method to explain why there is a variation in deflection at the bands along the length of the tanker in the models of drop and physical topple tests. The parametric study described above, which modelled drop test impacts on various tanker subsections, had firmly pointed to a strong correlation between deflection and energy per partition. To explain differences in deflections along the tankers' length witnessed in the full-scale topple tests carried out for the 2015 research, further work was commissioned to build a deeper understanding of

how the overall energy of impact is distributed (unevenly) across the bands and how a performance-based test method could account for such differences.

Full details are presented in Appendix C. A summary of the key findings is given in the following sections.

3.6.1 *How deflection varied along a tanker's length in full-scale topple tests*

The tankers used in the physical topple tests for the 2015 research were laser scanned before and after testing during that project. These scans were used to assess the deformation along the length of the tanker due to the topple impact. Two banded type 1 tankers were toppled, and one banded type 2. Other differences between the type 1 and type 2 tankers were that the type 2 tanker had 10 bands, which included a swept front end, compared to 8 bands for the type 1 tankers which had no swept front end. All three of these tankers were filled with water to a mass equivalent to the mass of a full load of petrol. As water is denser than petrol, the fill level was approximately 70% for water (petrol fill level is around 95%).

The deformation results in terms of the length of the flat created by the impact at each partition, compared with how far the partition was from the tank rear, are shown for the three tankers in the figure that follows.

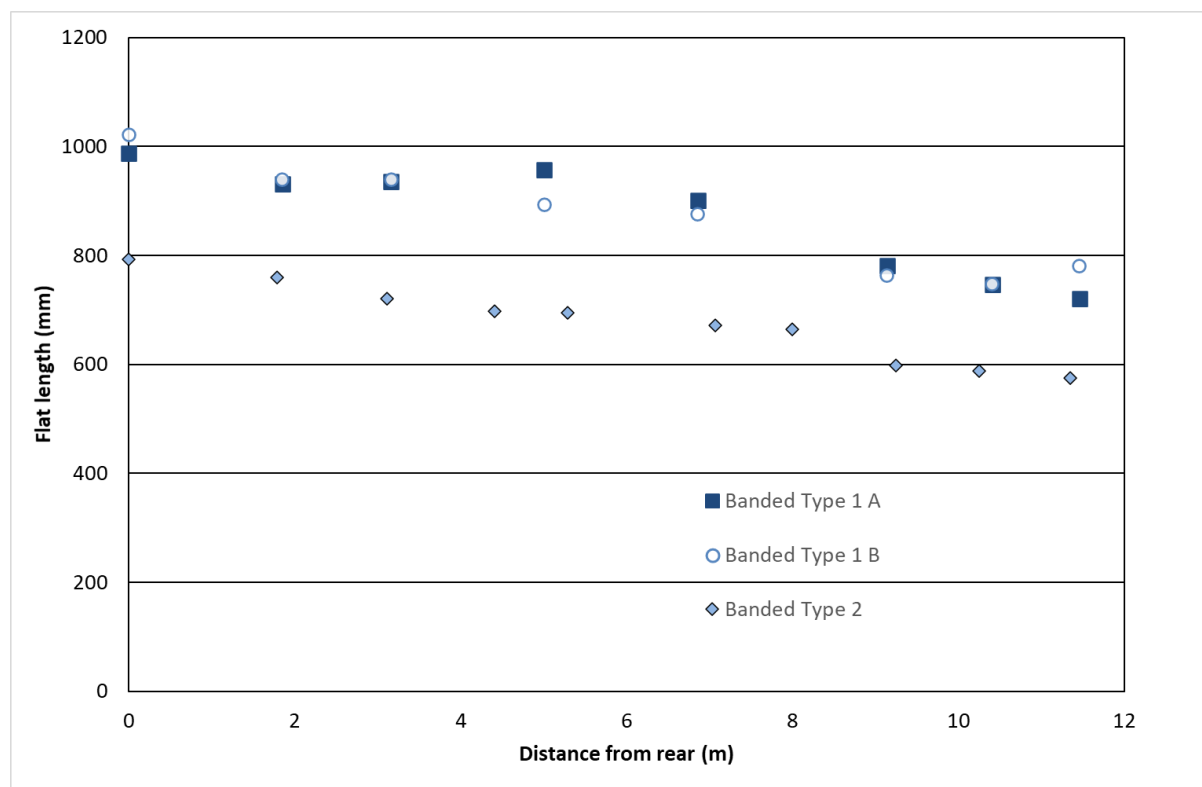


Figure 3-9: Results for the physical topple tests in terms of flat length at each partition

All three tankers showed higher levels of deformation at the rear of the tanker than the front. There is some variation between the results from the two, banded type 1 tankers although they were ostensibly of the same design.

The banded type 2 tanker had a significantly lower amount of deformation than either of the banded type 1 tankers. This is partly due to the addition of two surge plates (10-banded rather than the 8-banded type 1 tankers) and partly due to the stiffer joint design profile.

For the original models of the banded type 1 tanker, the flat lengths were also evaluated, and good agreement was found between the topple measurements and model predictions.

3.6.2 *Relating flat lengths to radial deflection*

For the new modelling work, the deflection of the centre point of the impacted side, relative to the unimpacted side, was used as the main measure of deformation, as this could be calculated quickly and easily from the model outputs. This “radial deflection” effectively measured the amount of crush across the width of the tanker. As the original laser scan data for all the partitions of the test tankers was not readily available, an estimate of the radial deflection at each band was generated from the measured flat length data from the topple tests and the radius of curvature of the sides of the tanker. For the front and rear bands, actual radial deflections had also been measured, allowing the validity of the estimates based only on flat length and tanker radius to be assessed at these points.

While reasonable agreement was found for the banded type 2 tanker, the actual test measurements of radial deflection were found to be substantially below the calculated values for the type 1 tankers tested. Various possible reasons for this discrepancy have been postulated (see Appendix C), mostly associated with reliance on over-simplified assumptions as a basis for the geometric calculation model or those arising from measurement inaccuracies for the toppled tankers. These factors would also tend to have greater effect for larger deformations, thus also explaining to some extent why the discrepancies were higher for the banded type 1 tankers than the type 2.

However, the main purpose of the assessment of the experimental data was to observe the deflection response trends along the length of the tankers. In this respect, there was good agreement between measured and modelled deflections with two observations confirmed: highest deflections occur at the rear of the tanker; and deflections increase at the front of the tanker where the compartments are higher above the kingpin (although the deflections are still lower than at the rear). The ratio between the max deflection (estimated from flat lengths or modelled) and the corresponding average deflections (across all bands/partitions) was found to be consistently in the range 1.15 – 1.24.

3.6.3 *Identification of the most vulnerable band/partition*

Topple tests on the two banded type 1 tankers had resulted in failures at the rear dish, while the banded type 2 tanker tested failed at the front dish. For a performance-based assessment based on a tanker subsection, it is vital that the subsection chosen adequately replicates the performance of the most vulnerable part of the complete tanker design.

The challenge investigated and described here was to develop a methodology to identify the most vulnerable part of a full tanker (with a metallic structure but of otherwise novel design) without access to a full-scale test or a complete FE model.

In an (ultimately unsuccessful) attempt to overcome this challenge, a methodology was developed based on the “Impulse Moments Principle”, which considers the moments about

the centre of gravity (CoG) of the tanker during the impact, and the rotation about the pivot line during a topple.

This methodology, in essence, replaces the balancing of forces and their moments acting clockwise and anticlockwise in the static case with the balancing of momentum and impulses acting on the tanker in the dynamic impact case.

Generally, the method was useful in helping to explain some of the variations observed with regards to deflection along the length of various tanks (simulated and physical measurements). The method helped identify some underlying relationships connecting spans between stiffening elements and mass distribution.

However, testing the method with one of banded tanker designs showed that a good match was difficult to obtain. In particular, the ratio of peak deflection to average deflection was sensitive to the overall energy/impulse.

It is likely that the method could be advanced and further developed using principles from solid mechanics. For example, Hertzian contact theory (cylinder to plate) could be used as a starting point and then Hertzian contact mechanics coupled with some iterative calculations (with respect to plastic work / plastic energy dissipation) could be integrated until the initial kinetic energy had been dissipated. However, any analytical method founded strongly in solid mechanics would get very complicated very quickly as one moved from idealized geometries to more realistic ones.

3.7 Penetration and abrasion testing

The approach taken to develop potential outline test methods for assessment of the resistance of a tank to penetration and abrasion in a rollover incident was to:

- Identify potential test method candidates from general review of test methods used for the assessment of penetration and abrasion, with focus on regulatory test procedures and testing of coupon type samples.
- Assess potential candidate test methods, on the basis of their advantages and disadvantages in terms of technical feasibility, practicality, and cost, to produce a short list for further assessment and development.
- For each short-listed candidate outline potential work required to develop test method further.

3.7.1 Penetration

From a general review of test methods, a test method within EN 13094 Annex B for the assessment of the specific resilience (i.e. resistance to penetration) of tanker shell material was highlighted. Specific resilience is defined 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.

The requirement is that non-standard tank shells (see clause 6.8.2.2(c)), i.e. those with a non-circular or elliptical cross-section, shall have a specific resilience 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

On the basis that this test method was appropriate and already incorporated in EN 13094, it was identified as the ideal candidate for further assessment and development.

Further development work could include the establishment of one or more performance limits in terms of specific resilience value(s), which would be indicative that the tank would offer equivalent safety to an ADR compliant one. This could be achieved by testing samples of shell material from current ADR compliant tankers and of reference mild steel as specified in EN 13094 and detailed above.

3.7.2 Abrasion

During a typical tanker rollover accident, the tanker will slide along the ground after rolling onto its side. This sliding action causes the side of the tanker to abrade as it passes along the ground. The review and analysis of accident data performed in an early part of the project (see Section 2.1) identified that abrasion of the tank shell may be a factor causing tanks to leak in rollover accidents, though possibly only at higher speeds. This suggests that the existing design requirements in ADR regulations result in materials with sufficient thickness to achieve a suitable level of abrasion resistance. If an alternative approval process did not account for this, there may be a risk that designs emerged that could pass the test but offer lesser performance in this respect than current tank designs.

It was found that there were no abrasion requirements in EN 13094. A review of a wide range of test methods which contained elements of abrasion resistance testing was carried out. The review took into account that an appropriate abrasion test should be able to:

- abrade a sample in a reliable, repeatable way;
- control the contact force between sample and abrasion surface so that it is repeatable;
- be representative of sliding during a rollover tanker accident; and
- assess the loss of material to ensure that this is not too great when compared with the loss of material for an ADR-compliant sample.

The test methods reviewed included:

- UN Regulation No. 22 – abrasion resistance tests for helmets
- Standard abrasion tests which include:
 - Blade-on-block testing

- Taber abrasion testing
- Pin-on-Disk wear testing
- Block-on-disk abrasive wear test
- Grinding wheel method
- Standard grinding and polishing based methods
- Tyre durability testing

The review did not identify any test method that was directly applicable without development of the method and rejected most, mainly on the basis of their technical feasibility. Concept test methods based on a grinding wheel and tyre durability testing type approaches were selected as the test methods most suitable for development into an abrasion resistance test. Outlines of the selected concept test methods are described below in terms of a description of the potential test method, potential issues that need further consideration and work proposed to develop them further.

The review of these test methods and reasons for their rejection is described in Appendix D.

As part of the peer review process, one further potential method was put forward, based on towing a weighted sample behind a vehicle on a test track. This concept test method has the advantages that it could represent high forces on impact by dropping the sample onto the road surface and the abrasive surface could be highly representative of actual roads, but it has the key disadvantage that it is likely to have poor repeatability, so was also not suggested for further consideration.

3.7.2.1 *Concept test method based on grinding wheel type approach*

Description

An outline sketch of one possible way of how to perform an abrasion test using a grinding wheel type method is shown in Figure 3-10.

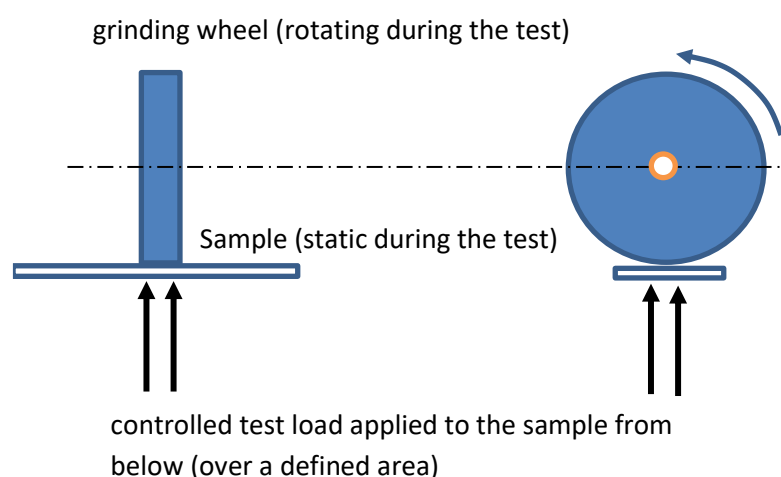


Figure 3-10: Sample in a grinding wheel abrasion test

The grinding wheel rotates to abrade the sample and a controlled load is applied to the sample from below, for example, from a screw jack with a load cell attached.

A jig will need to be designed to keep the sample in approximately the same position throughout the test and resist the lateral loads.

This method will allow the force between grinding wheel and sample to be accurately controlled.

The change in thickness of the test sample due to abrasion will be compared with the change of thickness of a benchmark ADR-compliant tank material, and a pass/fail performance limit will be decided.

Potential issues for consideration

- The grinding wheel could become clogged with filings which alters its abrasive properties. Therefore, the grinding wheel may need to be re-dressed regularly to the same level of abrasiveness.
- The type of grinding wheel to use would need to be specified for the test.
- Note that dressing a grinding wheel reduces its diameter. For a reduced diameter grinding wheel the tangential speed of its circumferential surface is lower. Therefore, for a smaller diameter wheel, it would be as though the tanker was sliding along the ground at a lower speed.
- Therefore, when the grinding wheel diameter has reduced by a certain amount, the wheel may need to be replaced. Increasing the rotational speed of the grinding wheel may be an option, but this may be outside the manufacturer's acceptable operational parameters for the wheel.

Work required to further develop the test method

The following tasks should be considered to further develop the test method. As the method is essentially a bench-top test with commonly available equipment, it is assumed that a manufacturer or test house could acquire the equipment to carry out this test without undue cost.

1. Develop the test method to decide on the most suitable ways to hold and load the sample in a controlled way. This will require a careful risk assessment to assess all the safety issues that are relevant to working with grinding wheels.
2. Discuss the following with grinding wheel suppliers: grit sizes⁸, hardness values, and how often grinding wheels normally need to be re-dressed (e.g. if finer wheels need re-dressing more frequently, then a coarser grade wheel may be more suitable etc.).

⁸ Grit size typically runs from coarse (16 -24 grit), medium (36 - 60 grit) and fine (80-120 grit). Superfine grits run from 150 and higher. Grinding wheels usually will be between 24 and 100 grit. A coarse grit is normally used for fast, aggressive stock removal and finer grits for less stock removal but better surface finish.

3. Discuss with the grinding wheel supplier what the maximum acceptable radial load is that can be applied to the grinding wheel and what speeds the grinding wheel has been designed for.
4. Review grinding wheel options and determine a grinding wheel with a defined grit size to use.
5. Design and build a prototype test rig.
6. In the test method, note that the diameter of the grinding wheel is a significant parameter, and this will reduce as the wheel wears. Therefore, the wheel diameter may need to be used in calculating the number of rotations that should be carried out for the test. This is to ensure that all tests have the 'equivalent' distance of sliding.
7. Carry out tests with several samples from an ADR-compliant tanker to assess the reliability and repeatability of the method.
8. Review the method and advise on any improvements that need to be made. It is likely that a second series of tests on a revised design may be needed before the test method can be finalised.
9. Research work on rollover accidents⁹ has shown that, at impact, the normal forces will be high. Hence the frictional force and associated abrasion rate will also be high during the impact phase compared with the subsequent 'slide-to-rest'. An abrasion test should take this into consideration when determining suitable test loads and a time period for the test.

3.7.2.2 *Concept test method based on tyre durability testing type approach*

Description

The test method is based on adapting a current test machine used for tyre durability testing. Examples of the equipment are shown in Figure 3-11 (a tyre rolling resistance test) and Figure 3-12 (a tyre endurance test).

⁹ Knight I and Dodd M (2020). Performance test procedures for petroleum road fuel tankers – Review and analysis of accident data impact conditions and regulations. Apollo Vehicle Safety. Report for Part A of DfT contract P4/030/0013c

Tire Endurance Test Machine



Figure 3-11: Tyre endurance test machine (photo reproduced by kind permission of www.tmsi-usa.com/51.html#page2 USA)

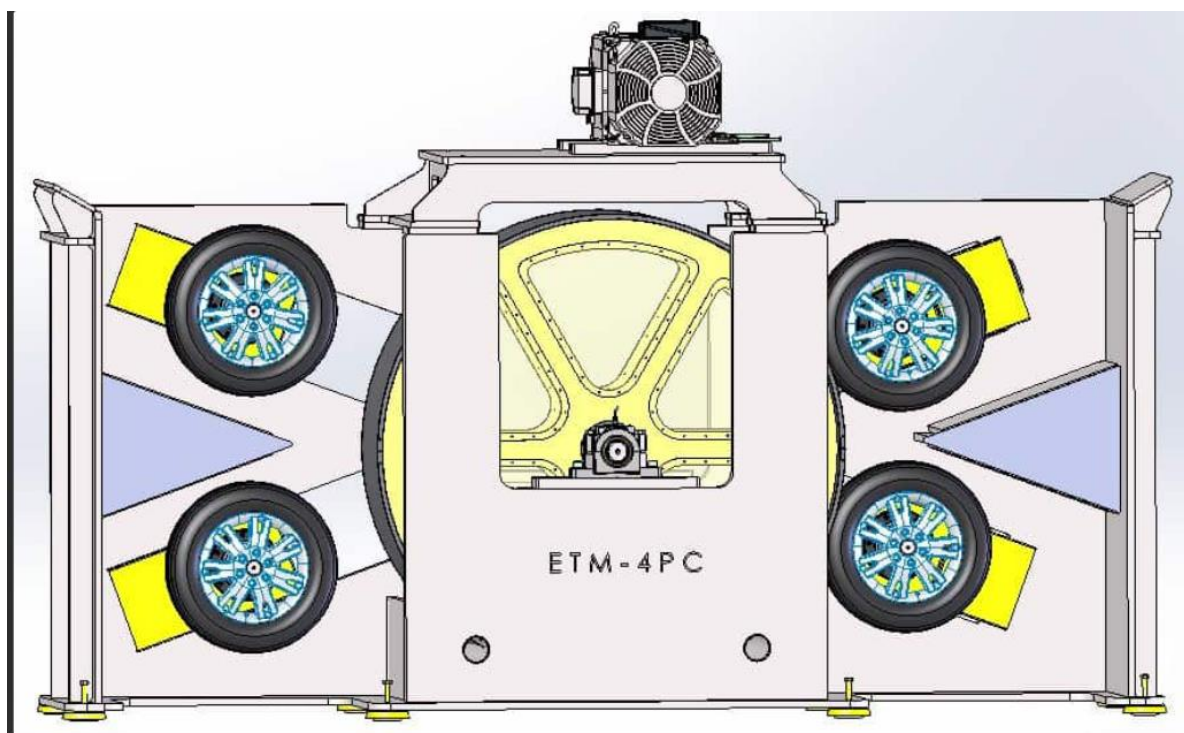


Figure 3-12: Endurance and high speed test machine (photo reproduced by kind permission of www.inmess.de/en/endurance-and-high-speed-testing-machine-etm/ Bremen, Germany)

There are other types of tyre test machines that place the tyre on top of a belt rather than against a drum. A suitable abrasive surface would need to be determined for the drum or belt.

The sample would need to be attached to a jig that can apply the sample to the drum surface with a controlled static force. The drum will rotate to abrade the sample and the type of abrasive surface on the drum will need to be specified.

As mentioned for the grinding wheel method, the change in thickness of the test sample due to abrasion will be compared with the change of thickness of a benchmark ADR-compliant tank material, and a pass/fail performance limit will be decided.

Potential issues for consideration

The abrasive surface may become clogged with filings which alters its abrasive properties. Therefore, the abrasive surface may need to be replaced regularly to the same level of abrasiveness each time. However, as the abrasive surface is that much longer than it is for the grinding wheel method, it should need replacing to a lesser extent.

The tests may need to be booked into specialist facilities (rather than using a test rig that could be developed in-house) and a specialist jig will need to be designed to hold the sample as the tyre mounts will not be suitable to hold the small test samples required. Also, the owners of the facilities (mainly tyre manufacturers) may choose not to make the test facilities available for this work. There may be some concerns about the sparks that are likely to be emitted in a tank shell abrasion test. Therefore, bespoke test facilities may need to be designed from a tyre testing facility specifically for these test purposes.

Work required to further develop the test method

1. Discuss with tyre machine manufacturers and users to assess how practical it is to adapt a machine for an abrasion test (consider both drum and belt machines).
2. Write an early draft test protocol to show the key stages required in the test.
3. Carry out visits to see machines in use and start to put ideas together for how a machine could be adapted, and what would be the most suitable type of machine.
4. Carry out an assessment of how practical this method will be (e.g. will there be sufficient providers of test services in the UK for manufacturers to go to? What effect would it have for tanker manufacturers if a provider of the test service chose to withdraw from the market? etc.) and assess what the potential cost could be to manufacturers (in time and price) if this method was to be used.
5. Research work on rollover accidents¹⁰ has shown that, at impact, the normal forces will be high. Hence the frictional force and associated abrasion rate will also be high

¹⁰ Knight I and Dodd M. (2020) Performance test procedures for petroleum road fuel tankers – Review and analysis of accident data impact conditions and regulations. Apollo Vehicle Safety. Report for Part A of DfT contract P4/030/0013c

during the impact phase compared with the subsequent ‘slide-to-rest’. An abrasion test should take this into consideration when determining suitable test loads and a time period for the test.

It should be noted that, at the moment the preferred method is the grinding wheel method due its simplicity and lower cost whilst not appearing to be any less accurate than the tyre test method.

3.8 Development of outline technical code

The intention from the outset of the research described above was for DfT to be able to take forward its ultimate findings into international standards and the ADR regulatory framework. The most appropriate route to achieve that would be via a Technical Code that could be referenced within that framework and thus be available to manufacturers to use as an alternative to the current design-based requirements.

It is expected that, where the design and construction of metallic gravity-discharge tanks deviates from the current ADR requirements, this code could be used to demonstrate (for approval purposes) that the tank would maintain an equivalent level of performance in a rollover type accident.

This section describes the steps completed as part of this research to start to develop such a Technical Code, specifically as it might relate to subsection drop-test modelling and/or testing. Appendix E provides further details of the calculations involved.

It has not been possible, within the constraints of the current work, to fully define and validate all the necessary requirements, hence for the purposes of this report it is referred to as the “Outline Technical Code”. This Outline Technical Code, developed by HSE SD, is provided in its full current draft form in Appendix G.

The Outline Technical Code incorporates the results of the research to-date on abrasion and penetration testing (described in the preceding section) and those arising from the topple/drop-test modelling. For the topple/drop test requirements, various options for approval are provided including a full-scale topple test option and variations on the drop-test method, two of which combine such testing with finite element modelling results, while others are based on testing only.

3.8.1 Establishing the drop test energy requirement

From the research findings, a procedure was developed for calculating the appropriate drop height to ensure that a drop test on a two-compartment subsection is sufficiently representative of a complete tanker topple. This involved estimating the impact energy, converting between topple and drop cases, and factoring for the number of partitions and variation of deflection along the length.

3.8.1.1 Impact energy of complete tanker topple

The first step in deciding the impact energy for a subsection drop test is to evaluate the impact energy for the scenario being replicated, i.e. a complete tanker topple with a full

load of petrol. To allow for design variations which could alter the rotational velocity at impact (such as a change in the height of the CoG), a method of establishing the impact energy is needed.

For a topple event, where the initial state at the point of topple is stationary (no kinetic energy), all the kinetic energy at impact will arise from the change in potential energy. As the mass stays the same, the only parameter that needs to be evaluated is the change in height of the CoG.

As the topple point will be when the CoG is directly above the pivot line (as a first-order approximation, this is the outer edge of the wheels), the height of the CoG will be approximately equal to the distance of the CoG from the pivot line. The angle of the tipping point does not need to be known.

By separating the tanker into its component parts – shell, fuel load, running gear, partitions etc, the individual component contributions to this overall potential energy can be calculated. For the banded type 1 tanker modelled (and topple tested as part of the 2015 research), these calculations suggest that by far the largest contribution to potential energy is from the petrol load (94%). The tanker shell makes up 5% if the partitions and ends are included and there is a contribution of 1.7% for the other structural components and ancillary equipment.

The remaining parts (supports and running gear) have low centres of gravity, so their contributions are smaller or even negative. A negative contribution arises when the CoG is higher when in the impact position than it is at the topple position.

The approach used here should be accessible for manufacturers when designing a new tanker without the need for complex modelling. CoG information should be available from computer drawings or could be estimated with reasonable accuracy. If the details of the supports or running gear were not known at the time of assessment, these could be ignored for this calculation, as their contribution is small and likely to be negative (although the overall height of the tanker must be known accurately). Therefore, leaving them out of the calculation would be likely to result in an increase in impact energy, and therefore be conservative.

The most difficult component to assess accurately would be the liquid load, partly due to the shape but mainly due to the shape changing as the tanker moves. Using an assumption that the compartments are full of liquid for the purposes of determining the location of the CoG would avoid this problem. The resulting increase in potential energy due to a slightly higher CoG would be small (approximately 5%) if the actual liquid mass was used (although the CoG is calculated based on a completely full compartment, the mass would still represent the actual fill level).

To check this method, the change in potential energy calculated using this approach was compared to model results. As described earlier, a simple model of a single compartment was run from the topple point to the impact point as part of this research and the resulting liquid motion at the point of impact was found to be complex due to movement of the liquid relative to the tanker. The most accurate simplification method was to represent the movement as a combination of a linear motion and a rotation about the centre of the tanker compartments.

Using this motion for the full tanker with a full load of petrol resulted in kinetic energy of 370 kJ at impact. Calculating the change in potential energy using an accurate estimation of the CoG of the petrol resulted in the same 370 kJ figure. This confirms that this approach is valid to assess the impact energy. Using the simplified centre of gravity approach resulted in a change in potential energy of 387 kJ, an increase of approximately 5%.

3.8.1.2 Allowance for differences between topple and drop

The proposed subsection test in the outline technical code will be a vertical drop test rather than a topple. This is likely to be a more practical test method and adjusting the impact energy would be quite straightforward.

However, in a vertical drop, all the initial motion is in the vertical direction, which is stopped by the ground. In a topple, as the tanker and contents are rotating, there is a horizontal component to the motion at the point of impact. Therefore, not all the initial kinetic energy will be absorbed on impact. As a result, for the same initial impact energy, more deformation occurs in a drop test than in a topple.

To account for this, the proportion of the initial potential energy in a topple that remains as kinetic energy after the initial impact with the ground needs to be deducted from the drop-test impact energy. From the complete tanker topple model with a full petrol load, the kinetic energy remaining after impact was approximately 100 kJ, i.e. 27% of the initial impact energy.

Modelling both a (full tanker) topple test and a complete tanker drop test, with the energies made equivalent by applying this 27% correction factor, produced minimum, average and maximum deflections very similar to each other. The adjusted average deflections were also found to align very closely with the line of best fit (a power curve) when impact energy per partition was plotted against deflection for all the subsection modelling results of the parametric study described above.

The adjustment is to allow for the differences between topple test and vertical drop test that arise due to the difference in the direction of the motion. Therefore, the adjustment (of 27%) should be reasonably independent of tanker design. One aspect that might alter this proportion would be a significant change to the height of the CoG. However, large changes are unlikely as the CoG will be kept low for stability reasons, but the ultimate extent to which it can be lowered is limited due to the inevitable need for running gear.

3.8.1.3 Allowance for differences along the length of the tanker

For complete tankers, the deflections tend to be higher at the rear. This trend was observed in the banded type 1 tanker physical tests and models (both topple and drop), and also the physical tests of a banded type 2 tanker and a stuffed design.

Reasons for this are due to differences in the distribution of the mass and stiffening elements (partitions and ends). If all the stiffness and energy absorption characteristics for the partitions are known, then the pattern of deflections may be predicted for a topple or a vertical drop. However, it is unlikely that all the necessary information would be known, and there is some uncertainty about the physics underpinning the calculations.

Therefore, in the outline technical code, for the test option where the approval is based on a subsection drop without the need for any finite element analysis, a standard adjustment is needed to take the variation in deflection along the length into account.

For the benchmark topple model (banded type 1 with a full load of petrol), the maximum deflection at the rear was approximately 25% higher than the average deflection along the whole length, as an upper bound. However, to get 25% higher deflection, a proportionally larger increase in energy is needed because the relationship between energy and deflection is a squared one.

The equation for the power law curve, that fits the data for deflections against impact energy per partition, is approximately:

$$d = 20.3E^{0.5}$$

where d is deflection and E is impact energy per partition. Therefore, to get a 25% increase in deflection, an increase in energy of 56% is needed ($1.25^2 = 1.56$). This factor is thus proposed in the outline technical code to obtain a suitable target value of impact energy per partition for the subsection drop test (if a complete tanker model is not available).

The banded type 2 and stuffed designs had different deflection/energy relationships, but the exponent of the best fit curve was still approximately 0.5. While the 1.56 factor above would seem to be appropriate (at least according to the evidence generated), no other design has been modelled as a complete tanker, so the variation along the length of the tanker is not known for other designs. Also, the data set for banded type 1 tankers is limited. Therefore, an additional uncertainty factor is likely to be appropriate.

3.8.2 Subsection drop test impact energy and drop height (worked example)

For the banded type 1 tanker modelled, when in an upright position, the height of the CoG was 1924 mm when using the actual 95% fill level, or 1963 mm if assuming the compartments were 100% full of petrol. The approximated horizontal distance from the pivot line (outer edge of the wheels) to the CoG was 1279 mm (assuming weight is distributed equally about the longitudinal centre line).

Therefore, the approximate height of the CoG at the tipping point was:

$$h_t = \sqrt{(1924 \text{ mm})^2 + (1279 \text{ mm})^2}$$

$$h_t = 2311 \text{ mm}$$

The total mass of the tanker (including the petrol load and all the supporting structures) was 36,768 kg. At the impact point, h_i , the height of the CoG can be assumed to be half the width of the tanker, which is 1275 mm. Therefore, the change in potential energy is:

$$\Delta PE = (h_t - h_i) \times g \times m$$

$$\Delta PE = (2.311 \text{ m} - 1.275 \text{ m}) \times 9.81 \text{ m}\cdot\text{s}^{-2} \times 36768 \text{ kg}$$

$$\Delta PE = 374 \text{ kJ}$$

This value obtained was slightly higher than the kinetic energy at the point of impact for the benchmark topple model (370 kJ), although the difference is only 1%.

Using the assumption that the CoG of the petrol is the same as the CoG of the compartments (i.e. assuming the compartments are completely full for the purposes of defining the CoG, but not mass), results in the combined CoG of the tanker being 29 mm higher when the tanker is upright. This translates to an increase in potential energy up to 385 kJ, an increase of 3% compared to the 95% fill level calculation.

For the purposes of this illustration, the kinetic energy at impact for the benchmark model will be used for further calculations (370 kJ). This is so that a true comparison of the subsection drop model can be made against the benchmark topple. It is acknowledged that using the above potential energy calculation is likely to result in kinetic energy values a few percent higher.

Next, the kinetic energy for the topple is reduced by 27% to take into account the differences between a topple test with rotation, and a straight vertical drop test.

Topple to drop adjustment:

$$\begin{aligned} KE_{Drop} &= 0.73 \times KE_{Topple} \\ KE_{Drop} &= 0.73 \times 370 \text{ kJ} \\ KE_{Drop} &= 270 \text{ kJ} \end{aligned}$$

The impact energy per partition is then calculated, using the number of partitions/ends in the complete tanker being assessed. Stiffening elements that are significantly different to the standard partitions, such as stiffening rings, should not be included. Surge plates, provided they are of a similar design to partitions, should be included provided that the hole in them does not exceed a certain proportion of the cross-sectional area (30% of the area is currently suggested). The number of partitions, n_{tot} , in the benchmark model is 8 (including the ends and a surge plate in the front compartment).

$$\begin{aligned} KE_{part} &= \frac{KE_{Drop}}{n_{tot}} \\ KE_{part} &= \frac{270 \text{ kJ}}{8} \\ KE_{part} &= 33.8 \text{ kJ} \end{aligned}$$

The adjustment for variation in deflection along the length must then be made. For the benchmark topple, the rear end of the tanker deflected by approximately 25% more than the average of all partitions. To get the change in energy required, this adjustment should be squared.

$$\begin{aligned} KE_{partAdj} &= KE_{part} \times 1.25^2 \\ KE_{partAdj} &= 33.8 \text{ kJ} \times 1.56 \\ KE_{partAdj} &= 52.7 \text{ kJ} \end{aligned}$$

If the variation in deflection along the length is assessed for the actual tanker using the finite element method, the 25% value can be replaced with the value obtained from the assessment. However, the new value should still be squared.

The subsection to test will be performed on the rear two compartments unless modelling suggests some other compartment will have a higher deflection. The number of

partitions/ends in the subsection to be tested, n_{sub} , is therefore likely to be 3, unless either of the compartments being tested would normally contain a surge plate. These should only be included if they were included in the total number of partitions for the complete tanker. Any stiffening elements not counted should not be included.

Total impact energy for test subsection:

$$KE_{Test} = KE_{PartAdj} \times n_{sub}$$

$$KE_{Test} = 52.7 \text{ kJ} \times 3$$

$$KE_{Test} = 158 \text{ kJ}$$

The mass of subsection with a water fill to 95% capacity, m_{sub} , was 13,702 kg.

Drop height, H , can then be calculated from

$$H = KE_{test} / (m_{sub} \times g)$$

$$H = 158 \times 10^3 / (13702 \times 9.81 \text{ m/s}^2)$$

$$H = 1.17 \text{ m}$$

To summarize, following the above methodology of the outline technical code for the banded type 1 tanker would dictate a subsection drop test of the two rearmost compartments loaded with water to a fill level of 95% and dropped from a height of 1.17 metres. The resulting deflections and risk of tank rupture would, this research suggests, be very similar to such a tanker experiencing a simple real-world topple when laden with petrol. The following section describes some further modelling carried out to help verify this claim.

3.8.3 Verification of drop-test methodology in outline technical code

The drop-test modelling described above, where more than one compartment was assessed, had always assumed compartments of equal length. The aim of this part of the research was to model a subsection representing the rear two compartments of a real tanker, each of a different length, in a way that would match the proposed subsection drop test defined by the outline technical code.

The geometry for the subsection model was based on the rear two compartments of the banded type 1 tanker. The compartment lengths were 1.85 m for the rearmost (number 6) and 1.31 m for the second-to-last compartment (number 5). In this subsection, both of the end dishes are orientated so that they bow-out of the compartments (convex from the outside), as would be the case in the full-scale tanker (where the “end dish” of the number 5 compartment would be a partition separating it from compartment number 4).

The parameters used for the model of the drop test were as follows (and as per the preceding worked example application of the outline technical code):

- The fill level used was 95%. This gave the maximum mass for the size of the compartments tested, meaning that the drop height could be kept to a minimum while achieving the required impact energy.

- The liquid used was water with a density of 1000 kg/m^3 . Use of any other liquid would be either prohibitively expensive or have safety and/or environmental concerns.
- The impact speed was 4.8 m/s , equivalent to a drop from 1.17 m and with a total mass of $13,702 \text{ kg}$ giving an average impact energy of 52.7 kJ per partition (three partitions).

3.8.3.1 *Results of code-compliant subsection drop test modelling*

Unlike the previous two-compartment (of equal length) subsection models, there was in this case a variation in deflection values between the two ends. The target was to achieve deflections in this subsection model that match the maximum deflection in the benchmark petrol topple model (145 mm). The average deflection value from this subsection model was 141 mm , with maximum and minimum values of 149 mm and 133 mm . Therefore, while the average deflection was slightly below the benchmark topple test value (by 2.8%), the maximum deflection was slightly above the target value (2.3%).

The above results are based on the impact energy of the benchmark topple model (370 kJ). Calculating the energy for the subsection drop test based on change of potential energy and a simplified CoG for the petrol would result in the impact energy for the drop test being about 4% higher than the value used. With the deflection being proportional to the square root of the impact energy, this would be likely to result in a deflection about 2% higher, bringing the average deflection very close to the target value. However, the highest deflection would then be about 5% higher than the target value. Given the insensitivity of omega value to variations in deflection over the range of energies per partition of interest, however, a 5% over-target value for maximum deflection is unlikely to have a dramatic effect on the omega value calculated and, therefore, the likelihood of structural failure.

3.9 Stakeholder input

3.9.1 *Tanker Manufacturers*

Two meetings were held: one with a manufacturer of banded design tankers, and another with a manufacturer of stuffed design tankers, and the various options outlined in the technical code were explained to them.

Manufacturers said that they would need to think about how this could improve their offering to customers, but both manufacturers felt that the full-scale topple option (option 1) was likely to be too expensive. Neither manufacturer has strong in-house capability for finite element modelling, but they both have access to partners that can supply capability if needed.

In general, manufacturers stated that they are producing legacy designs with the occasional small innovation with respect to the manufacturing process rather than new and different products.

3.9.2 *Appointed Inspection Bodies (AIBs)*

Two meetings were held, each with a single representative from two different AIBs. The views of the inspection bodies are described here. The first was that the design of tankers in the UK is driven by customer requirements and vehicle gross weight limits. This results in two main types of tanker: semi-trailers; and rigid tankers (smaller size).

Second, manufacturers' design decisions will be based on considerations of costs and benefits from their point of view and their customers. They tend to design based on historical knowledge and the rules within EN 13094 and ADR with small innovations where needed.

Third, another major design consideration is fatigue failure between the tank and chassis. The tank is quite a stiff structure, but the vehicle chassis is more flexible. The position and design of the connections between the two is important to reduce the likelihood of fatigue failure of these connections.

Fourth, in the longer term, the market will probably decline because of the move away from fossil fuels for vehicles; so the number of deliveries to petrol stations, which are mainly carried out by semi-trailer fuel tankers, will reduce.

Finally, the AIBs stated that more freedom to change the geometry of tanks, for example, to make them more box shaped could enable vehicles to carry more product per unit length, but the degree to which these changes can be made is limited by various factors including the vehicle gross and axle weight requirements.

3.10 **Discussion and summary of findings (rollover)**

The preceding sections have presented a detailed picture of the research carried out regarding the potential development of performance-based alternative requirements for rollover safety. This has included a review of candidate physical test methods, modelling of whole tanks and subsections under topple and/or drop-test conditions, consideration of abrasion and penetration issues, and the development of an outline technical code as a first step towards integrating performance-based requirements into the ADR framework.

This section summarises the key research findings, as they pertain to the rollover scenario, discusses some important limitations of the work performed to-date and makes suggestions for potential further work to address them.

3.10.1 *Key findings*

Having identified drop-testing of tanker subsections as being the likely most practical alternative to whole tanker topple testing, detailed modelling has shown very strong correlation between impact energy per band or per partition and deflection. When normalising energy by the number of partitions, impact responses followed the same trends for one, two and three compartment subsection models, and for the complete banded type 1 tanker models with eight and ten bands.

There was good correlation between deflection and impact energy per partition for each of the three tanker designs assessed (two banded, one stuffed). The response of the stuffed joint design followed a similar trend to the banded designs. All three designs showed a

power-law relationship between the deflection and impact energy per partition with strong correlations.

For the metallic tankers assessed, this study has thus identified strong relationships (large coefficient of determination R^2) between impact energy and structural response parameters for tanker drop models. This shows that it should be possible to replicate structural responses observed in full-scale topple tests in a subsection drop test, with appropriate impact energy (which is directly related to compartment length, fill volume, liquid density and drop height).

The use of forming limit diagrams is a widely established method for the analysis of the sheet metal forming process, as it defines failure criteria - if the forming limit has been exceeded, then failure is likely (i.e. the shell or bulkheads have been bent and strained beyond the point of material structural failure). 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. Failure becomes more likely the higher the omega value is above 1. Omega is thus a useful measure of how close the strain conditions are to or beyond the forming limit and the research has demonstrated its potential applicability to assessing the risk of structural failure in a tanker topple and/or subsection drop-test scenario.

The deflections and likelihood of major loss of containment experienced by tankers in real-world rollover scenarios can be replicated in a suitably specified, two-compartment subsection drop-test (or a full-scale physical topple test) supplemented by abrasion and penetration tests.

Drop-testing of the two rearmost compartments is likely to be most appropriate, as those are most susceptible to structural failure in a topple of conventional tanker designs, but applying a safety factor to the tested impact energy per partition can mitigate risks that other compartments may be more susceptible in a novel tanker design.

In a rollover scenario, loss of containment can arise via any one of three mechanisms; the initial impact with the ground, abrasion of the outer shell structure as the tanker then slides along the road surface, or penetration of a hard object through the tank structure during this sliding phase, after a subsequent additional impact or indeed during the initial impact after toppling. Any novel design of tank should be verified as having no lower risk of containment loss than any conventional design (as defined by the current ADR requirements or range of designs available on the market) under all three conditions.

Resistance to penetration requirements already exist in ADR via the referenced standard EN 13094 and can readily be adopted and/or adapted for approving novel designs.

Various existing test methods potentially relevant to abrasion testing of novel tanker designs were assessed but none was found to be directly applicable without further development. Concept test methods based on a grinding wheel and/or tyre durability testing were selected as the test methods most likely to be suitable for further development into an abrasion resistance test for tanker rollover scenarios.

3.10.2 *Research limitations*

The findings above are based on somewhat simplified and idealised representations of only two designs of banded tanker and one generic design of a stuffed tanker. They may not represent the full range of actual real-world designs.

All three designs modelled represent tank structures that have been approved under ADR requirements. No “novel designs” have been assessed and thus extrapolating the results from compliant designs to novel designs involves inherent risks that some important safety aspects of the novel design remaining un-considered.

The results obtained should, with the above caveat, be applicable to novel metallic tank designs but non-metallic structures have not been investigated.

Subsection drop-tests are considered to be the most cost-effective option (instead of full-scale topple tests), but the likely savings have not been quantified.

3.10.3 *Discussion of strategic review and possible further development of the Outline Technical Code*

It is reasonable to assume, based on some informal discussions, that a number of manufacturers may wish to make use of the greater flexibilities to innovate in tanker design that any revisions to ADR and its associated standards and technical codes arising from this research may provide. New designs to accommodate potential future increases in vehicle weight limits were identified as being of particular interest. To make use of a performance-based technical code, however, the cost and regulatory burdens associated with gaining approval via performance-based testing would need to be viable so as not to undermine the business case for such innovation. As the research progressed, it became increasingly evident that fully providing for greater flexibilities to innovate in tanker design will inevitably be a lengthy and complex undertaking, especially where a much larger number of “what-if” scenarios would need to be considered and any risks of adverse safety outcomes fully mitigated, across the fewer frontal, side and rear impacts as well as for the more frequent rollover impacts. Even limiting the scope of new performance-based tests to just metallic gravity-discharge tanks is likely to require much detailed study and careful validation to satisfy the relevant authorities.

Given the limitations, we suggest that any further work include an assessment of the market potential for novel petroleum fuel tanker designs (for example against an expected background of productivity concerns and declining petrol/diesel sales) and use the results to quantify the likely cost-effectiveness of any detailed further work.

If this assessment is positive, additional work to complete the development and validate/demonstrate the suitability of the Outline Technical Code could proceed. The following areas of potential further research could be considered to reduce some of the specific uncertainties in the current draft of the Outline Technical Code:

- Modelling of end partitions in the two-compartment subsection

In the modelling work on subsections so far, the end partitions are convex as viewed from the outside. However, other tanker designs may carry out the two-compartment subsection test where at least one of the end partitions are concave. This may affect the subsection

response in the drop test. To assess the significance of this, further subsection modelling work with concave end partitions could be carried out and results compared with those from existing models.

- Topple test modelling of a stuffed tanker

A model of a complete stuffed tanker (generic and/or actual) could be toppled and the deflection at each partition assessed to see if the upper bound of ratio of the maximum deflection to average deflection remains at 1.25 which is the ratio suggested from this work (but based on banded designs). While further use of the generic model already developed would be useful in this regard, a new model based on a real-world stuffed tanker design would have the further advantage that results arising from it could potentially be validated against further physical tests – this aspect of model validation has to-date only been completed for (two) banded designs.

- Further consideration of abrasion testing methodologies

At present two methods are being considered for abrasion testing: a method using a grinding wheel, and a method using tyre testing machines. The grinding wheel method should be something that could be developed using benchtop equipment accessible to tanker manufacturers and test houses. The tyre testing machine method will require larger, specialist equipment that would need to be adapted for abrasion testing. Further activities could usefully be carried out to assess which method would be the more suitable overall.

- Detailed modelling of the most market-prevalent current joint designs

To identify “worst-case” from point of view of omega value in drop test and thus build a baseline evidence base to inform pass-fail criteria development and better ensure the minimum-allowable level of rollover impact performance of novel designs is at least as “safe” as the lowest performing current design(s).

- Validation of drop test (subsection) modelling with equivalent physical tests

To date the only physical testing carried out has been full-scale topple testing. While the modelling approaches used in this research have been validated against that benchmark, they have not yet been demonstrated beyond doubt to accurately replicate structural deformations and other material impact performance aspects under drop-test conditions. Subsection predictive modelling of a real-world design would need to be validated against an equivalent physical drop-test to fully validate such application of the modelling approach.

- Drafting of detailed requirements for novel circumferential joint designs

The likely most straightforward application of the research to date for the future approval of novel tank designs regards specifying the detailed requirements for novel circumferential joint designs only (option 4 in the Outline Technical Code as currently drafted). This potential next step is much more limited in its scope, and thus potentially much closer to future regulatory acceptance and application than the other options so far developed.

Whether or not the suggested strategic review concludes that further work on petroleum road fuel tanker requirements for novel designs is warranted, there is a potentially useful additional application for the research already completed, namely to:

- Explore potential relevance of research carried out to other ADR tanker types

We further suggest that a process of engagement with as full a range of expertise in the materials, impact modelling, simulation and testing community be considered. This could inform the above strategic review and any subsequent further development of the Outline Technical Code but crucially it could also usefully explore the potential relevance of the research carried out to date to other ADR tanker types. If, for example, the review concludes that there is likely to be a much stronger case for innovation and the deployment of novel designs in such non-petroleum markets, it would be apposite to define how much of the work needed to develop appropriate new standards for them can make use of the research carried out already.

4 Part B: Development of impact modelling and associated test parameters - frontal impact

4.1 Introduction and context

As mentioned in Section 2, the accident analysis work reported that almost all significant releases of flammable liquids arise from traffic collision incidents involving rollover and/or collision with another heavy vehicle. These represent only a small proportion of all traffic collision incidents involving flammable liquid (FL) vehicles.

Within this small group of high-risk collisions, rollover is by far the most likely to result in significant releases. However, front to rear collisions can present significant risks of substantial releases of flammable liquid. In the case of articulated vehicles with self-supporting trailers, when such a vehicle is hit at the front and collision forces are transmitted to the tank indirectly through the 5th wheel and king pin assembly.

In detail, analysis of the UK ADR collision reports performed in the Part A accident data work (see Section 2) highlighted a collision in which 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, 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 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.

It was evident that the rear of the cab was modestly damaged in a collision with the front bulkhead of the tank. However, the main damage to the tank which caused the release of flammable liquid was in a region behind the king pin location. 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 4-1, 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, caused the tank to deform and buckle in the region just behind the king pin 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 deformation, above the king pin, and almost all of the fuel in the front compartment of the tanker (c. 7,000 litres) was released.

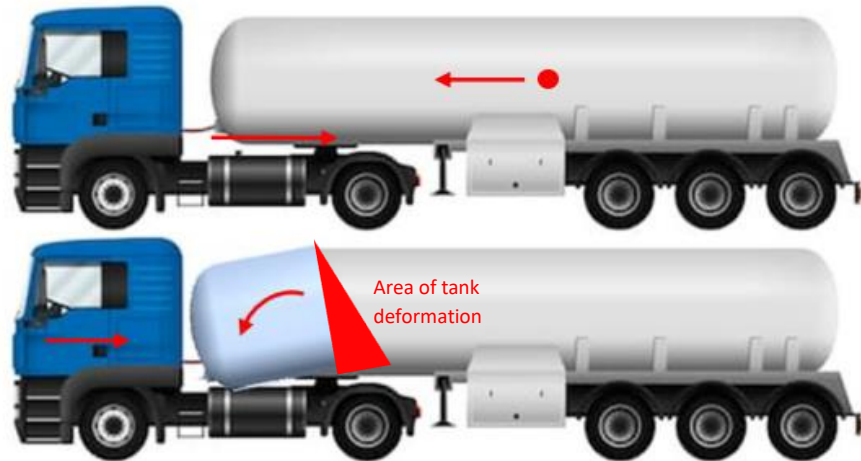


Figure 4-1: Illustration of crash loading of following tanker through king pin assembly in tanker front to rear collision and the resulting deformation (Top: before impact, Bottom: after impact)

This type of loading via the kingpin assembly will also occur for head on (front to front) collisions between two heavy vehicles. The loading in these collisions is likely to be higher on average than for front to rear collisions because, on average, impact velocity changes (ΔV s) in front-to-front type collisions were found to be higher, which results in higher loading.

On the basis that loading via the king pin assembly in frontal collisions (i.e. front-to-rear collisions and front-to-front collisions) can present significant risks of substantial releases of flammable liquid, this type of impact loading scenario was chosen as one for further investigation.

The objective was to model this impact loading scenario to better understand damage and failure mechanisms and help set baselines for future potential test methods.

The Part A accident analysis work also proposed candidate test methods which could form the basis of a future test procedure. Loads could be applied via a sled acting as a tractor unit replacement as illustrated in Figure 4-2. However, a test of this nature would probably prove too expensive, so substitute methods of applying a load to the king pin assembly were proposed such as quasi-static loading or loading applied using a drop or pendulum tests, methods used in current regulatory tests.

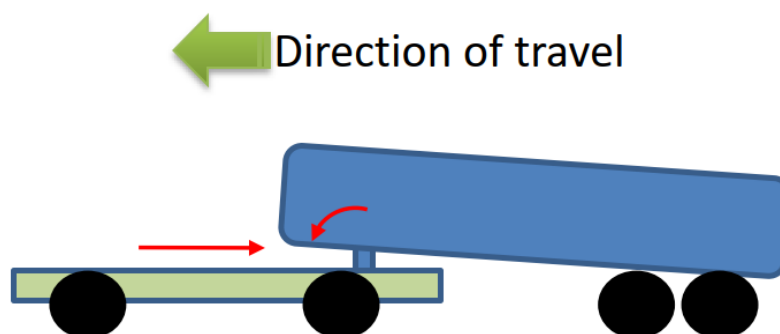


Figure 4-2: Diagram to illustrate king pin assembly loading during a frontal collision

The modelling methods and programme, results, and discussion of the investigation are summarised in the sections below.

4.2 Modelling methods and programme

4.2.1 Tanker model geometry

The baseline model used for this work was a full tanker finite element model developed and validated as part of a previous study for the DfT which investigated roll-over type impacts¹¹. The model was of a banded type 1¹² tanker design whereas the tanker involved in the collision analysed in the Part A work was a stuffed type design and had a shorter king pin assembly support structure length than the banded type 1 tanker design.

Given that costs and time to develop a model of the tanker design type involved in the collision were high and the objective of the work ‘to better understand damage and failure mechanisms’, could be achieved using the previously developed tanker model if additional model runs were performed to investigate items such as king pin assembly structure length, it was decided to use the tanker model developed previously for this work.

As mentioned above, it was noticed that the tanker involved in the collision had a shorter king pin assembly support structure than the tanker FE model chosen for use in the work. Also, based on examination of photographs of the crashed tanker, it was noted that the shorter king pin assembly structure in the collision involved tanker did not extend the full length of the front compartment, whereas the longer structure in the FE model did. A shorter length of king pin assembly structure will increase the forces required to resist the moment of loads applied to the kingpin as illustrated in Figure 4-3 which could lead to an increased risk of tank deformation / failure.

¹¹ Atkin C, Hobbs J, and McDonald A. (2015) Technical assessment of petroleum road fuel tankers. Work Package 1 extension – full scale testing and associated modelling – overall summary ES/14/39/00rev8. Available from: <https://www.gov.uk/government/publications/petroleum-fuel-tankers-technical-assessment-november-2015> ; note download Work Package 1 files

¹² See Section 3.5.1 for a description of banded and stuffed type tanker designs.

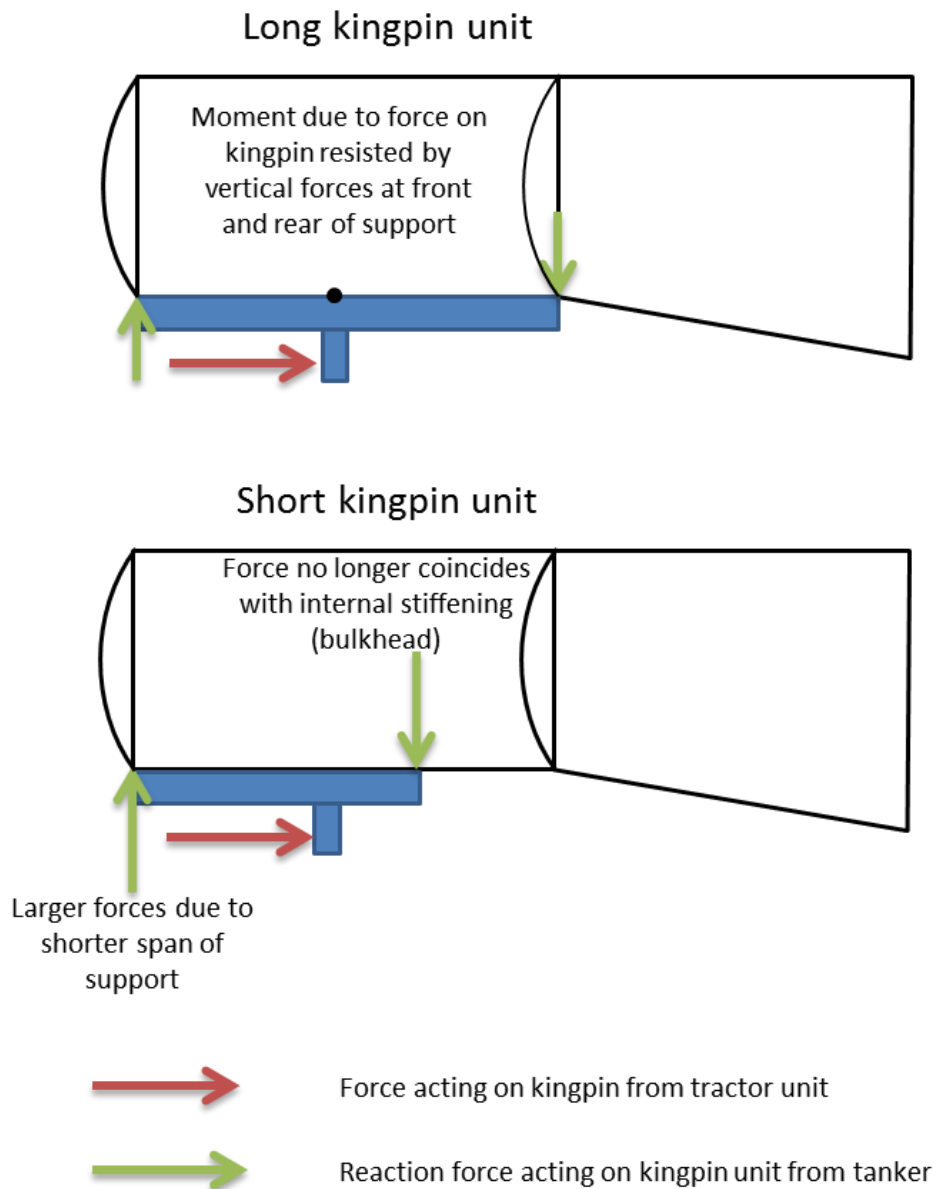


Figure 4-3: Schematic diagram illustrating likely effects of shortening the king pin assembly unit

Also, depending on where partitions are positioned, this increased force could act upon an un-strengthened part of the tank which could increase the risk of tank deformation / failure further. On this basis both short and long king pin assembly support structures were modelled as shown in Figure 4-4 to investigate the effect of king pin assembly support structure length on tanker response. The long king pin assembly unit model consisted of the banded type 1 tanker with no changes. The short king pin assembly unit model, which better represented the collision involved tanker, was developed by modifying the long king pin unit model; the king pin assembly unit length was shortened and a simple saddle was added to distribute the load from the shortened unit and better represent the collision involved tanker.

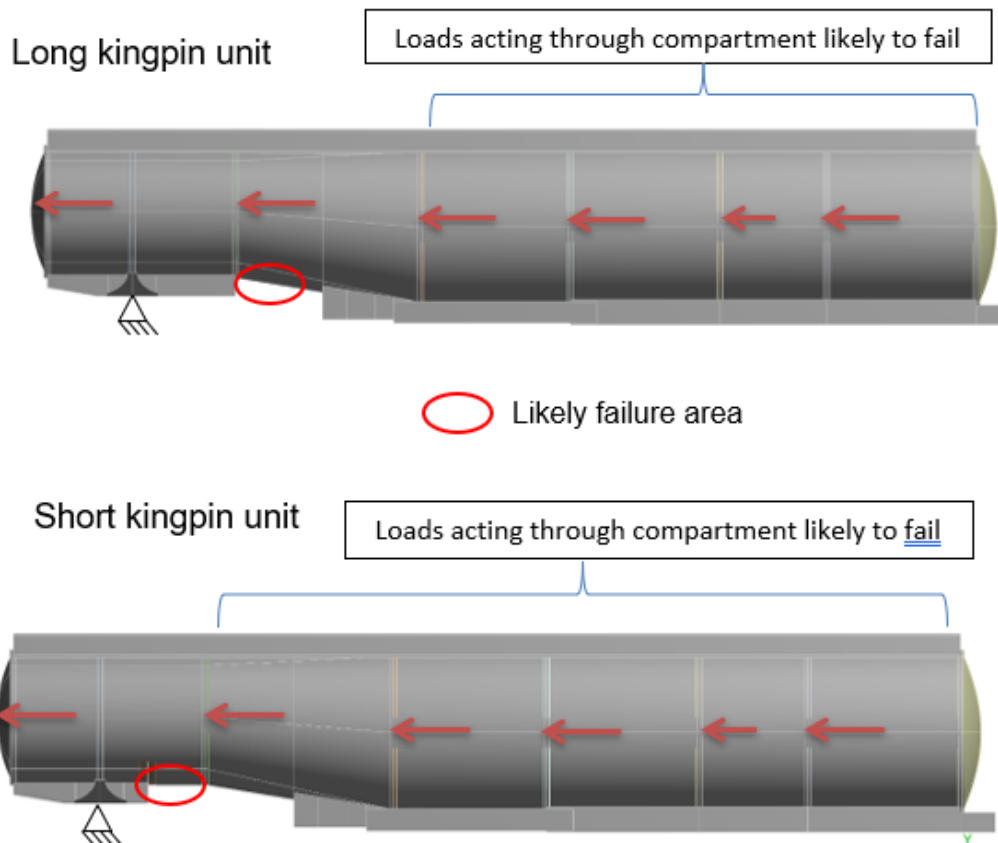


Figure 4-4: Models of tanker with long and short king pin assembly units showing frontal impact loads and likely failure areas

4.2.2 Loads and loading methods

Loads

The following standards set requirements for tankers related to king pin loading.

EN 13094:2022 'Tanks for the transport of dangerous goods – metallic gravity-discharge tanks - - design and construction' requires that the design stress for tank shells and their attachments is not exceeded for longitudinal accelerations (2 g) in the direction of travel.

The standard 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. 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.

Because some tankers may be supported along part of their length, it is understood that this 2 g requirement is generally evaluated using an instrumented tanker undergoing braking with a deceleration of approximately 0.6 g and extrapolating the results to the 2 g requirement. However, because the semi-trailer brakes as the tractor unit brakes under the test condition, this will reduce the king pin loading. Therefore, although this test replicates

the distribution of loading seen under heavy braking in service, it does not replicate the loading via the kingpin that would be seen in crash scenarios. This is because in crash scenarios most of the load to decelerate the semi-trailer would be applied through the king pin.

ISO 1496-3:2019 'Series 1 Freight containers – Specification and testing Part 3: Tank containers for liquids and pressurised dry bulk', specifies that the containers can withstand a force equivalent to an acceleration of 2 g.

On the basis of the 2 g loading specified in the standards above it was decided to perform simulations which effectively applied loads to the king pin of a sufficient magnitude to decelerate the maximum design mass of the trailer at least circa 2 g. Decelerations in the collision investigated in the accident analysis work were likely to have been greater than 2 g.

Loading methods

To achieve loading representative of crash type loads the following three different methods were used:

- Static (displacement controlled)
 - Longitudinal displacement of up to 40 mm maximum applied to king pin and restraint applied to rear of tanker.
 - Notes:
 - Implicit solver used.
 - Using this method to apply the load, the full force is transmitted through every compartment which is somewhat unrealistic.
- Quasi-static (force controlled)
 - King pin restrained (in x, y, z) and forces in the form of a ramp applied to tanker partitions/bulkheads to represent loading to decelerate liquid in corresponding compartments up to 4 g.
 - Notes:
 - Implicit solver used.
 - Using this method to apply load gives a more realistic distribution of the forces along the length of the tanker compared to the static method above.
- Dynamic (solid mass in tanker)
 - Solid elements with the same density of water to represent liquid in the tanker rear four compartments were added to model and a rearward acceleration was applied to the king pin location, one simulation with 2 g and another with 3 g.
 - Notes:
 - Explicit solver used.
 - Solid elements were used to reduce computer run time for models.

- Because solid elements were used the two front compartments could not be filled otherwise, because of elements solid nature, they would have prevented deformation of these compartments.

Further details of the loading methods can be found in Appendix F.

Long and short kingpin structures were simulated using all three modelling methods and king pin loads measured.

4.3 Results

For all three loading methods and both lengths of king pin units, the king pin loads were limited because of failure (buckling) of the tank structure (Table 4-1). Much lower loads could be sustained without a buckling failure (and possible rupture) for the short king pin unit compared to the long one, on average a 30% higher load could be sustained for the long unit.

Table 4-1: Comparison of king pin maximum loads for the different loading methods and king pin units

Modelling loading method	King pin unit maximum load (kN)	
	Long Unit	Short unit
Static (displacement controlled)	855	713
Quasi-static (load controlled)	1048	756
Dynamic (2 g acceleration)	980	760
Dynamic (3 g acceleration)	1045	781
Average	982	753

The buckling is illustrated for the Dynamic (solid mass in tanker) method in Figure 4-5 which shows the different failure positions for the different king pin unit lengths. Further details of the results can be found in Appendix F.

The dynamic model showed that the force applied at the kingpin could exceed what would be expected under a steady acceleration, if the acceleration was rapidly applied. Therefore, any performance requirements based on acceleration should include the manner of its application and its duration. On this basis, an approach based on a requirement in terms of a force equivalent to a steady acceleration could be better because this could be defined more easily and precisely. Also, testing using a quasi-static force, rather than a dynamic load, is likely to be easier and more controllable. A quasi-static type approach is commonly used in regulatory tests for assessment of vehicle structural components, for example:

- UN Regulation No. 29 'Roof strength of cabs'
- UN Regulation No. 58 'Rear underrun protective devices'

and therefore can be an acceptable method. However, careful thought would be required as to how the differences arising from loads being applied to the bulkheads would be considered.

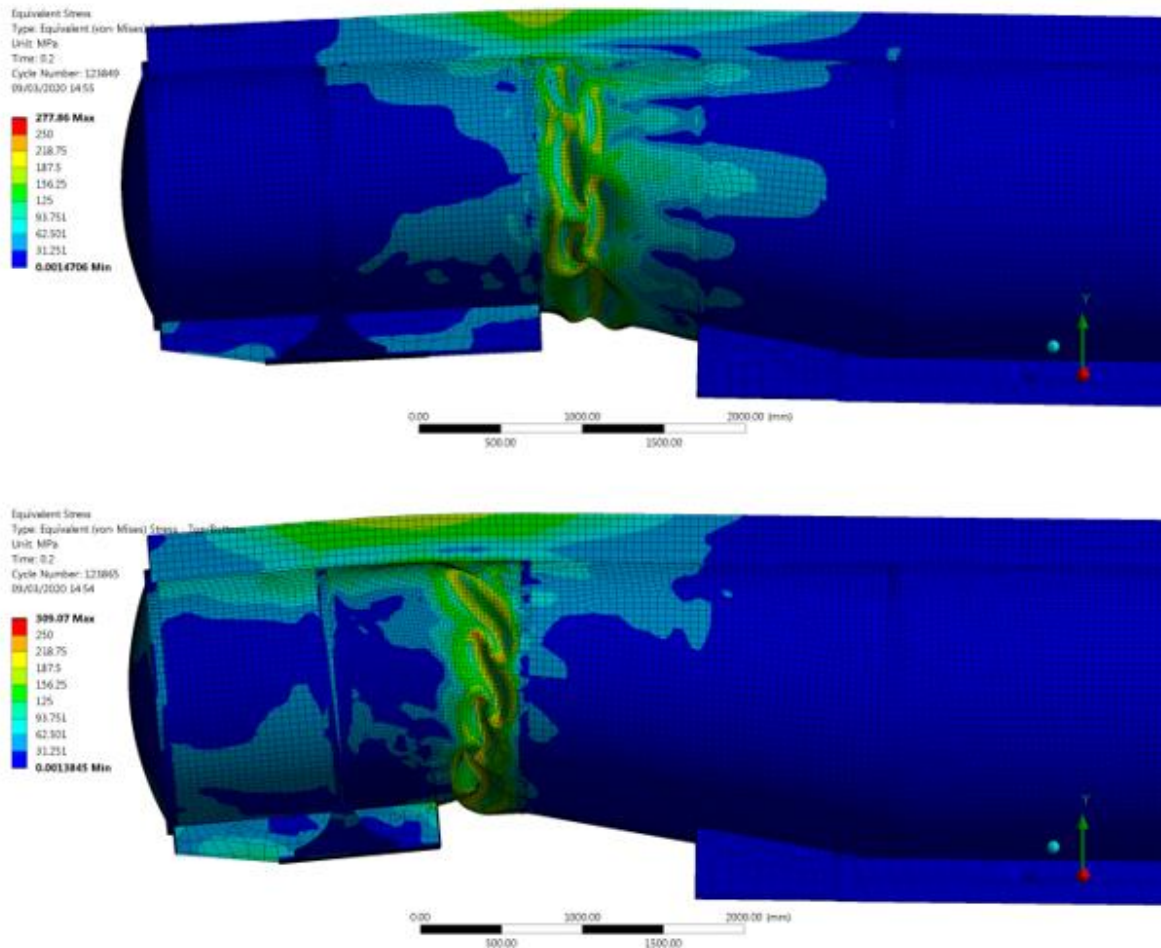


Figure 4-5: Deformation of the tanker under 3 g deceleration (dynamic model)

4.4 Discussion and summary of findings (frontal impact)

As mentioned earlier, in Section 4.2.2, EN 13094 sets requirements for tanker shells and attachments related to king pin loading, but the manner in which these requirements are implemented means that they do not replicate king pin loading that would be experienced in frontal crash scenarios. These requirements must be met in order for the tanker to achieve ADR compliance.

The modelling results show that much lower loads could be sustained without a buckling failure (and possible rupture) for the short king pin assembly unit compared to the long one, on average a 30% higher load could be sustained for the long king pin unit. In detail, the modelling results indicate that buckling in the region of tanker front compartments (and possible rupture) could occur with king pin loads ranging from 855 – 1048 kN, (average 982 kN), for long assembly units and 713 – 781 kN, (average 753 kN), for short assembly units.

Simple calculations using $F = ma$ show that if it is assumed that the majority of the load is via the king pin assembly in collisions for a 2 g deceleration with a semi-trailer mass of 37 t the king pin load will be circa 725 kN and for a 3 g deceleration, 1090 kN. Comparison of these loads with the modelling results indicates that, assuming king pin crash loads are somewhere between 2 and 3 g, there is a high likelihood that failures in the region of the king pin will occur for tankers with short king pin unit lengths and some likelihood for long king pin unit lengths if high enough decelerations are experienced during the frontal collisions, i.e. circa 3 g or more. These deceleration magnitudes are less than half of the 6.6 g minimum required in UNECE Regulations No. 67, 110, and 100 for the mechanical integrity of propulsion system components in crashes, such as the attachments of pressurised fuel tanks (R67 & 110) and battery mechanical integrity (R100).

The accident analysis only found one example of tank failure in the region of the king pin assembly in frontal impacts recorded in the ADR obligated reporting system. Given that results from the structural analysis indicated a high likelihood of failure for short king pin assembly unit lengths for deceleration levels that are less than half those required in regulation, the question arises why more examples were not found. There are a number of possible reasons for this which include:

- Collision type rare

Fuel tanker frontal impact collisions with heavy vehicles which produce the king pin loading conditions modelled are rare, the Stats19 accident analysis estimated about 2.0% of all fuel tanker collisions for all impact severities (Table 4-2). For the 2009 to 2018 time period, this was about 15 cases. If it is assumed that in about 30 to 50 percent of these, deceleration levels of 2 to 3 g were experienced - note that regulation requires minimum deceleration levels of 6.6 g, so 2 to 3 g is relatively low, this is 5 to 8 cases in which tank damage and a fuel spillage could be expected to be found in the ADR collision reports. However, the accident analysis only reviewed ADR collision reports for the period of 2014 to 2018, half the time period of the Stats19 analysis, so this halves the number of cases to 2 to 4 in which tank damage and a fuel spillage could be expected to be found. Also, it is not certain how complete reporting of spillages via the ADR obligation is, although it is expected that all major spillages would be recorded. Therefore, in summary, taking the points above into account perhaps it could be expected that only one example case was found.

Table 4-2: Collisions of FL tankers by impact point and collision partner Source: STATS19 database (2009-2018) n = 744

Collision partner	Front	Rear	Side	Total
Heavy vehicle	2.0%	3.4%	3.6%	9.0%
Light vehicle	37.1%	12.0%	31.0%	80.1%
Two-wheel vehicle	3.6%	1.6%	4.3%	9.5%
Other	0.7%	0.2%	0.5%	1.4%
Total	43.4%	17.2%	39.4%	100%

- Few tankers in fleet prone to failure type
There may be few tankers with no longitudinal frame support (i.e. chassis) behind king pin area and with short king pin assembly unit lengths in the current fleet.
- FE models (initial) not fully representative of real-world tanker designs and/or associated loading not fully representative of real-world collision loads
The model of short king pin assembly unit, in particular its attachment and bulkhead support, may not be representative of real-world designs. A representative model may predict a greater king pin load carrying capacity before tank failure. The loads applied to the model may be greater than in real-world collisions, thus application of real-world loads should predict less likelihood of failure.

The following work elements could be performed to gather information to further assess the performance of articulated fuel tanker trailers in frontal impacts, understand better the risk of tank failure and flammable liquid spillage in these collisions, and help determine whether, or not, this risk is similar for all types of design of tanker or if it is related to particular types of design.

- Accident analysis to:
 - Expand analysis of ADR collision reports to identify further relevant collisions.
 - Compare national (Stats19) data and ADR reports to understand better reporting levels for incidents involving petroleum road fuel tankers and gain confidence that reporting system is sufficiently robustly capturing such incidents to be a reliable guide as to frequency.
 - Estimate better king pin loadings in relevant collision(s).
- Tanker fleet survey to:
 - Determine proportion of tankers in current fleet potentially prone to buckling failure, i.e. those self-supported (i.e. without chassis) in area behind king pin and/or with a short king pin assembly unit length
- FE modelling to:
 - Improve the tanker model so that it is representative of the tanker involved in the collision highlighted in the accident analysis.
 - Improve the loading method and the load magnitude so that it is more representative of the loading experienced in the collision highlighted in the accident analysis.
 - Compare and contrast performances of different tanker trailer designs.

This work should be able to identify if a noticeable real-world issue exists for frontal impacts in general or with specific makes / models of tanker trailer and how it may be best addressed in view of ADR requirements, referenced standards and technical codes.

5 Conclusions, research limitations and suggested next steps

5.1 Overall conclusions and summary of key findings

The aim of the research reported here was to develop ‘performance-based’ finite element modelling approaches and appropriate physical test methods to approve tankers with novel designs that would not meet current ‘design-based’ approval requirements, *i.e. to provide an alternative means of approval that gives more freedom to innovate while maintaining an equivalent (the same or better) level of safety.*

It is anticipated the new test procedures will enable manufacturers to develop tank shells using design and construction methods not necessarily depicted in the existing standards, but which are nevertheless able to sustain rollover, frontal, rear and side impacts without having to use a series of more costly full-scale tests to demonstrate the suitability of their tank shells.

While full achievement of this aim has not been possible within the time and budgetary constraints of this specific project (‘the Part B research’), substantial progress towards its achievement has nevertheless been made.

5.1.1 Rollover impacts

For the rollover impact scenario, the research points firmly towards subsection drop-tests (or whole tanker topple tests), combined with requirements for abrasion and penetration resistance, as being the most appropriate alternative, performance-based approval methodologies. An outline technical code has been drafted as a first step towards the future adoption of such methodologies within the ADR framework.

In developing this outline technical code, the researchers have made comprehensive use of finite element and other fluid and structural impact modelling techniques, forming limit parameters and other metallic materials properties, and their knowledge of physical test considerations and a wide range of existing test methods of potential relevance.

For the metallic tankers assessed, this study has thus identified strong relationships (large coefficient of determination R^2) between impact energy and structural response parameters for tanker drop models. This shows that it should be possible to replicate structural responses observed in full-scale topple tests in a subsection drop test, with appropriate impact energy (which is directly related to compartment length, fill volume, liquid density and drop height).

The forming limit for metallic tank structures 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. Failure becomes more likely the higher the omega value is above 1. Omega is thus a useful measure of how close or how far the strain conditions are to or beyond the forming limit and the research has demonstrated its potential applicability to assessing the risk of structural failure in a tanker topple and/or subsection drop-test scenario.

The main conclusion regarding the development of performance-based requirements for rollover safety is thus that:

The deflections and likelihood of major loss of containment experienced by tankers in real-world rollover scenarios can be replicated in a suitably specified, two-compartment subsection drop-test (or a full-scale physical topple test) supplemented by abrasion and penetration tests.

In a rollover scenario, loss of containment can arise via any one of three mechanisms; the initial impact with the ground, abrasion of the outer shell structure as the tanker then slides along the road surface, or penetration of a hard object through the tank structure.

Resistance to penetration requirements already exist in ADR via the referenced standard EN 13094 and can readily be adopted and/or adapted for approving novel designs.

Various existing test methods potentially relevant to abrasion testing of novel tanker designs were assessed but none was found to be directly applicable without further development. Concept test methods based on a grinding wheel and/or tyre durability testing were selected as the test methods most likely to be suitable for further development into an abrasion resistance test for tanker rollover scenarios.

5.1.2 Other impact scenarios

The research has not been able to develop outline alternative requirements for other fuel tanker accident scenarios such as frontal, rear or side impacts.

For frontal impacts, however, the research has identified a potential weakness with existing ADR requirements, specifically regarding how impact loadings may be transferred to the tank shell via the king pin on an articulated tractor-trailer combination. EN 13094 sets requirements for tanker shells and attachments related to king pin loading, but the way these requirements are implemented means that they do not replicate king pin loading that would be experienced in a frontal crash scenario.

Modelling of this scenario indicates there is a high likelihood that failures in the region of the king pin will occur for tankers with short king pin unit lengths and some likelihood for long king pin unit lengths if high enough decelerations are experienced during frontal collisions.

5.2 Research limitations and suggested next steps

The following sections describe some important and specific limitations of the research carried out and make suggestions for further detailed research to address them.

It is reasonable to assume, based on some informal discussions, that a number of manufacturers may wish to make use of the greater flexibilities to innovate in tanker design that any revisions to ADR and its associated standards and technical codes arising from this research may provide. New designs to accommodate potential future increases in vehicle weight limits were identified as being of particular interest. To make use of a performance-based technical code, however, the cost and regulatory burdens associated with gaining approval via performance-based testing would need to be viable so as not to undermine the business case for such innovation. As the research progressed, it became increasingly

evident that fully providing for greater flexibilities to innovate in tanker design will inevitably be a lengthy and complex undertaking, especially where a much larger number of “what-if” scenarios would need to be considered and any risks of adverse safety outcomes fully mitigated, across the fewer frontal, side and rear impacts as well as for the more frequent rollover impacts. Even limiting the scope of new performance-based tests to just metallic gravity-discharge tanks is likely to require much detailed study and careful validation to satisfy the relevant authorities.

Given the limitations, we suggest that any further work include an assessment of the market potential for novel petroleum fuel tanker designs (for example against an expected background of productivity concerns and declining petrol/diesel sales) and use the results to quantify the likely cost-effectiveness of any detailed further work.

If this assessment is positive, additional work to complete the development and validate/demonstrate the suitability of the Outline Technical Code (for rollover) could proceed, and frontal, side and rear impact scenarios considered in detail.

We further suggest that engaging wider expertise in the materials, impact modelling, simulation and testing community be considered. This could inform the above assessment and any subsequent further development of the Outline Technical Code but could also explore the potential relevance of the research carried out to date to other ADR tanker types.

5.2.1 Rollover impacts

The research into rollover impacts and the development of an outline technical code for future performance-based requirements has the following main limitations:

- The findings are based on somewhat simplified and idealised representations of tankers. They may not represent the full range of actual real-world designs.
- All three designs modelled represent tank structures that have been approved under ADR requirements. No “novel designs” have been assessed.
- Non-metallic structures have not been investigated.
- Subsection drop-tests are considered to be the most cost-effective option (instead of full-scale topple tests), but the likely savings have not been quantified.

The following areas of potential further research could be considered to reduce some of the specific uncertainties in the current draft of the Outline Technical Code:

- Modelling of end partitions in the two-compartment subsection.
- Topple test modelling of a stuffed tanker.
- Further consideration of abrasion testing methodologies.
- Detailed modelling of the most market-prevalent current joint designs.
- Validation of drop test (subsection) modelling with equivalent physical tests.
- Drafting of detailed requirements for novel circumferential joint designs.

5.2.2 *Frontal impacts*

The following work elements could be performed to gather information to further assess the performance of articulated fuel tanker trailers in frontal impacts, understand better the risk of tank failure and flammable liquid spillage in these collisions, and help determine whether, or not, this risk is similar for all types of design of tanker or if it is related to particular types of design.

- Accident analysis to:
 - Expand analysis of ADR collision reports to identify further relevant collisions.
 - Compare national (Stats19) data and ADR reports to understand better ADR reporting levels and gain confidence that reporting system is sufficiently robustly capturing incidents to be a reliable guide as to frequency.
 - Estimate better king pin loadings in relevant collision(s).
- Tanker fleet survey to:
 - Determine proportion of tankers in current fleet prone to buckling failure, i.e. those self-supported (i.e. without chassis) in area behind king pin and/or with a short king pin unit length
- FE modelling to:
 - Improve the tanker model so that it is representative of the tanker involved in the collision highlighted in the accident analysis.
 - Improve the loading method and the load magnitude so that it is more representative of the loading experienced in the collision highlighted in the accident analysis.
 - Compare and contrast performances of different designs.

This work should be able to identify if a real-world issue exists for frontal impact type impacts in general or with specific makes / models of tanker and how it may be addressed in view of ADR requirements.

6 Acknowledgements

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

In particular, we would like to thank Dr Michael Sutcliffe of CUED for his insight and expert guidance as a peer reviewer throughout the project. We also owe a debt of gratitude to the stakeholders from the UK fuel tanker industry, including manufacturers and inspection bodies, that have so diligently contributed their time, knowledge and experience to the research.

We also gratefully acknowledge the contributions of our many current and former colleagues in HSE SD, TRL, Apollo Vehicle Safety and TWI who have worked with us, including Duncan Webb (HSE SD), Damaso De Bono (TWI), Martin Dodd (Apollo), Stuart Greenshields (TRL) and Tanya Robinson (TRL).

Finally, we must acknowledge the enormous contribution made by our client, the Department for Transport's Dangerous Goods Division and, especially, the tireless and invaluable support provided by Steve Gillingham and David Adams.

Appendix A Review of potential rollover impact test methods

Tables A1, A2, A3, A4, A5 and A6 and accompanying text present a detailed review into the potential advantages and disadvantages of pendulum, drop, horizontal, ramp impact tests (two types) and topple testing. Each method was assessed initially on a range of criteria including whether such a method was already in use for international standards purposes, if adaption to cater for tanker rollover impacts was feasible, how practical such a test would be to implement, how repeatable its results would be, how representative of a real-world rollover impact it would be, etc.

A.1 Pendulum tests

Table A-1: Pendulum tests on tank subsections

1	2 Tank Full	3 Tank Empty
4 Advantages	5 It is unlikely that the high impact energy needed is possible to obtain the necessary deformations for a subsection tank that is full of water. Therefore the pendulum method is probably not suitable for the tank-full case.	6 Larger deformations can be achieved with the tank empty than with the tank full. This means equivalent deformations to a full-scale rollover can be achieved with much lower impact energy.
	7	8 No additional filling/emptying required (not a significant advantage).
	9	10 May be able to use the same test rig design as for UN/ECE Reg 29 Cab tests.
	11 The tank can be oriented to a range of positions if particular areas of the tank are required to be struck. This may make it easier to impact a precise area (if this is needed).	
12 Disadvantages	13 The width of the impactor would probably only span 2 bands at the most (even on a 10-banded tanker). The impactor in UN/ECE Reg 29 is 2.5 m wide. An impactor much wider than this would mean the pendulum rig would have to be very large and heavy. Therefore, the pendulum method is probably only suitable for impacting one band/bulkhead zone.	
	14 As the rig has to take the reaction from the tank on impact, any movement (flexure or looseness) in the rig would mean that some of the kinetic energy would be going into the rig movement rather than being absorbed by the tank). Therefore, very specific design requirements would be needed to ensure that rig movement on impact is very low. Test results would probably be less reproducible if different tests houses designed different rigs. This should not be the case for drop testing where only the mass and rigidity of the impact pad needs to be specified.	
	15 A restraint has to be designed so the tank does not move at impact. Again, any flexure or looseness in the restraint system reduces the impact energy into the tank.	
	16 The pendulum method is less similar to a real-life impact scenario than a drop test. By itself this may not be a disadvantage, however other stakeholders may prefer to see a test that is closer to real-life.	
	17 The response of the water will be different to what it would be in a drop and topple scenario. In a drop/topple scenario, the water will be accelerated onto the impact side of the tanker immediately after impact. Therefore, this will require determining equivalent levels of damage.	18 Pendulum tests on an empty tank rely on equivalence between deformations from an impact on an empty tank, and deformations from a rollover on a water-filled tanker. This has only been modelled for one case: a banded aluminium tanker. At the moment it is not known if this 'equivalence' is similar for stuffed aluminium tankers, steel tankers, and tankers with new materials. ¹³

¹³ If a 55 kJ impact with a pendulum gave x mm deflection at a band on an empty aluminium tanker, and x mm of deflection was also observed in a water-filled rollover test, this gives an 'equivalence' i.e. the pendulum test gives an indication of the damage that would occur in a rollover. However, if a tank made of another material is tested and the pendulum impact produces x mm of deflection again, but a water-filled rollover test produced a deflection of y mm, the difference may be due to a difference in tank mass or the motion of the

A.2 Drop tests

Table A-2: Drop tests on tank subsections

	Tank Full	Tank Empty
Advantages	Closer to a real-life topple scenario so there is no need to relate the damage from an empty tank to a tank full of liquid. If new tank material were used in the future, this could be a significant advantage.	No additional filling/emptying required (not a significant advantage).
	It will be easier to target the area of interest on the tank as the drop height is only likely to be around 1 m and the greater inertia means it is less likely to rotate out of position during the drop. Also, there is less likely to be secondary bouncing.	Lifting the subsection should be achievable with proprietary lifting equipment.
	Leakage after the test would indicate if the tank had failed (although it may be necessary to leave the tank for a period of time to determine if a leak has occurred). Therefore, there may be no need for a hydraulic or pneumatic test. However, if one of the pass/fail criteria was to be whether the bulkheads have failed, then a pressure test would probably be needed).	As the empty tank is much lighter than the full tank, it should be possible to raise and drop it just with the use of a crane and some additional lifting points attached to the subsection.
	Unlike the pendulum rig, the reaction is taken by the target pad and not the test rig. Providing the drop target is suitably unyielding (e.g. 50 mm thick steel plate bolted to a concrete slab), all tests results from different test facilities should be reproducible.	
	There are less parameters that need to be specified in the test method than the pendulum, topple or ramp tests and the test rig design and build costs are likely to be lower.	
Disadvantages	Achieving an exact point of impact is more difficult than in a pendulum test, particularly for the tank empty where the drop height will be around 10 m.	
	Tanks are generally not designed to be lifted from above. Instead, slings reaching under the tank (under-slinging) are used. In a drop test these may affect the test result, so lifting points on the upper side of the tank will probably be needed (discussed further in the row below, and in Appendix 2).	
	For drop testing, this would be a greater mass to lift and drop (around 10 to 15 tonnes). Many tank manufacturers and test houses may need to organise a contract lift as it is likely to be above the specification of their crane, or build a dedicated drop test frame. Many crane suppliers may consider a drop test with this mass too high for their cranes. Also, lifting points for the drop test would need to be designed into the tanker subsection. The banded type 1 tanker bands were 16 mm deep in the centre, and 50 mm wide on the outer side. Early modelling suggests that this will be sufficient metal to install lifting points. With stuffed tankers a welded saddle or welded bosses etc. in the shell at the bulkhead is a possible answer.	Less suited for a drop test; the height would need to be higher, which gives 2 problems: 1. Getting the tank to hit the target in the specific area of interest becomes progressively more difficult the greater the drop height. 2. It is likely there will be secondary bouncing impacts which could provide additional less-controlled damage. Also, there is still the issue of, matching equivalent deformations on an empty tank to a tank full of liquid as described for the pendulum tests.
	May be more difficult (but not impossible) to do other impact scenarios as additional lifting points may be needed to suspend the tank in the right orientation for rear and front impacts (but this becomes an issue only if testing for these other scenarios becomes important).	
	If pressure relief is fitted as standard, then fully functioning pressure relief valves will be needed for the test to ensure pressure changes in the compartment are similar to a real-life scenario. This may require the tank to be in a more completed form than an empty tank (this will need a comment from tank manufacturers as this might not be seen as an issue to them).	If pressure relief is fitted as standard, then some form of pressure relief will still be needed for the tank to vent the pressure build up, but it may be a simpler form than for a tank full of liquid, and it may be possible to do this after the test rather than during the test.
		Loss of containment would have to be verified by a pneumatic or hydraulic pressure test.

fluid in the tanks being different due to a different shape etc. So the pendulum test may no longer represent a rollover test in the 2nd case. At the moment we don't have enough information to know if the 'equivalence' would also be similar for tanks of other materials and designs than banded aluminium tankers. This needs to be considered for drop or pendulum tests on empty tanks.

A.3 Horizontal impact tests

Table A-3: Horizontal impact tests on tank subsections

	Tank Full	Tank Empty
Advantages		Larger deformations can be achieved with the tank empty than with the tank full. This means equivalent deformations to a full-scale rollover can be achieved with much lower impact energy, but a higher impact speed would be needed.
		No additional filling/emptying required (not a significant advantage).
Disadvantages	Specialist sled- test equipment may limit the number of places that could carry out the test and may also need to be uniquely designed for such a large sample which is quite different to impact testing a vehicle. With a limited numbers of test houses, there is a risk that, if the test houses discontinue with work in this area, there may be few places that can actually carry out the test.	
	The test specimen would be 5 to 6 m wide. Accelerating this horizontally for a broadside impact on a platen which is also 5 to 6 m wide (or vice versa) may prove impractical.	
	The test specimen would weigh 10 – 15 tonnes. It may require a significant power and control system to lift, position, then accelerate this to the impact speed.	
	Water spillage could be a problem in the specialist test environment.	
	As the rig has to take the reaction from the tank on impact, any movement (flexure or looseness) in the rig would mean that some of the kinetic energy would be going into the rig movement rather than being absorbed by the tank). Therefore, very specific design requirements would be needed to ensure that rig movement on impact is very low. Test results would probably be less reproducible if different tests houses designed different rigs. This should not be the case for drop testing where only the mass and rigidity of the impact pad needs to be specified.	
	A restraint has to be designed so the tank does not move at impact. Again, any flexure or looseness in the restraint system reduces the impact energy into the tank.	
	This method is less similar to a real-life impact scenario than a drop test. By itself this may not be a disadvantage, however other stakeholders may prefer to see a test that is closer to a real-life scenario.	
	<p>The response of the water will be different to what it would be in a drop scenario. The effect of water inertia during acceleration will tend to move the water more to the rear side of the tank. At the impact side there will be less water. In a drop test, the water will cover the impact side at (or immediately after) impact. The water response in a horizontal impact test will be different and it is not clear what difference this would make.</p> <p>If the tank is stationary and the impactor is accelerated, the water would still not be in the same position as it would be in a drop or topple scenario.</p> <p>Therefore, assessing equivalent levels of damage will be needed, as mentioned for pendulum test method.</p>	Tests on an empty tank rely on equivalence between deformations from the impact on an empty tank, and deformations from a rollover on a water-filled tanker. This has only been modelled for one case: a banded aluminium tanker. At the moment it is not known if this 'equivalence' is similar for stuffed aluminium tankers, steel tankers and tankers with new materials.

A.4 Drop and slide test on a ramp (RAMP 1)

The next test option is a drop test with the tank sliding down an incline (i.e. an “oblique angled drop/slide test”) using a ramp as shown in Figure A-1.

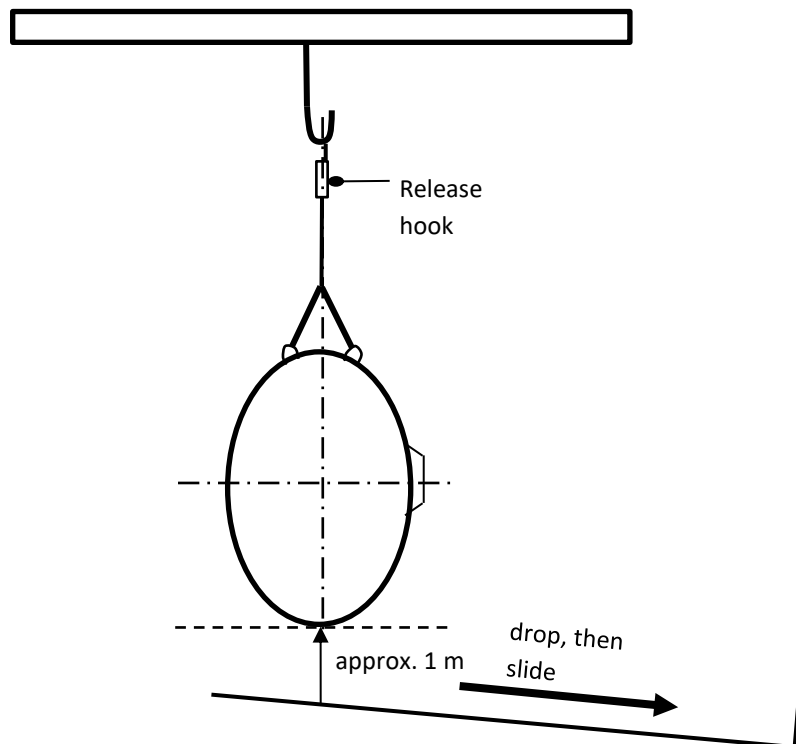


Figure A-1: Drop and slide test on a ramp

Table A-4: Drop and slide test on a ramp (RAMP 1)


	Tank Full (there would be no value in doing this test with the tank empty)
Advantages	This could incorporate abrasion effects into the test. In rollover impacts observed, the tank tends to slide after the initial impact.
Disadvantages	The repeatability challenges due to friction could be significant, and may make this method extremely challenging. The ramp surface would have to meet certain roughness parameters, and after X number of tests the surface may need to be re-ground.
	There could be significant preparatory work (including testing) in deciding what the test surface should be for the sliding part of the test.
	Dropping a 16-tonne weight onto a ramp and ensuring the ramp is ‘unyielding’ could be very challenging. Therefore, a raised impact block with a ramp may need to be built. This may require a specialist test rig which may limit the number of places that could carry out the test. With a limited numbers of test houses, there is a risk that, if the test houses discontinue with work in this area, there may be few places that can actually carry out the test.

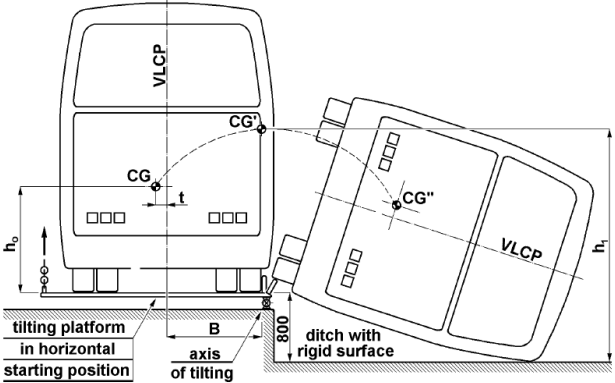

A.5 Topple test

Designing a test chassis to attach to the tank when carrying out a topple test is the next option to consider. Its advantage is that it is closer to a real-life rollover impact scenario, but the mass and stiffness of the chassis itself would be a variable in the test, and this could affect the accuracy between different test centres if they are not using the same design of chassis. However, if the test chassis was designed to be relatively unyielding in relation to the deflection of the tank, and this chassis was specified as a requirement of the test, then reproducibility issues between test houses would be reduced. The test chassis would need to be designed so it is at the same height and width as a semitrailer.

There is little need to carry out a topple test with the tank empty. The worst-case scenario will be with the tank full as the impact energy is much higher, and as filling the tank should not be an issue, topple testing with the tank empty is not considered as an option.

Table A-5: Topple test on full tank

Tank Full (testing with the tank empty is not considered a suitable option)	
Advantages	A release hook is not required which removes the potential challenges that this could have.
	This test would be recognised by other stakeholders as being closer to a rollover impact scenario than all the other test methods (it might need less explaining).
	The type of test rig required is well understood as it will be similar to the 5 th wheel structure (but bigger and maybe in two pieces) that was used in the previous work.
	 <p>The test results from the previous work have already shown that this method is a controllable and repeatable test. These test results showed that the impact energy was sufficient to cause loss-of-containment in tanks that had defects due to lack-of-fusion in the welds.</p>
Disadvantages	The tank will need to be welded or bolted onto the test chassis; it would then need to be removed from the chassis at the end of the test. This will require some extra preparation between tests.
	However, the manufacturer may be able to supply the sub-section with the rear rails already fitted. These could then be bolted to a chassis.
	HSE's modelling work has shown that petrol load creates higher levels of deflection than water load (for equivalent mass). Therefore we would need to consider increasing the water mass above the maximum payload to give the equivalent level of damage to a petrol load.
	<p>The mass of the test chassis should be designed so it has a mass per unit length similar to a semitrailer. The test chassis will need to be specified.</p> <p>The drop height is not easily adjusted. If rigid vehicle tanks are at a slightly different height to a semi-trailer, then a spacer plate may need to be added/removed to the chassis, a different test chassis may need to be designed, or thick shims could be bolted onto the feet of the chassis (platform shoes) to adjust the height.</p>

	Tank Full (testing with the tank empty is not considered a suitable option)
	Variation of the topple test (UN/ECE Reg 66)
	<p>A variation of the topple test is to carry out the UN/ECE Reg 66 test. This is the same as the topple test except the impact surface is 800 mm lower creating a high rotation rollover.</p>  <p>This test is probably not suitable for tanker testing for the following reasons:</p> <ol style="list-style-type: none"> 1. US research (examined by Part A of this research, Accident Analysis Group, 2002) states that high rotation rollover is less common than 90 degrees rollover. So, the scenario is not representative of most rollover accident cases. 2. The impact area would be close to the top of the tank and the tank is quite likely to continue to roll onto its roof and possibly beyond. This may cause higher levels of damage, so if this was a pass/fail test, then improvements in design above what is currently required in ADR may be necessary. 3. As the impact area is different to the impact area in the validated model, the models from the previous work will not be suitable to use as benchmark models for future work. <p>An example of a test rig can be seen at http://www.cranfieldimpactcentre.com/r66-rollover-rig.html (see image below). However, note that this test rig is 8 m long with a maximum load of 15 tonnes. This would probably prevent 3-compartment tank sub-sections being used as they are likely to be over the load limit for this test rig. If just a 90° topple was being considered, this rig would not be suitable without modifications as a raised impact block would be needed to bring ground level up to rig level etc.</p> 

A.6 Guided ramp impact test (RAMP 2)

Designing a rail trolley to attach under the tank to carry out a ramp impact test is the final option considered. The rail trolley travels along a railway track, so it is a guided impact test. Figure A-2 shows the tank and trolley parked on the ramp prior to the test.

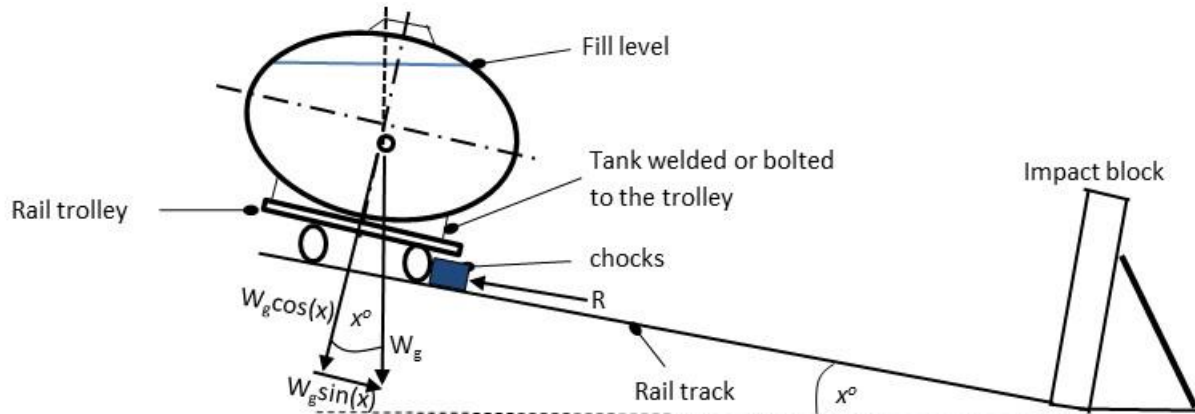


Figure A-2: Guided ramp impact test (tank in position at the beginning of the test)

W_g = the total weight of the tank and trolley

R = the reaction to the component of the weight that acts parallel to the ramp, $W_g \sin(x)$

Figure A-3 shows the trolley accelerating down the ramp under gravity towards the impact block (after the chocks were removed).

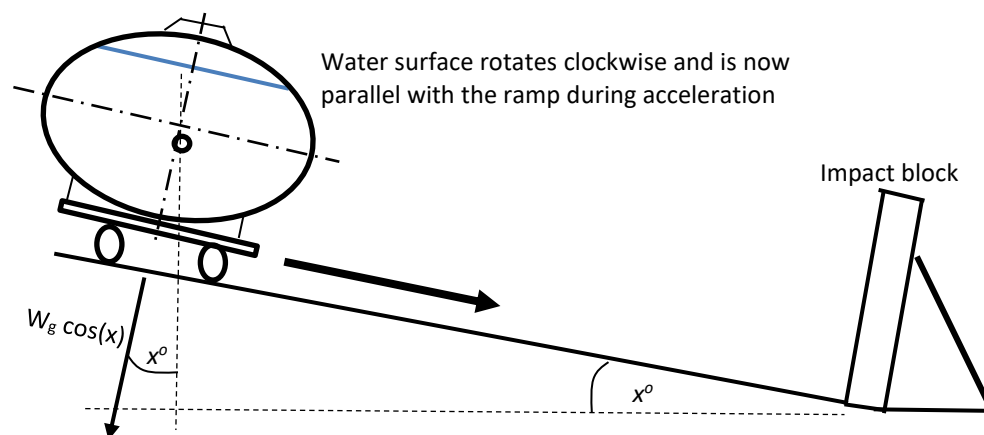


Figure A-3: Weight changes and water surface movement in the tank due to acceleration (trolley in motion)

The reaction (R) disappears, and the trolley begins to accelerate down the ramp at an acceleration of $g \sin(x)$. The static weight reduces and changes direction slightly to the component that is normal to the ramp ($W_g \cos(x)$). The water rotates so the surface is now parallel with the ramp, but in practice the surface will probably oscillate due to vibration and viscous effects may mean this change in condition is not fully achieved. Also, if the angle is steep, the weight component $W_g \cos(x)$ will be low. This means the movement of the water to its new position will be slower and the acceleration of the tank down the ramp will

be greater as $g \sin(x)$ is greater. Therefore, the water will probably not have time to move to its new position before impact.

Assuming an impact speed of 4.5 m/s (as for the drop test), the ramp height compared with ramp angle is shown in Figure A-4.

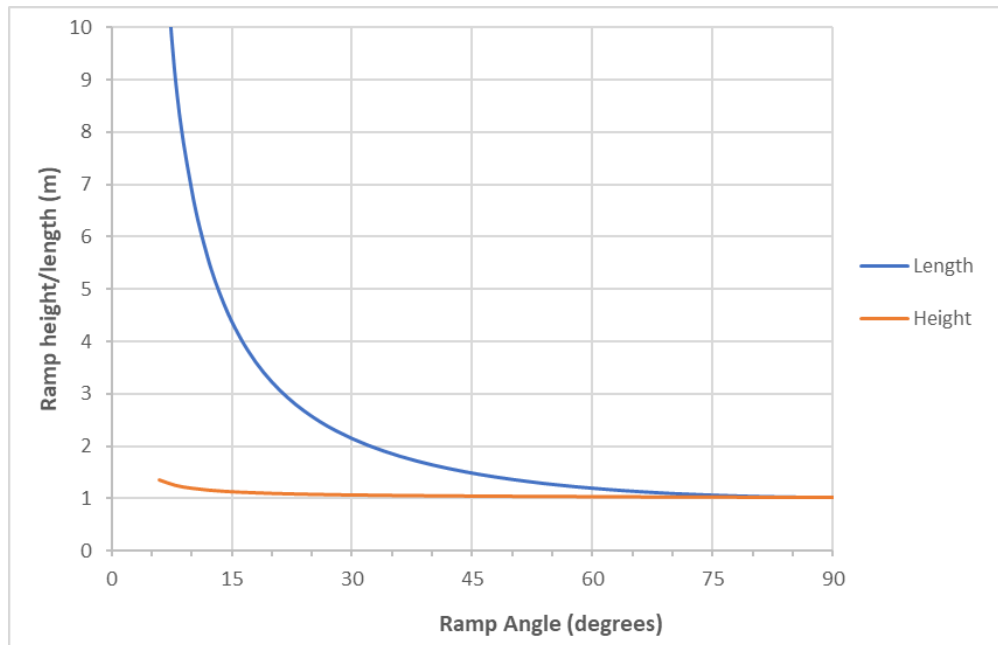


Figure A-4: Ramp height and ramp length as a function of ramp angle

A ramp angle of 10° to 25° would give a *theoretical* ramp length of about 7 m (10°) reducing to 3 m (25°). For ramps steeper than 28° the length will be shorter, but the tank will become increasingly unstable and will tend to roll forward as shown in Figure A-5.

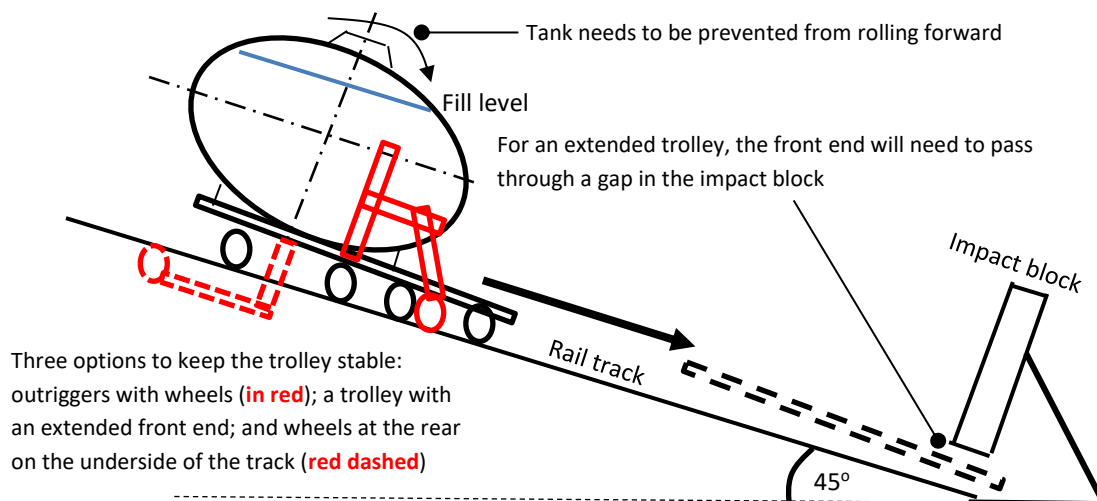


Figure A-5. Ramps steeper than 28° (trolley in motion)

Three options for stabilising the tank are shown in Figure A-5:

- front outriggers; and
- an extended trolley at the front
- rear wheels (on the underside of the track)

A 45° impact was considered in this work. Modelling has already shown that a vertical impact of 4.5 m/s in a drop or topple test causes greater damage than a horizontal impact at 4.5 m/s. This is due to the effect of gravitational force acting in the vertical direction. The 45° impact test is in between these two extremes.

However, for a 45° ramp angle, the length of the ramp can be shorter and trolley acceleration will be greater than it is for lower ramp angles. Unlike the drop and topple options, the water will still be supported by a weight component at impact as described above. The weight of the tank and trolley will be acting in the direction normal to the ramp immediately before impact but will be reduced to about 70% of the static weight ($1/\sqrt{2}$) in the case of a 45° ramp¹⁴. However, as the aim of the work is to achieve an equivalent level of damage to the damage in a topple scenario, these issues may not be of concern.

Table A-6: Guided ramp impact test (RAMP 2)

Tank Full (testing with the tank empty is not considered a suitable option)	
Advantages	<p>It may be possible to use this rig for a dynamic impact test on the kingpin, but this would depend upon the impact speed required for the kingpin impact. If it was much higher than 4.5 m/s, a longer track would be needed; or a method of propulsion for the tank would be required which is not thought to be practical (see below). Also, the track gauge and length for one impact scenario may not be suitable for the other as tank width could be very different in each scenario (head-on in one scenario: side-on in the other). Additional rails may be needed, and also the attachment plates between trolley and tank would be different for each of these impact scenarios.</p> <p>Therefore it is not certain that the same rig for the rollover and front impact scenario could be used. If a front impact test was to be carried out on the same rig, then the rig should be designed for both at the same time so all design factors can be taken into account (if it is decided that this is the most practical method for rollover). This would require full knowledge of the requirements of the frontal impact tests prior to the design stage for the rig commencing.</p> <p>Propulsion methods are:</p> <p>Acceleration from a hydraulic ram; acceleration from bungee cords; and engine attached to the rear of the tank</p> <p>All these methods would require a significant increase in test time and equipment cost and would be highly unlikely to be accepted in a standard test method. In practice it would be more suitable to build a separate test rig with the right track gauge and length for a front impact test.</p>
Disadvantages	<p>The rig is likely to be more costly than the drop test option.</p> <p>The trolley will have to be relatively large and strong to ensure it is not damaged from the repeated impacts.</p> <p>The trolley wheels and bearings (see below) may need to be specified in the test method as the mass and stiffness of the trolleys are a variable in the test. The trolley chassis may not need to be specified, but its maximum and minimum mass will need to be stated as this forms part of the impact energy.</p> <p>If an angle over 25° is specified, then the trolley will need to be stabilised to stop the tank from rolling forward</p> <p>The effects of small variations in friction in different wheel bearings at different test facilities could be significant in altering the impact velocity. As impact energy is proportional to the square of impact velocity then this could be important. Therefore a ramp angle AND an impact velocity may need to be specified. A flat section or low gradient of track may be needed at the top of the ramp to get the trolley moving slowly and overcome static friction in the wheels before the test run commences. This will improve the accuracy of obtaining the correct impact velocity. This is not an issue in a drop or topple test.</p> <p>The tank will need to be welded or bolted to the trolley. The tank would then need to be removed from the trolley at the end of the test. This will require some extra preparation between tests.</p> <p>Also the tank and trolley may need to be lifted off the ramp after the test before this is done as there could be a potential safety issue having technicians working to remove the tank when it may not be in a completely stable position immediately after the impact.</p> <p>If further height is needed during the testing to increase the impact energy, and the track has not been built for it, then significant modifications would be needed to increase the length of the track, or the tank would need to be 'pushed' onto the target to give a higher impact velocity. For a drop test it is a simple case of raising the tank a bit higher using the hoist.</p> <p>This method becomes increasingly challenging for 2 or 3 compartment samples as the width of the sample is much greater.</p>

¹⁴ This effect will be 'switched on' in the model at the start of the run (just before impact).

Appendix B Detailed assessment of liquid motion in topple test

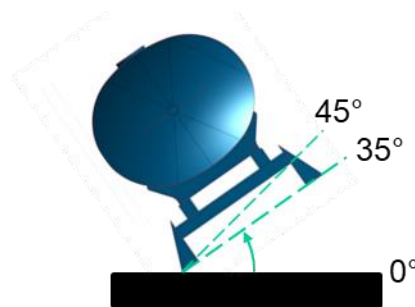
B.1 Objective and methodology

To inform the research team on whether the topple test method shows similar sensitivity to liquid position and liquid motion as the slide method.

Evaluated using a two-step approach:

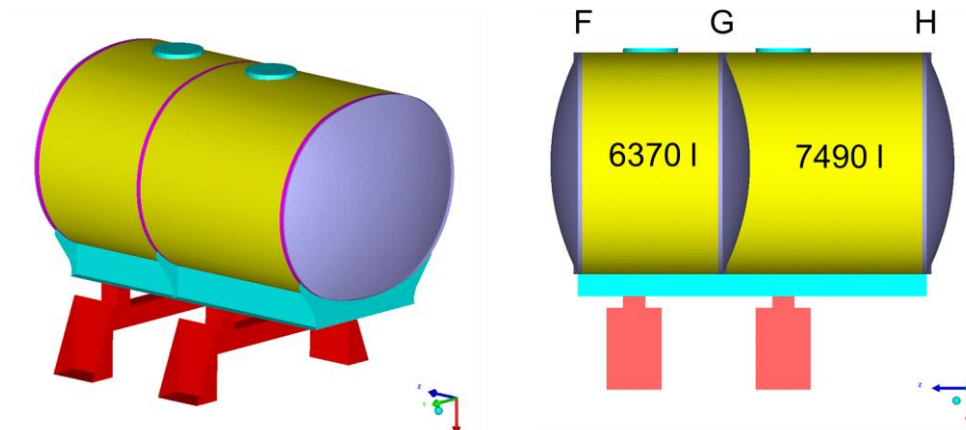
Step 1 – Liquid behaviour model to estimate the liquid angle and liquid motion at impact.

- Single compartment.
- Rigid tank (for 1st part of topple, explained below).
- Support represented by fixed pivot line at the outer edge of the steel wheels, and point mass.
- As initial movement very slow, liquid behaviour model started at 45°.
- Angular rotation at 45° established using completely rigid model from 35° (topple point) to 45° (this saves on run time).



Step 2 – Structural response model to establish sensitivity of topple test to variation in liquid angle and liquid motion at impact.

- Two compartments. Modelling 2 compartments (compared to complete tanker) saves time and still enables evaluation of sensitivity.
- Models started just before impact.
- Assumed liquid behaviour based on liquid behaviour models in step 1. This is varied to assess sensitivity.

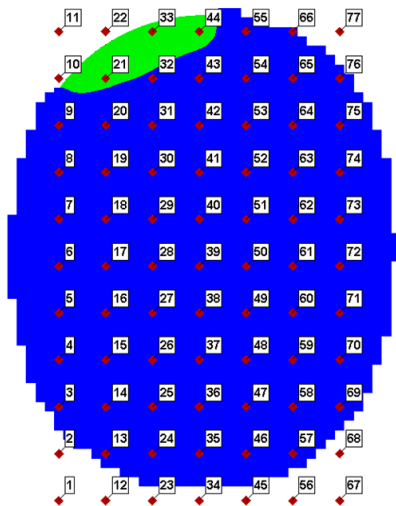


B.2 Results – petrol models

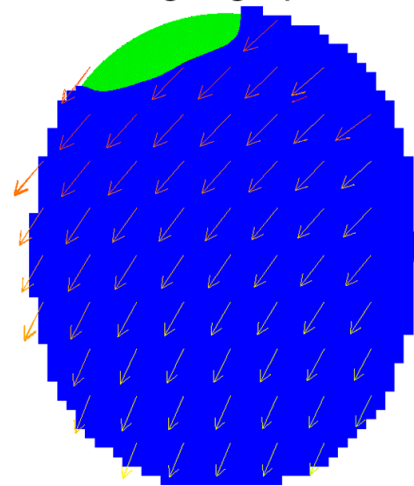
B.2.1 Step 1 – Liquid behaviour (petrol)

Image shows tanker at impact:

Location of gauge points



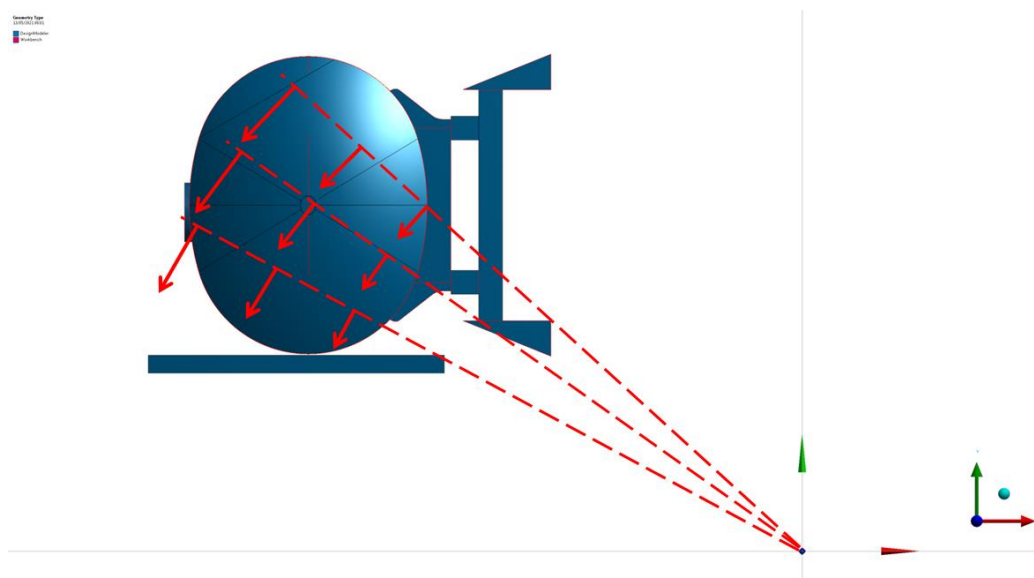
A velocity vector is obtained at each gauge point



Three possible approximations can be derived from the step 1 model. These can be assessed by comparing errors in liquid motion of each approximation, and the actual results obtained from the step 1 model:

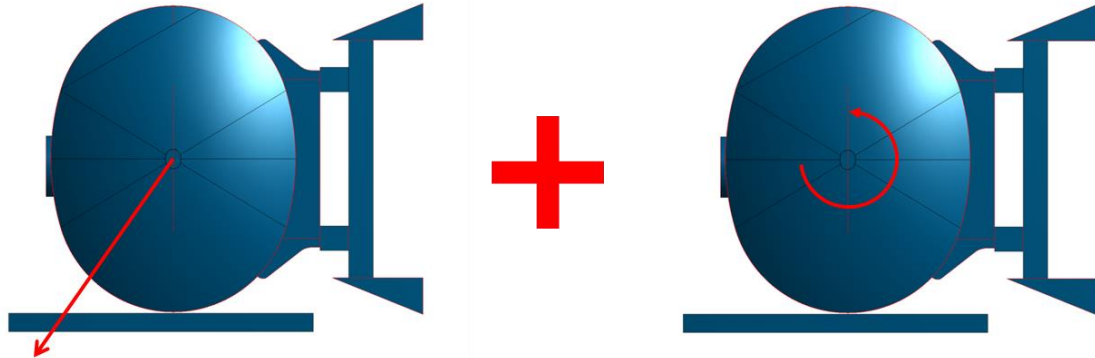
a. Rotation about different centre (used in the previous work).

Centre found by assuming angular velocity and tracing back from each gauge point based on angle and magnitude of point velocity and taking average. Angular velocity found by minimising errors (actual velocity – predicted velocity).



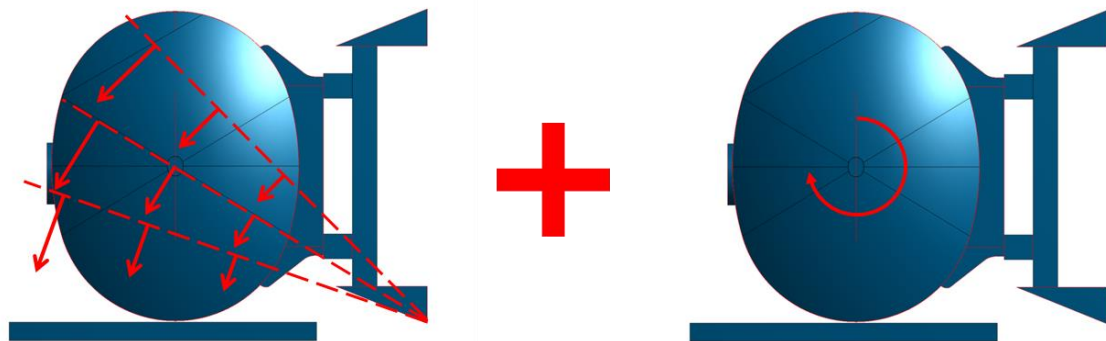
b. Translation (average) and rotation about centre of the tank.

Translation velocity taken as the average for the petrol (given as single figure by the software). Angular velocity of rotation found by minimising errors.



c. Rotation about pivot line (as for tanker) and rotation about centre of the tank.

Angular velocity of the rotation about the wheel pivot line taken to be the same as the tanker. Angular velocity about centre of tank found by minimising errors.



Variables in each of the three approximations obtained by minimising the sum of the errors squared or x and y components at each gauge point.

$$\Sigma (\text{actual velocity} - \text{predicted})^2$$

Found that b) translation plus rotation gave the lowest error value (5.6) closely followed by a) single rotation about different centre (6.2). Pivot rotation and tanker rotation approximation (c) was less accurate (8.6).

Therefore, approximation b) (translation + rotation) is the one used in the Step 2 models:

- Translation: X= -2.59 m/s, Y= -3.72 m/s.
- Rotation about centre of tank: $\omega = 0.91$ rad/s.
- Surface angle of liquid estimated to be approximately 24°. Angles of 20° and 30° used in Step 2 models (10° range roughly centred on approximation).

B.2.2 Step 2 – Subsection topple models (structural response, petrol)

Model 1:

- Translation and rotation of petrol (approximation b)
- 20° liquid surface angle

Model 2:

- Translation and rotation of petrol (approximation b)
- 30° liquid surface angle

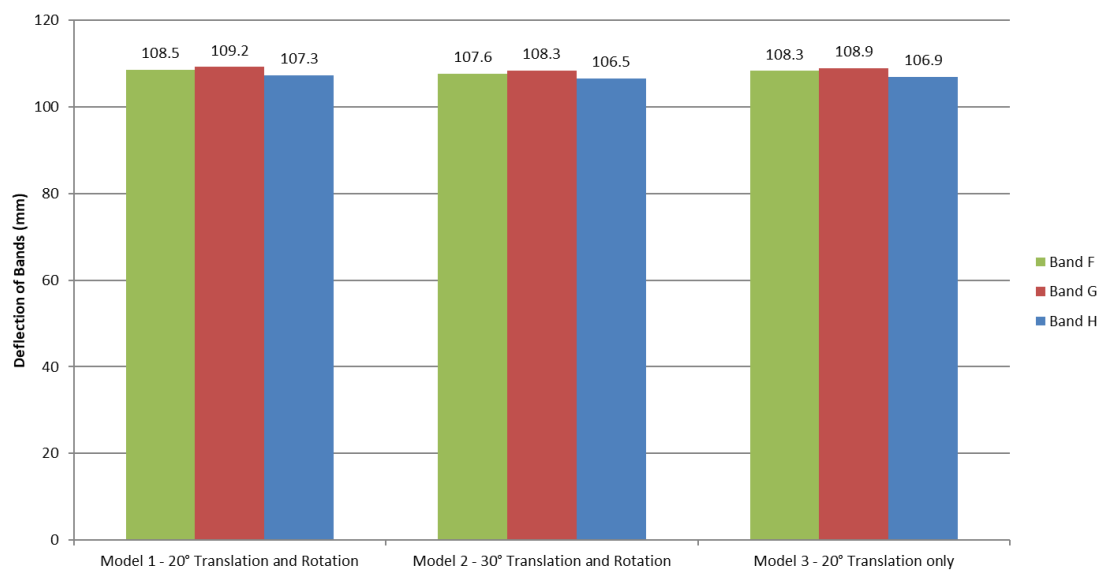
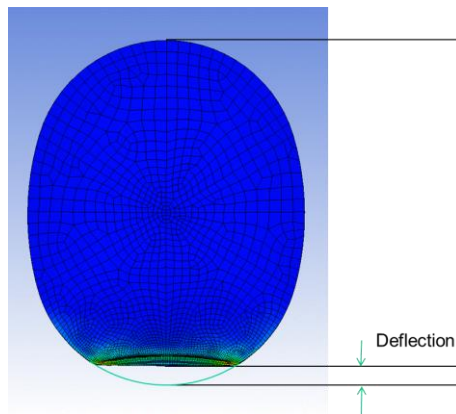
Model 3:

- Translation only (no rotation)
- 20° liquid surface angle

Band deflections (2 compartments, petrol)

Very little variation for either assumed angle of petrol surface, or petrol motion (<1 %).

All deflections well below benchmark deflections from the complete tanker model (108 mm for 2 compartments compared to 148 mm for complete tanker).



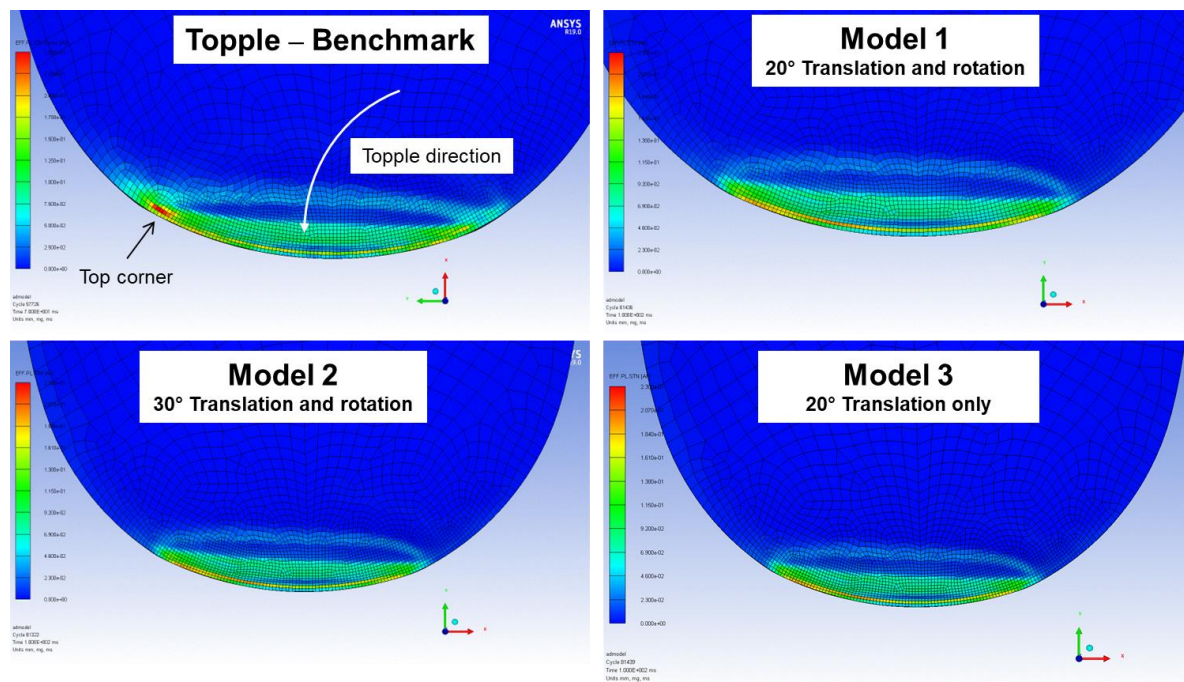
NB: Benchmark = 148 mm

Maximum plastic strain (bulkheads, petrol)

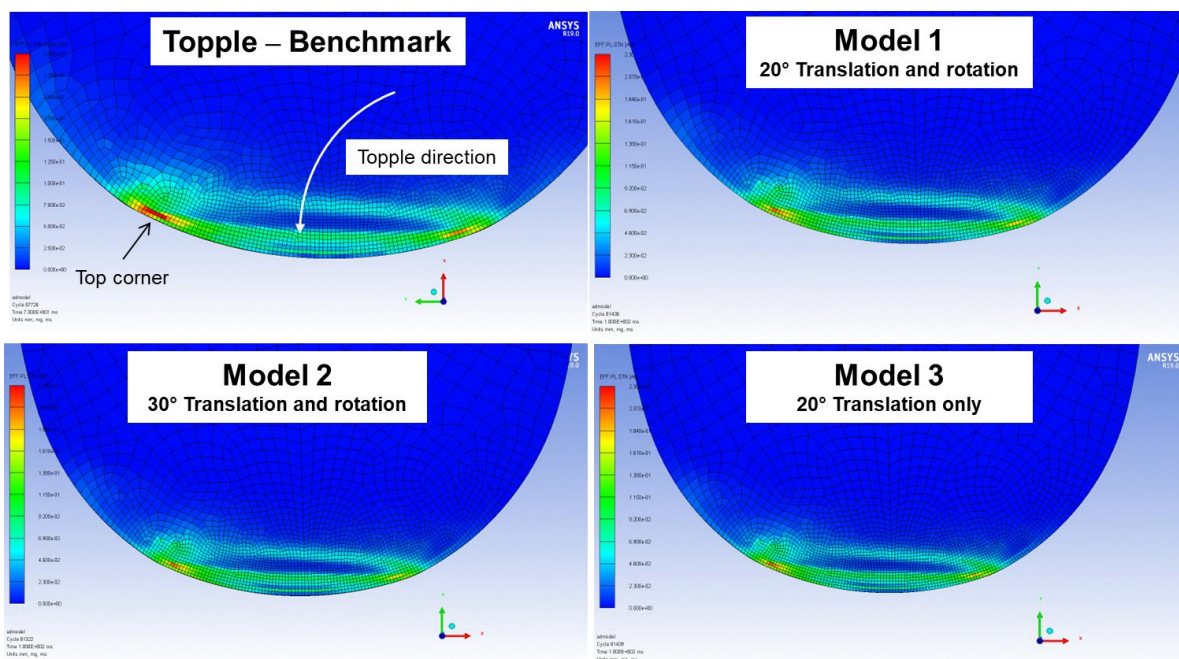
Maximum plastic strains (occurring in bulkhead G):

Benchmark	28%
Model 1 - 20°	20.7%
Model 2 - 30°	20.2%
Model 3 - simplified	20.9%

Note: in following images, strains shown on undeformed body as otherwise hidden in folds.



Plastic strains – Rear bulkhead (H)



Plastic strains – Bulkhead at band G

Plastic strains show similar pattern to full scale topple (benchmark) but with lower values. Very consistent behaviour between two-compartment models (not sensitive to liquid angle or precise liquid movement).

B.3 Results – water models

Both liquid behaviour (Step 1) and structural response (Step 2) models repeated for water. Same volume of liquid used (equivalent volume rather than equivalent mass).

B.3.1 Step 1 – Liquid behaviour (water)

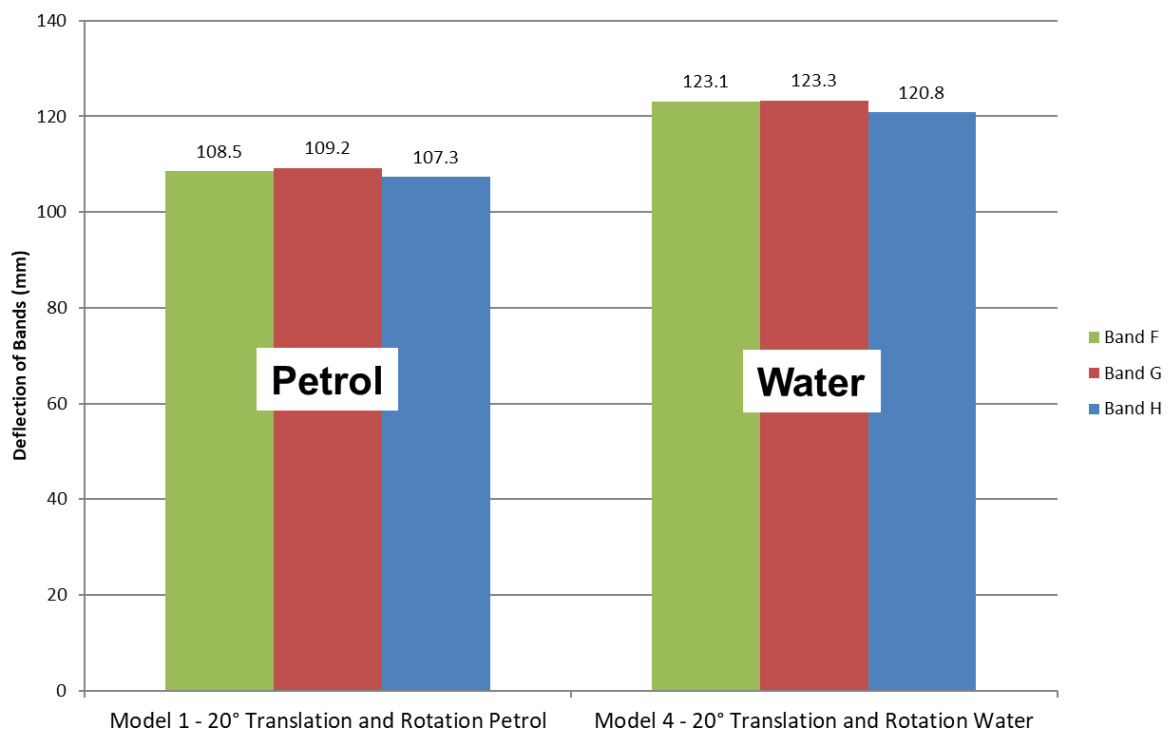
- Translation: X= -2.60 m/s (petrol -2.59 m/s), Y= -3.72 m/s (petrol -3.72 m/s).
- Rotation about centre of tank: $\omega = 0.89$ rad/s (petrol 0.91 rad/s).
- Angle estimated to be approximately 24° (same as for petrol).

B.3.2 Step 2 – Subsection topple models (structural response, water)

Model 4:

- Translation and rotation of water (approximation b above).
- 20° liquid surface angle (allows direct comparison to Model 1).

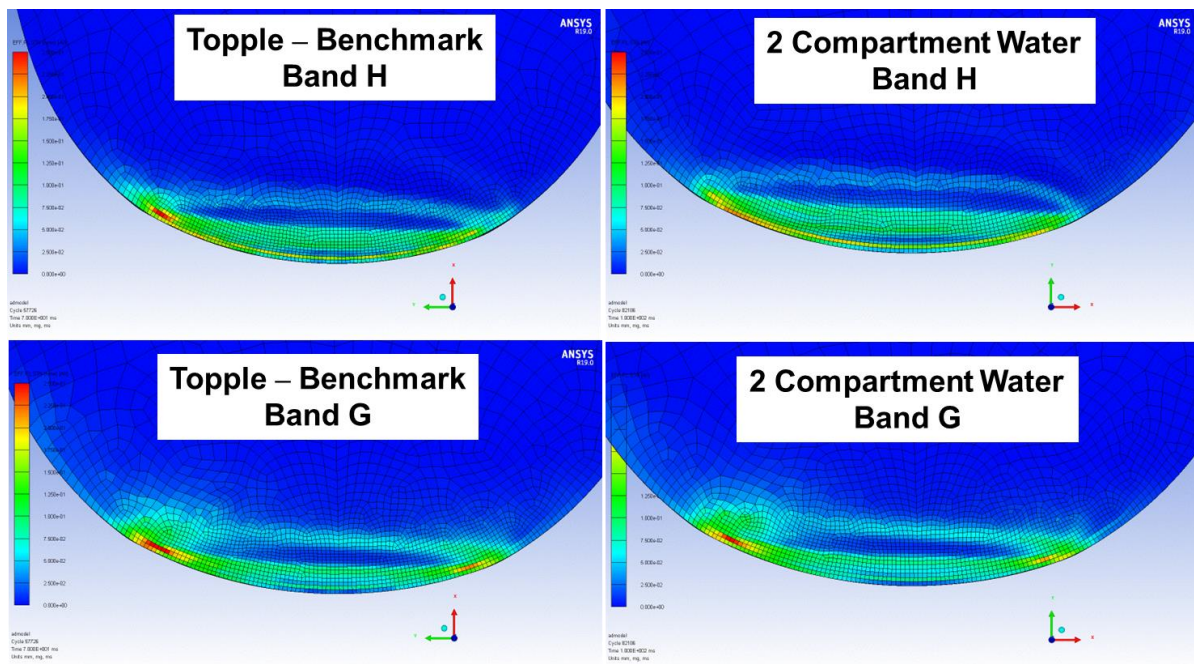
Band deflections (2 compartments, water)



Structural response model shows higher deflections than petrol (123 mm vs 109 mm) but still lower than benchmark (148 mm).

Maximum plastic strain (benchmark vs water, 2 compartments)

Plastic strains also higher than petrol (25 % water, 21 % petrol) but still lower than benchmark (28 %).



B.4 Overall findings

- Liquid behaviour complex (more so than drop) but can be approximated.
- Assuming translation plus rotation about tank centre or rotation about different point both give good approximations.
- Water and petrol liquid behaviour are similar.
- Structural response not sensitive to liquid angle or precise movement of liquid.
- Deformations and strains lower than the petrol benchmark, even using water by equivalent volume.

Appendix C Detailed drop test modelling of tank subsections

A parametric study was instigated to:

- Conduct further subsection drop test modelling to establish the correlations between key test variables and assess whether these correlations make this test method approach viable, and;
- If the drop test method approach is shown to be viable, to provide initial variables/conditions for test method design/development.

Further details of the study and its results are presented here.

C.1 Modelling Approach

The values used for each test parameter (numeric factor) are shown below:

- Compartment length – 1.85 m, 2.93 m and 4.00 m
- Liquid density – 0.75 kg/l, 0.88 kg/l and 1.00 kg/l
- Liquid fill level – by volume – 0.70, 0.83 and 0.95 of compartment volume
- Impact velocity – 4 m/s, 6 m/s and 8 m/s

These were combined into a series of *model-runs*, which are shown below in Table C-1. There are 25 runs: each run was a unique combination of the four test parameters above.

Table C-1: Number of model-runs carried out for the Step 1 modelling work

RUN	Length of Compartment (m)	Fluid Density (kg/litre)	Fill Proportion	Impact Velocity (m/s)
1	1.85	0.88	0.70	6.00
2	1.85	1.00	0.83	6.00
3	1.85	0.75	0.83	6.00
4	1.85	0.88	0.83	4.00
5	1.85	0.88	0.83	8.00
6	1.85	0.88	0.95	6.00
7	2.93	1.00	0.70	6.00
8	2.93	0.88	0.70	4.00
9	2.93	0.75	0.70	6.00
10	2.93	0.88	0.70	8.00
11	2.93	0.75	0.83	4.00
12	2.93	1.00	0.83	4.00
13	2.93	0.88	0.83	6.00

RUN	Length of Compartment (m)	Fluid Density (kg/litre)	Fill Proportion	Impact Velocity (m/s)
14	2.93	0.75	0.83	8.00
15	2.93	1.00	0.83	8.00
16	2.93	0.88	0.95	8.00
17	2.93	0.75	0.95	6.00
18	2.93	1.00	0.95	6.00
19	2.93	0.88	0.95	4.00
20	4.00	0.88	0.70	6.00
21	4.00	0.75	0.83	6.00
22	4.00	0.88	0.83	8.00
23	4.00	0.88	0.83	4.00
24	4.00	1.00	0.83	6.00
25	4.00	0.88	0.95	6.00

Prior to the step 1 work, it was not clear if all 25 runs would need to be repeated for steps 2 to 4. This was found not to be the case once the step 1 results were analysed (discussed later).

Therefore, Table C-1 is only relevant for step 1. Fewer runs were completed in subsequent steps.

The impact of the tanker on the ground during a drop test creates two main reactions: a 'ground reaction force'; and forces arising from pressure in the liquid (Figure C-1).

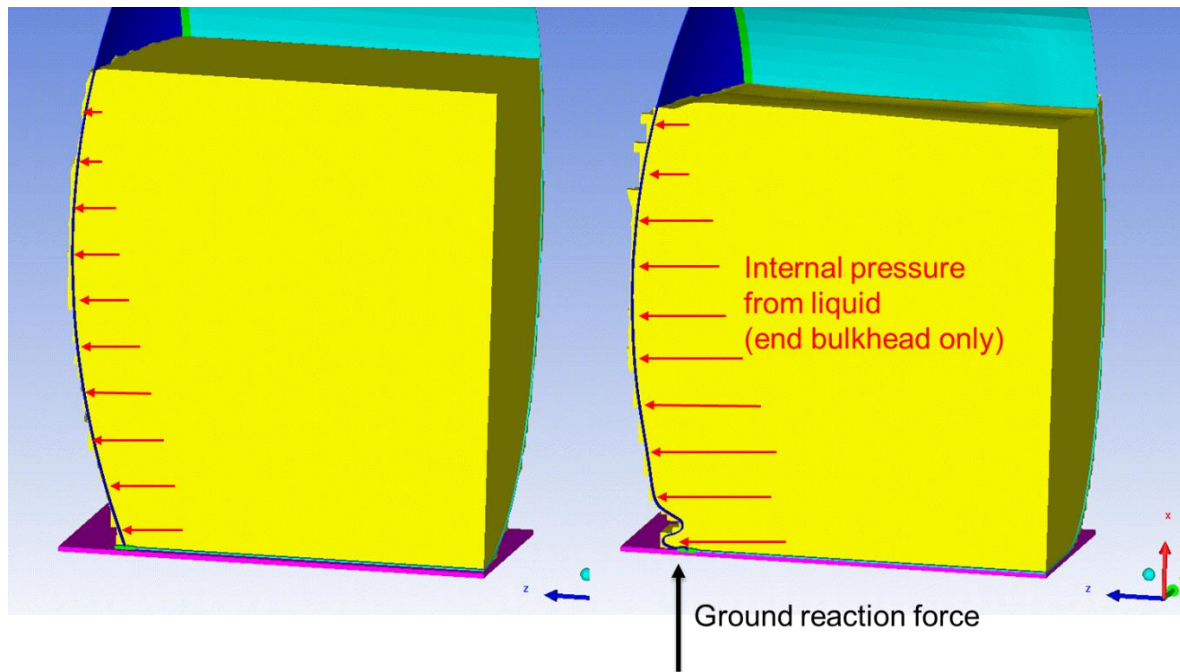


Figure C-1: Main characteristics of impact event

The pressure in the liquid will depend on the liquid depth and the deceleration on impact. The pressure will act to force the bulkhead outwards as shown in Figure C-1. This has been referred to as the *pressure force* in this work.

If these two reactions are similar between models of the same tanker design but with different numbers of compartments, then the structural response (and likelihood of failure) should also be similar.

The structural response parameters that were modelled are:

- deflection (amount of crush);
- plastic work;
- plastic strain; and
- forming limit factor (the 'omega value').

The regression analysis carried out by TWI used the Design of Experiments (DoE) software DesignExpert, and the following regression analyses were performed:

- Results from FSI models were provided as tables of "numerical factors" and "responses".
- Analysis of variances (ANOVA) and regression analyses were performed to generate response surface functions.
- Combinations of numerical factors were assessed to determine which were meaningful.

C.2 Results

In this section the modelling results for each of the four steps are shown. This also includes some explanation of why 3D solid modelling elements were added to the 2D shell element methods, and the options that were considered when it was recognised that shell elements alone were insufficient.

C.2.1 *Step 1 – Two band, one compartment Banded type 1 subsection*

The results are shown for each of the three outputs mentioned in the previous section.

Deflection

For the deflection results, TWI obtained excellent correlation in the regression analysis by finding quadratic equations (a 'lower order polynomial'¹⁵) that generated the response surfaces. The analysis showed that the only numeric factors that were needed were length and drop height; and impact energy could be controlled by adjusting the drop height. This meant that the other numeric factors (fill level and fill volume) could be ignored as they were found to be variations of impact energy alone (i.e. they had no additional effect on the tanker response).

Therefore, the number of runs for the next steps could be reduced from the 25 used here.

Figure C-2 shows the response surface for deflection as a function of the numeric factors of impact energy and compartment length. This surface had a coefficient of determination (R^2)¹⁶ of 0.999.

¹⁵ A 'polynomial' is an expression of more than two algebraic terms that contain different powers of the same variable. An example of a quadratic polynomial is $4x^2 + 3x + 1$

¹⁶ R^2 is the 'coefficient of determination'. The closer it is to 1, the stronger the relationship is between the independent variables and dependent variables (in this case the independent variables are energy and length of compartment, and the dependent variable is deflection).

Factor Coding: Actual

Deflection (mm)

Design Points:

● Above Surface

○ Below Surface

49.8871 313.86

X1 = A

X2 = E

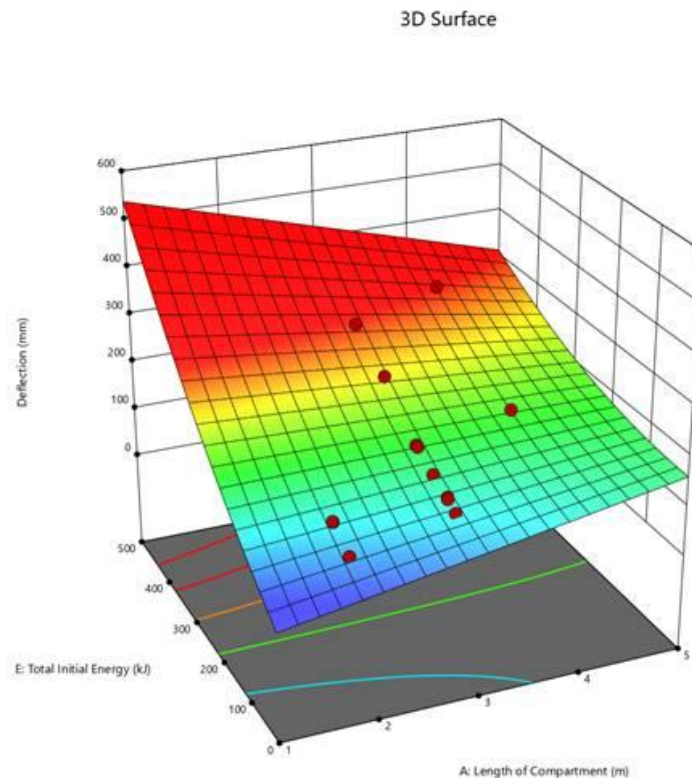


Figure C-2: Response surface for deflection values

The red points are the data points obtained from HSE SD's FSI models. The regression analysis links all these points together on the response surface. This does not mean that each of the red points lies exactly on the response surface, in the same way that a regression curve of data points on a two-dimensional graph do not pass through every point – the response surface is a 'best fit'.

Plastic work in the bulkhead

For plastic work in the bulkhead, a strong correlation against length and impact energy was also found in the regression analysis with an $R^2 = 0.989$. (Figure C-3).

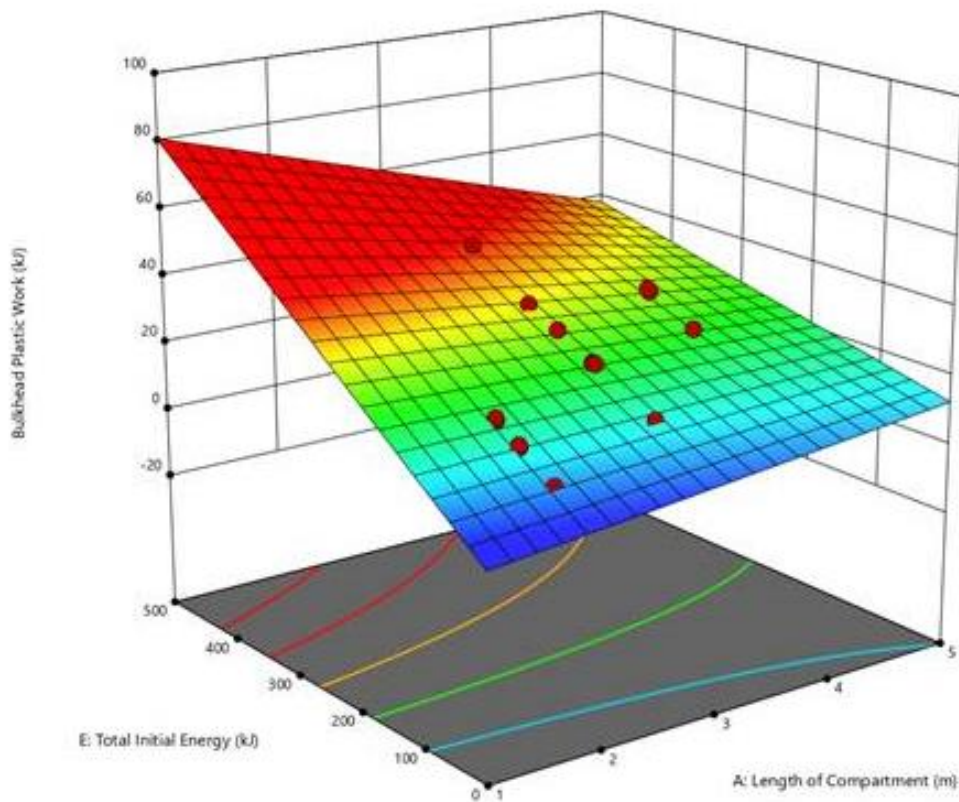


Figure C-3: Response surface for plastic work

Plastic strains and omega

The omega (forming limit) values were found to be above one for all models with very little variation with different numeric factors. It was decided to refine the mesh, re-run the models, and assess the sensitivity of these values. However, when the models were re-run, there were large variations in the omega value, which indicated no mesh convergence.

Maximum plastic strain results also showed mesh sensitivity and little correlation. Therefore, the fluid structure interaction (FSI) shell model was unable to generate reliable, converged omega values. This was a consequence of the strain singularity that was developing at the area of high strain in the dish. Therefore, the shell theory was not applicable in this region. The local stresses (or more importantly strains) from the 2D shell model were not valid as they did not represent the true three-dimensional strain state.

A modified solid model – static implicit approach was selected to address this issue and is described as follows:

- Static model: at each step, model is in equilibrium.
- Displacement applied by moving the ground into the tanker.
- Coupling forces: The liquid pressure forces acting on the end bulkhead can be output from the HSE FSI models (single force value for the full bulkhead). Pressure applied as a hydrostatic head (with pressure varying with liquid depth) ramped up with displacement.

Omega Values

Two solid models were run with accelerations of 75 m/s^2 and 100 m/s^2 . These accelerations were selected as 75 m/s^2 is the lower bound to the coupling force in the benchmark topple, and 100 m/s^2 is around the values of the lower regression models. Figure C-4 shows omega as a function of deflection from these two runs.

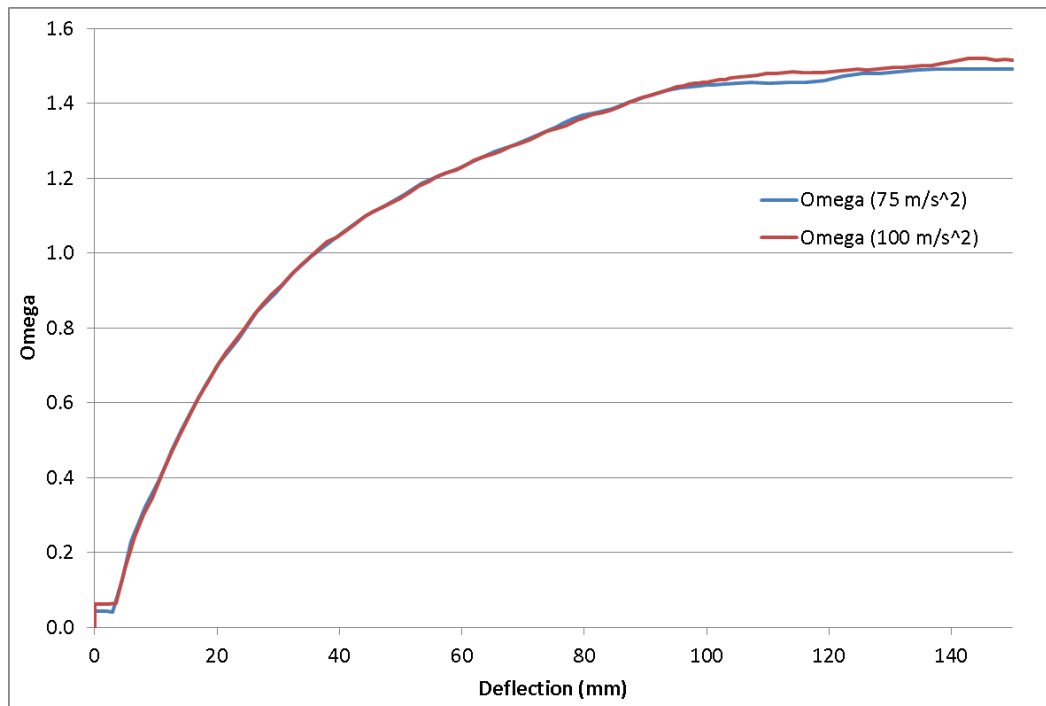


Figure C-4: Omega values using low and moderate pressure forces in the static models

There was negligible difference in omega values with around a 30% increase in acceleration (which also equates to a 30% increase in pressure force). Therefore, omega is insensitive to the pressure force in this range. Figure C-4 shows that omega reached a value of 1 (indicating the forming limit had been reached) at a deflection of around 40 mm.

Comparison to the benchmark topple model

To assess if the response from the single-compartment subsection model could demonstrate an equivalent level of damage to the benchmark topple model, it was necessary to compare the results from the subsection predictions with the benchmark topple predictions.

Benchmark petrol topple values from the eight-band banded type 1 tanker were as follows (for the rear-end bulkhead Band H):

- Deflection: 148 mm
- Plastic work: 16.2 kJ
- Coupling (pressure) force: 156 kN

These values can be set as a target in the regression software to find the best numeric factors (length and impact energy) for a single compartment model.

Matching just the deflection and the plastic work responses could be achieved with a range of different lengths and impact energies. However, all three of the above response values were required to give a unique answer (i.e. pressure force needed to match as well).

Figure C-5 shows the two numeric factors (red dots) that were required to give the three response values (blue dots) that were close to the values from the benchmark topple shown above.

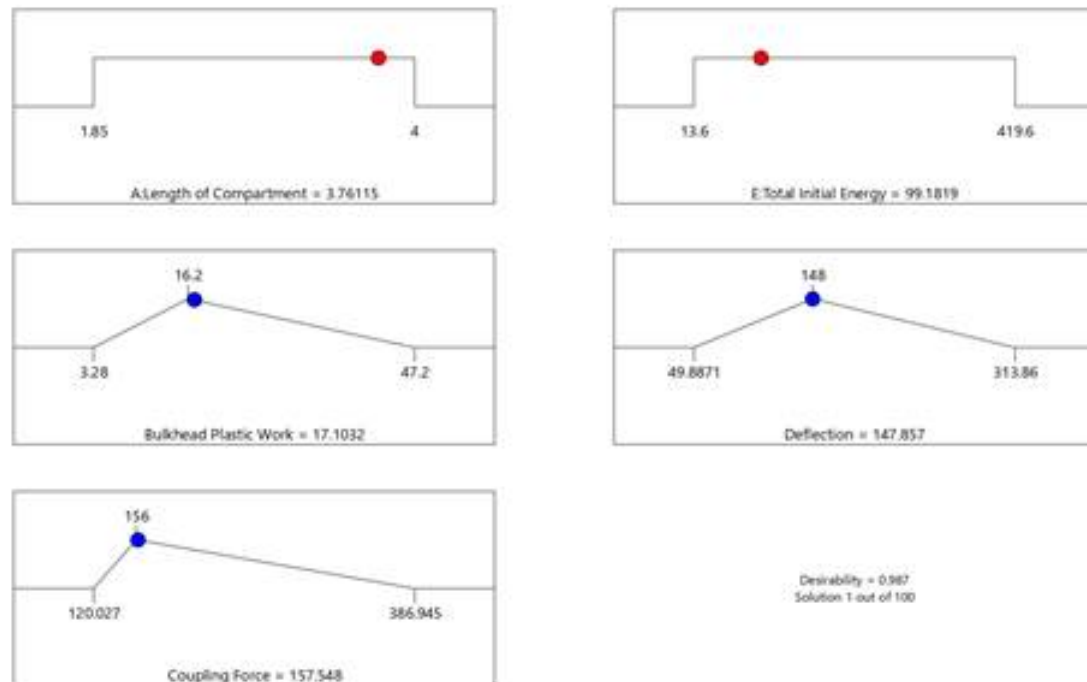


Figure C-5: Input parameters required to match the benchmark topple outputs

Therefore, to match the benchmark topple for deflection, plastic work, and pressure force, the analysis in the top two boxes in Figure C5 suggest that the numeric factors should be:

- Length = 3.8 m
- Impact energy = 100 kJ

The analysis showed that it did not matter what combination of liquid density, fill level and impact velocity are used to obtain energy (within the ranges modelled). This has practical importance as it would be difficult to carry out an impact test with any liquid other than water due to safety, environmental, and cost implications of using alternative liquids.

Step 1 Findings

For the identified test parameters (length = 3.8 m, impact energy = 100 kJ) assuming the test would use water (density = 1000 kg/m³) and medium fill level (0.83), the test parameters would equate to a tanker mass of 14 tonnes, and a drop height of 730 mm.

These could be altered slightly by changing the fill level without affecting the results (higher fill level would mean higher mass and lower drop height).

As the solid models showed that omega is insensitive to a 30% change in pressure forces, matching pressure forces may not be so important. Pressure forces are likely to be slightly

higher for a shorter compartment or a higher drop height, but this would be unlikely to have a significant effect on omega within the range currently modelled¹⁷. This would lead to a wider range of candidate test parameters for the joint geometry used. This would need to be checked for other joint types.

The deflection and plastic work outputs can be matched with a range of lengths and impact velocities, which would allow for shorter, lighter subsections to be dropped from a greater height for the same deflection and plastic work.

This match is **only** for the banded type 1 benchmark topple to the banded type 1 one-compartment subsection. If the benchmark tanker had been a different design, and the plastic work had been different (e.g. more energy was converted to elastic energy than in the banded type 1 tanker, and therefore less energy was converted into plastic work), then this may have an influence on the test parameters that are set. Therefore, some knowledge of the variation in energy absorbed per partition for different tanker designs would need to be obtained before subsection testing could be considered for all types of metallic tanker design.

C.2.2 Step 2 – Three band, two compartment banded type 1 subsection

The model in step 2 is based on the same banded type 1 joint design from the single compartment model in step 1.

The findings from step 1 showed that some of the numeric factors could be fixed. This reduced the number of model-runs from the 25 that were used for step 1. The two test parameters for liquid density and fill level were fixed at 1000 kg/m³ (water) and 95% (0.95) fill proportion. These were chosen as they are practical for a drop test. The two test parameters that were varied to give different model-runs were compartment length and impact energy. As impact energy had been chosen rather than impact velocity, it meant that the results tended to cluster over a shorter range. The number of model-runs was reduced to 13 (Table C2).

Table C-2: Number of model-runs carried out for the Step 2 modelling work

RUN	Length of Compartments (m)	Fluid Density (kg/litre)	Fill Proportion	Impact Velocity (m/s)	Impact energy (kJ)
1	1.40	1.00	0.95	4.04	100
2	1.40	1.00	0.95	6.38	250
3	1.40	1.00	0.95	8.07	400
4	1.80	1.00	0.95	4.76	175
5	1.80	1.00	0.95	6.49	325

¹⁷ A check would need to be made if the pressure forces occurring in shorter compartment/higher drop scenarios were significantly higher than those already modelled.

RUN	Length of Compartments (m)	Fluid Density (kg/litre)	Fill Proportion	Impact Velocity (m/s)	Impact energy (kJ)
6	2.20	1.00	0.95	3.28	100
7	2.20	1.00	0.95	5.18	250
8	2.20	1.00	0.95	6.56	400
9	2.60	1.00	0.95	4.01	175
10	2.60	1.00	0.95	5.46	325
11	3.00	1.00	0.95	2.83	100
12	3.00	1.00	0.95	4.48	250
13	3.00	1.00	0.95	5.66	400

The main test outputs were:

- deflection;
- plastic work; and
- pressure forces (coupling forces).

As plastic strain is a function of omega, it was decided after step 1 that only the above three response parameters were necessary, and plastic strain has not been reported. Omega values were found to depend primarily on deflection for a given joint design. These are reported in later sections when comparing omega values for different joint designs. Figure C-6 shows deflection as a function of impact energy, and Figure C-7 shows plastic work as a function of impact energy.

Factor Coding: Actual

Deflection H (mm)
 ● Design Points
 — 95% CI Bands

X1 = D

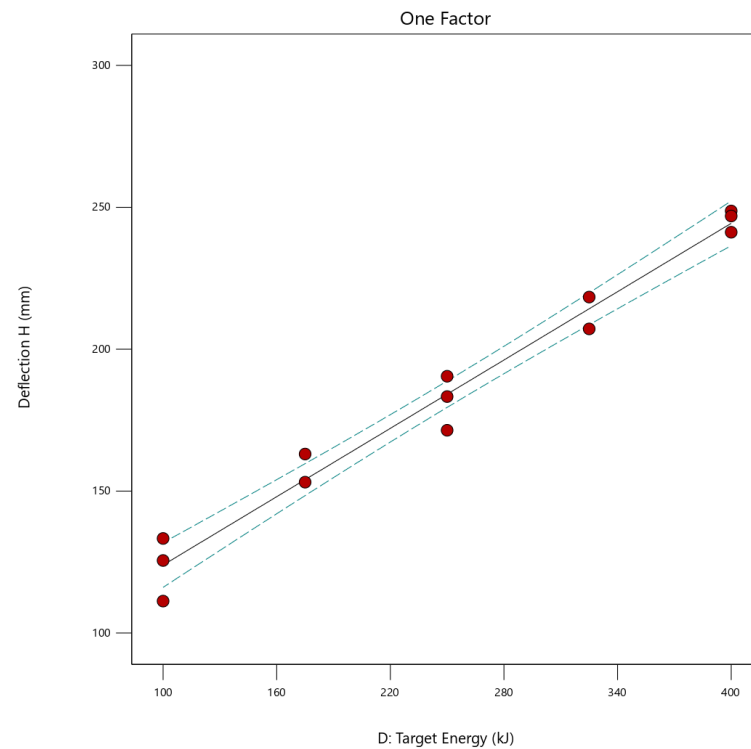


Figure C-6: Deflection response as a function of the target energy (impact energy) – two-compartment banded type 1 design

Factor Coding: Actual

Plastic Work H (kJ)
 ● Design Points
 — 95% CI Bands

X1 = D

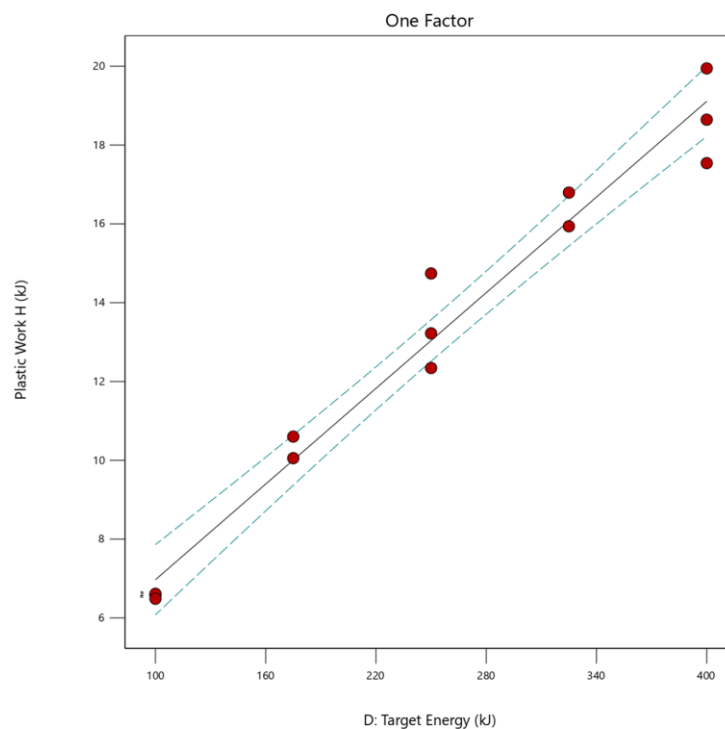


Figure C-7: Plastic work as a function of the target energy (impact energy) – two-compartment banded type 1 design

Note that these graphs refer to the letter 'H', which is a reference to band H (the rear band of an 8-banded tanker). In these figures, it is one of the end bands of the two-compartment subsection forming the end-bulkhead. It was noted that the plastic work in the end-bulkheads was higher than for the central partition; this additional plastic work was mainly in the extrusion rather than the dish (~35%). It appears to be caused by a slight twisting of the extrusion due to pressure bending the dish outwards. However, as this was also seen in the benchmark topple model, this effect is being replicated in the subsection model. This is supporting evidence for 'equivalence' between the two test methods.

Good correlation was found between deflection and impact energy, and also plastic work and impact energy. One factor correlation (just considering impact velocity) gave a coefficient of determination of $R^2 > 0.97$; and two factor correlation (adding the length factor) improves the fit ($R^2 > 0.99$).

The pressure forces were obtained from the model-runs for variations of the two numeric factors: compartment length and impact velocity. To get an indication of how the two-factor variations affect the pressure forces, a regression analysis was carried out on these pressure force values to obtain a straight-line through the points. The force values on this straight line were referred to as the *predicted force* (i.e. the pressure forces predicted at other values of the two numeric factors that were not modelled). The discrete values predicted from the models are referred to as the *actual force*.

Figure C-8 shows the predicted pressure force as a function of the actual pressure forces obtained from the model.

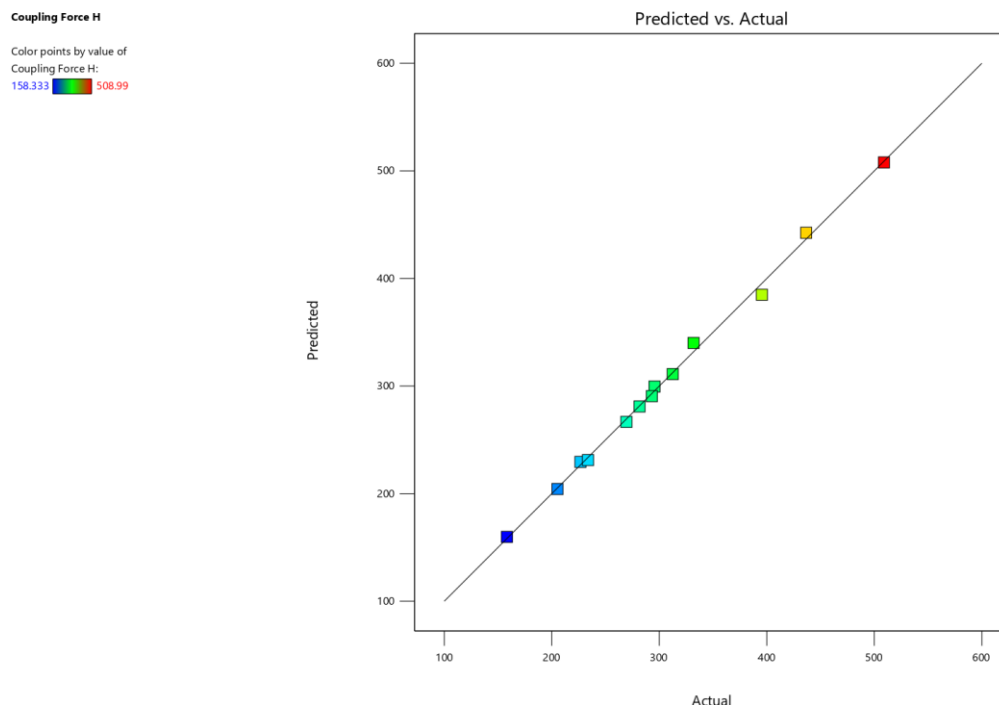


Figure C-8: Modelled and predicted pressure force

There is very little difference between the predicted and actual values. Therefore, the pressure force depends mainly on impact velocity with a small effect caused by variation in the compartment length.

Comparison to the benchmark topple model

The benchmark petrol topple values from the 8-banded type 1 tanker were as follows (form band H):

- Deflection: 148 mm
- Plastic work: 16.2 kJ
- Coupling (pressure) force: 156 kN

These values can be set as targets in the regression software to find the best numeric factors (length and impact velocity) for a double compartment model.

The numeric factors that give a response in the two-compartment model close to the above three response values are as follows:

- Impact velocity: 3.8 m/s
- Compartment length: 2.5 m

giving an impact energy of 150 kJ

Table C-3 compares the test parameters for the single compartment model from step 1 with the double compartment results from step 2.

Table C-3 Comparison of parameters between complete banded type 1 tanker and single and double compartment test pieces

	Single	Double	Full banded type 1 Tanker*
Impact energy	100 kJ	150 kJ	390 kJ
Impact velocity	3.8 m/s	3.8 m/s	4.3 m/s
Compartment length	3.8 m	2.5 m	1.9 m average (1.3 m to 2.4 m [†])

Care should be taken when comparing full tanker **topple to single and double compartment **drop** tests as they are not directly equivalent*

[†]The longest compartments (front 2 compartments) contained stiffeners

It was not possible to fully match the pressure force with the fill level fixed at 95%. If the fill level was reduced slightly and the length was increased, while keeping the overall mass the same, the pressure force in the double compartment model could be reduced to similar pressures as in the single compartment model. This would extend the length of each

compartment in the double-compartment model closer to the single compartment value in Table C-3.

The impact energy/partition (~50 kJ/partition) was similar for both single and double-compartment models. The compartments for both models are longer than the longest compartment in the banded type 1 tanker, and the impact velocity is slightly lower.

For the double compartment, the test parameters to achieve a representative test result to the full banded type 1 tanker would result in:

- Compartment length: 2.5 m (from Table 3)
- Total length: 5 m
- Structural mass: 700 kg
- Liquid mass: 20.3 tonnes (95% fill)
- Drop height: 730 mm

Comparing these results with a single compartment, this gives a longer overall length (3.8 m increased to 5 m); higher mass (14 tonnes increased to 21 tonnes) and the same drop height (730 mm).

For both single and double compartments, the mass and drop height can be altered (keeping the same energy) if pressure forces are deemed less important.

C.2.3 Step 3 – banded type 2 Design

A single compartment banded type 2 subsection was considered in step 3.

In reviewing the design drawings, a difference between the angle of dish edge and the support on the extrusion (~9°) was observed. Previous discussions between TWI and the banded type 2 manufacturer suggested that the dish is bent during fabrication to make it match the angle of the extrusion support. Sections taken from a banded type 2 tanker used in the previous topple test project at HSE showed a gap with a difference of angle of approximately 4°, indicating that the dish was not bent enough to fully match the angle of the extrusion support. Therefore, the two scenarios of a 'clean fit' and a 4° gap have been modelled as shown in Figure C-9.

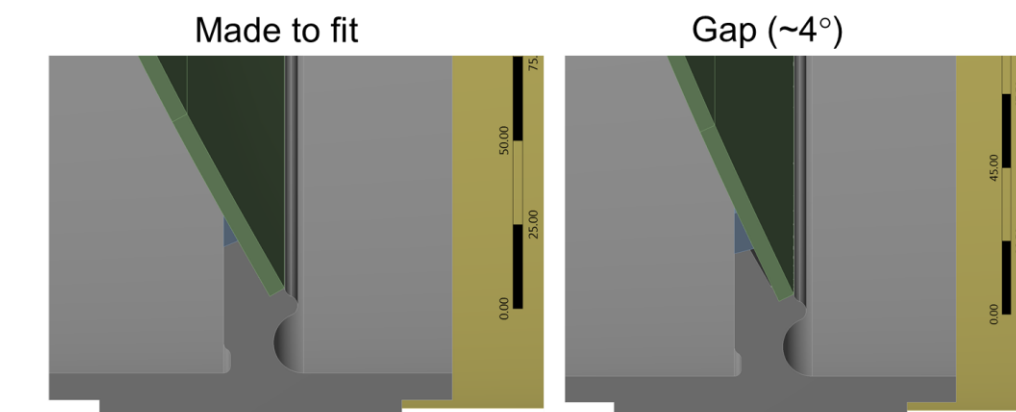


Figure C-9: Details of extrusion and dish geometry showing different assumptions about fit

For the modelling work, the two numeric factors of liquid density and fill level were fixed at 1000 kg/m³ (water) and 83% fill (mid-level); and as for step 2 the two numeric factors that were varied to give the different model-runs were compartment length (1.85 m to 4 m) and impact velocity (3 m/s to 6 m/s). The full list of model-runs is given in Table C-4.

Table C-4: Number of model-runs carried out for the Step 3 modelling work – banded type 2 Tanker

RUN	Length of Compartments (m)	Fluid Density (kg/litre)	Fill Proportion	Impact Velocity (m/s)	Impact energy (kJ)
1	2.93	1.00	0.83	4.5	112
2	2.93	1.00	0.83	3.0	50
3	2.93	1.00	0.83	6.0	200
4	1.85	1.00	0.83	4.5	75
5	1.85	1.00	0.83	3.0	33
6	1.85	1.00	0.83	6.0	133
7	4.00	1.00	0.83	4.5	149
8	4.00	1.00	0.83	3.0	66
9	4.00	1.00	0.83	6.0	265
10	2.93	1.00	0.83	3.8	78
11	2.93	1.00	0.83	5.3	153
12	2.39	1.00	0.83	4.5	94
13	2.39	1.00	0.83	3.8	65
14	2.39	1.00	0.83	5.3	127
15	3.46	1.00	0.83	4.5	131
16	3.46	1.00	0.83	3.8	91
17	3.46	1.00	0.83	5.3	178

As the assumptions made about the fit of the dish to the extrusion (whether made-to-fit or with a 4° gap) may affect the results, it was decided to assess the effect that this has on omega before doing all of the model-runs.

Figure C-10 shows omega as a function of deflection for the fully-fitted design (made-to-fit design); and the 4° gap design.

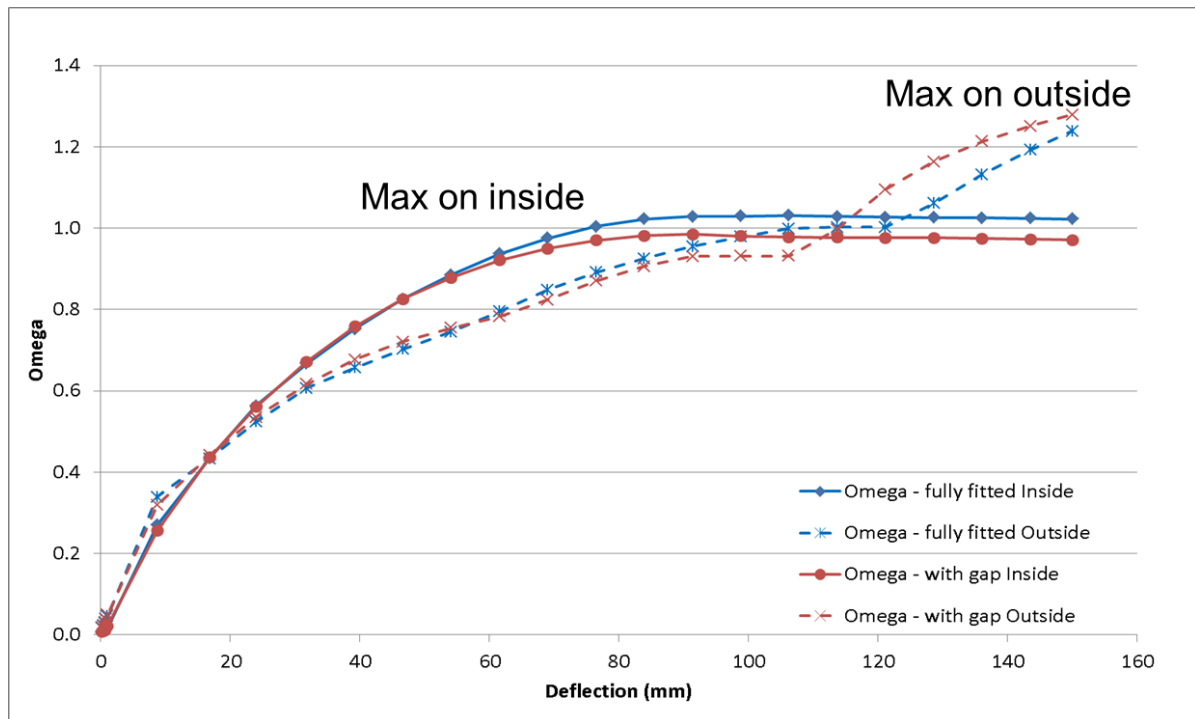


Figure C-10: Omega values on the inside and outside surfaces of the end dish for both models (fully fitted dish, and with a small gap remaining between the extrusion and the dish) – banded type 2 tanker

It was observed that the locations of the highest omega values change as the deflection increases. At low deflections, the highest omega values occur on the inside of the dish where the dish bends outwards around the extrusion (Figure C-11).

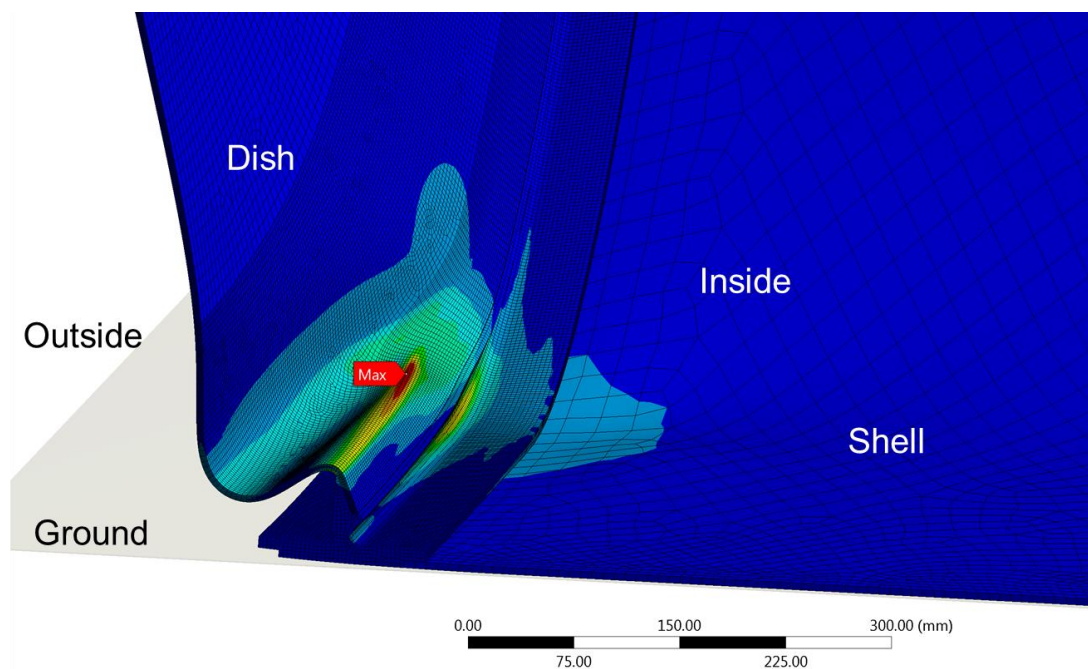


Figure C-11: Location of the maximum omega values at low levels of deformation – banded type 2 tanker

These omega values reach a maximum at a deflection of approximately 80 mm and then stay constant with further deflection.

At higher deflections (over approximately 120 mm in this case), the omega values on the outside of the dish become higher than those on the inside. The highest values occur close to the end of the flat formed by the impact, as shown in Figure C-12. It was noted in TWI's previous investigation of ruptures of the end dish that were experimentally observed for the banded type 1 tankers, that the rupture initiated towards the end of the flat. A similar mechanism appears to be occurring here for the banded type 2 subsection. This is because the region near the centre of the flat accommodates more of a biaxial strain-state, whilst towards the end of the flat, the structural and geometric constraint limits the forming limit strain¹⁸.

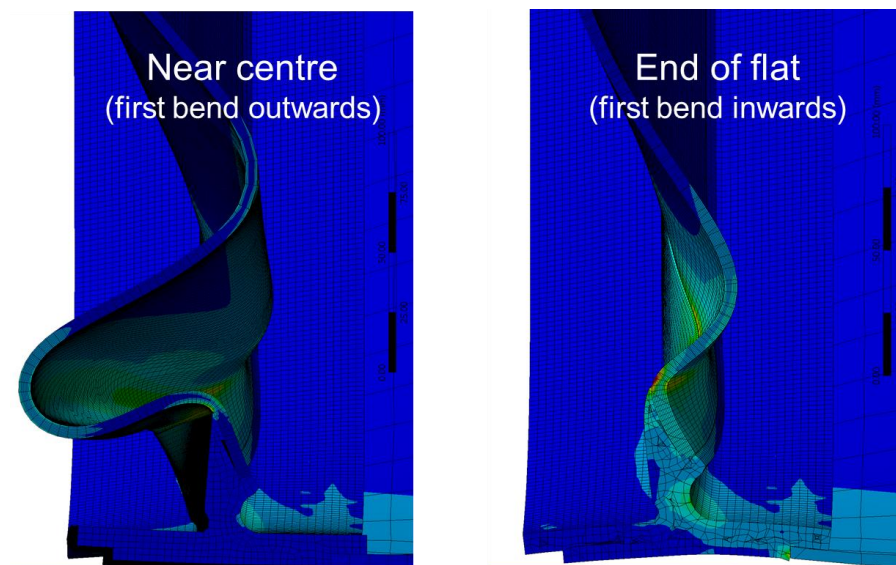


Figure C-12: Comparison of deformation in the dish near the centre of the flat (left) and towards the end of the flat (right) showing different directions for the initial fold – banded type 2 tanker

Since the sample from an actual tanker showed a difference in angle of 4° between extrusion and dish creating a small gap, this was the model that was used for the subsequent model-runs; although only the solid model (for the omega evaluation) contained enough detail to distinguish between the two fits.

The three responses of deflection, plastic work and pressure force obtained from TWI's regression analysis for the banded type 2 subsection are as follows:

Deflection

Figure C-13 shows the response surface for deflection.

¹⁸ A strain condition that is more bi-axial means that the major strain will reach a higher value before the forming limit is reached.

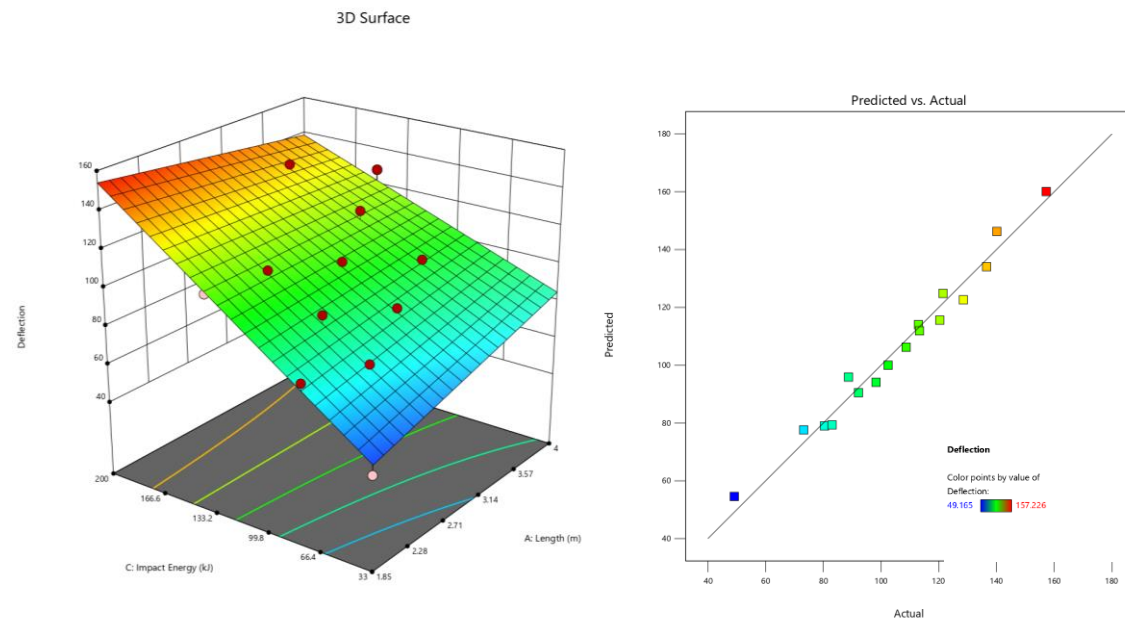


Figure C-13: Response surface for deflection values – banded type 2 tanker

Std. Dev.	-04.58
Mean	106.27
C.V. %	4.31
R ²	0.9770
Adjusted R ²	0.9717
Predicted R ²	0.9397
Adeq Precision	47.4893

- Strong correlation with impact energy.
- Length has less effect.

Plastic Work in Dish

Figure C-14 shows the response surface for plastic work.

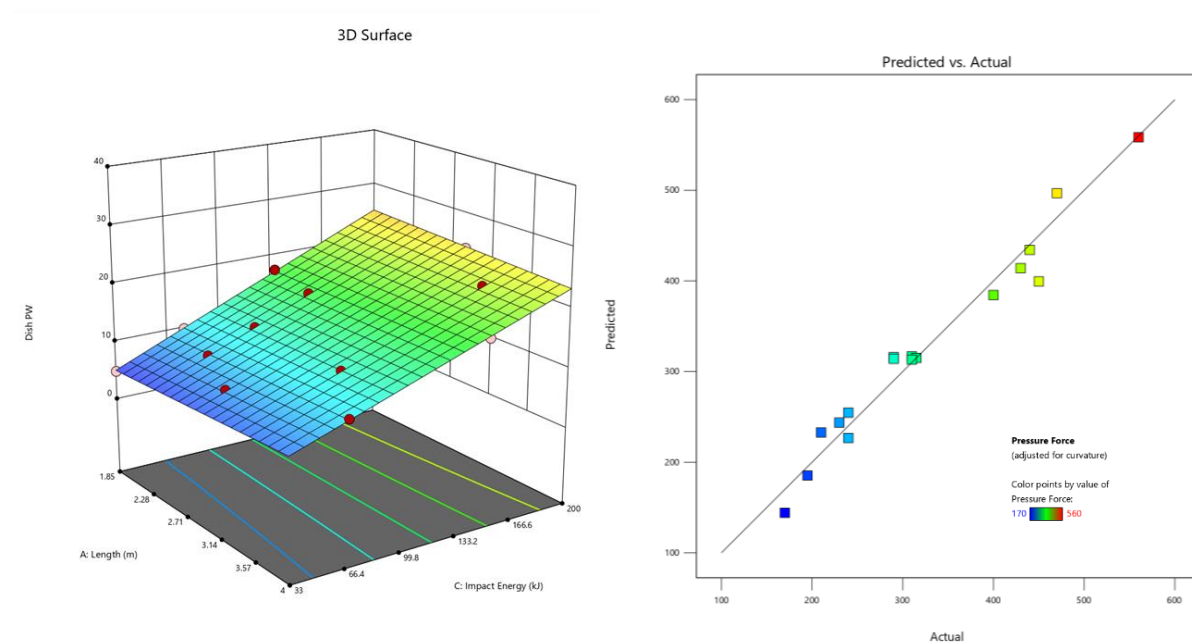


Figure C-14: Response surface for plastic work – banded type 2 tanker

Std. Dev.	22.33
Mean	326.47
C.V. %	6.84
R ²	0.9655
Adjusted R ²	0.9606
Predicted R ²	0.9512
Adeq Precision	44.1730

- Strong correlation with impact energy.
- Little effect from length.

Pressure Force

Figure C-15 shows the response surface for pressure force.

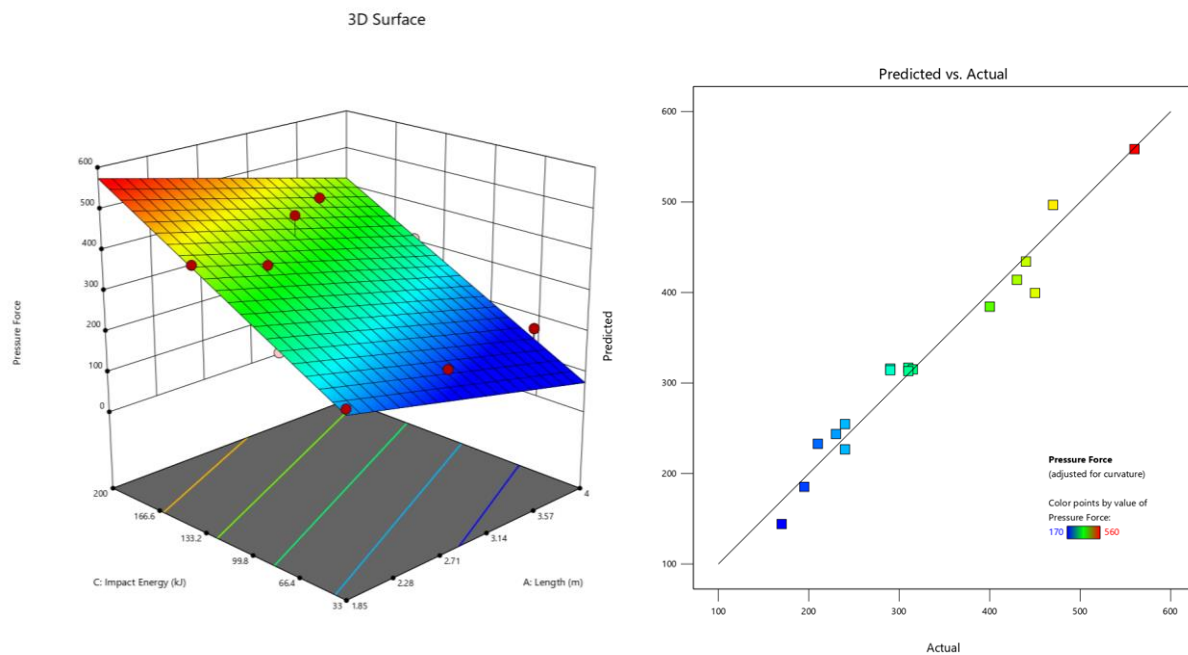


Figure C-15: Response surface for pressure force – banded type 2 tanker

Std. Dev.	22.33
Mean	326.47
C.V. %	6.84
R ²	0.9655
Adjusted R ²	0.9606
Predicted R ²	0.9512
Adeq Precision	44.1730

- Both impact energy and length have an effect.
- Pressure force mainly affected by impact velocity.

C.2.4 *Comparison between banded type 2 and banded type 1 Models*

Figure C16 shows the relationship between strain energy (elastic and plastic strain energy combined) and deflection.

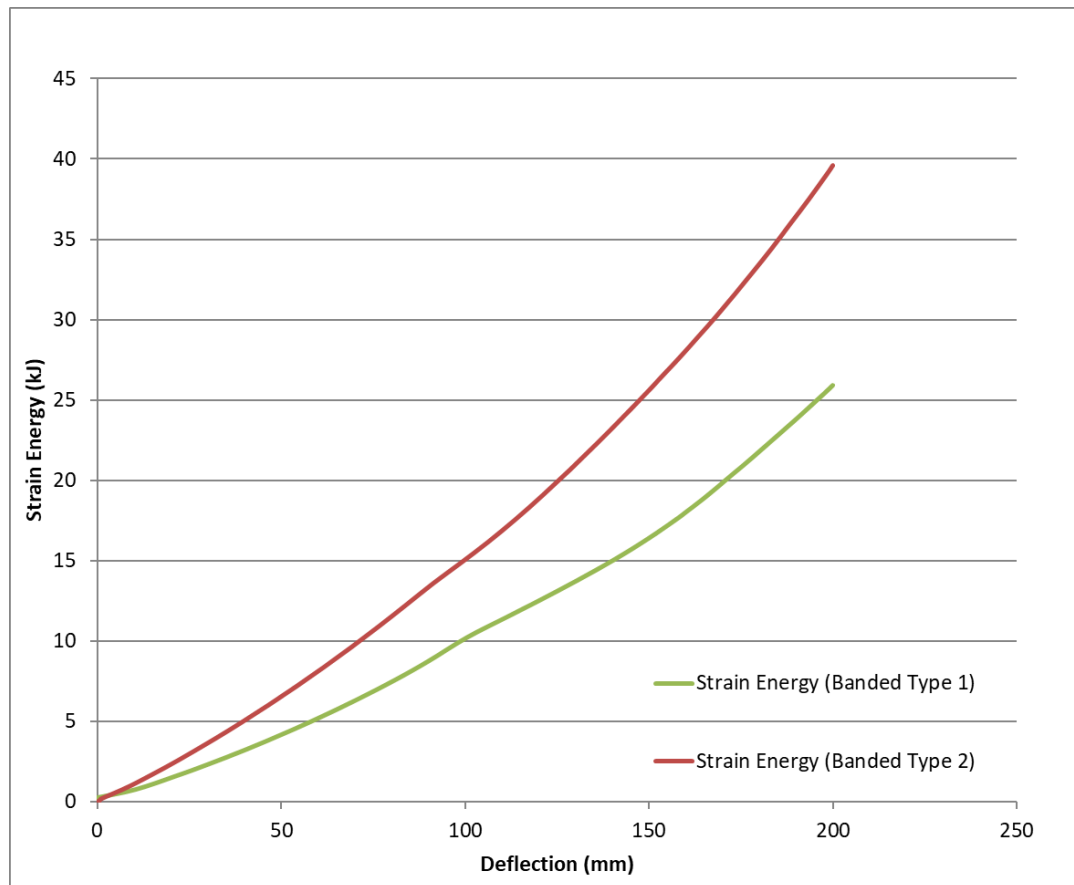


Figure C-16: Comparison of strain energies in the dish for the banded type 2 and banded type 1 extrusion designs

As these curves appear to be increasing as a function of a quadratic equation, and as elastic strain energy is proportional to the square of deflection, this suggests that a significant part of this strain energy is elastic. Therefore, the banded type 2 design is stiffer than the banded type 1 design and to achieve the same deflection as the banded type 1 design, it requires more strain energy to be introduced to the system.

Therefore, more work must be done to achieve an equivalent deflection in the banded type 2 design than for the banded type 1 design.

Deflection

A comparison of the deflection responses for the banded type 1 one and two-compartment models; and the one-compartment banded type 2 models are shown in Figure C-17. These have been plotted against impact energy per partition rather than total impact energy. This is to take account of the differences arising from different numbers of compartments.

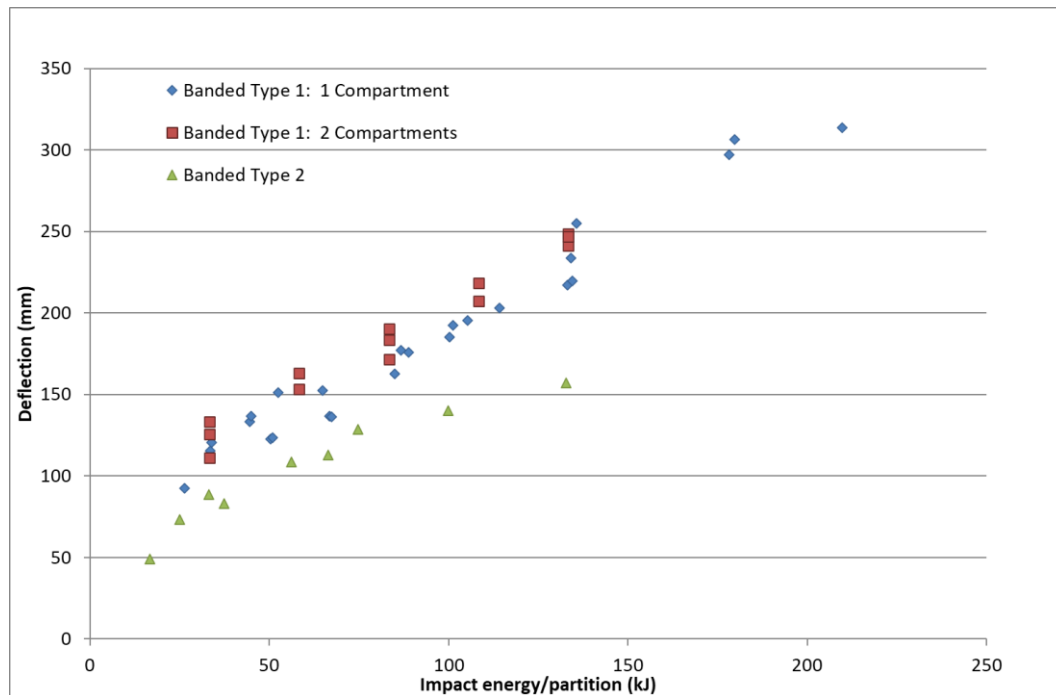


Figure C-17: Maximum deflections of the tanker shell at the partition location for different sets of models, comparing banded type 1 and banded type 2 designs

Results for the two-compartment banded type 1 models are clustered at specific energies, as impact energy was a test parameter as explained earlier. For the other two sets of models (categoric factors), length and impact velocity were test parameters, giving a wider range of energy values. The results clearly show that the banded type 2 design results in lower deflections for a given impact energy per partition compared to the banded type 1 design.

Plastic work

The amount of plastic work in the partition dishes is shown in Figure C18. As expected, the banded type 2 design has lower levels of plastic work for a given impact energy per partition compared to the banded type 1 design.

This shows that less of the impact energy is absorbed by plastic work for the banded type 2 design. The impact energy may be converted into more elastic strain energy; or there may be more energy transferred to the liquid due to the higher decelerations due to the banded type 2 being stiffer.

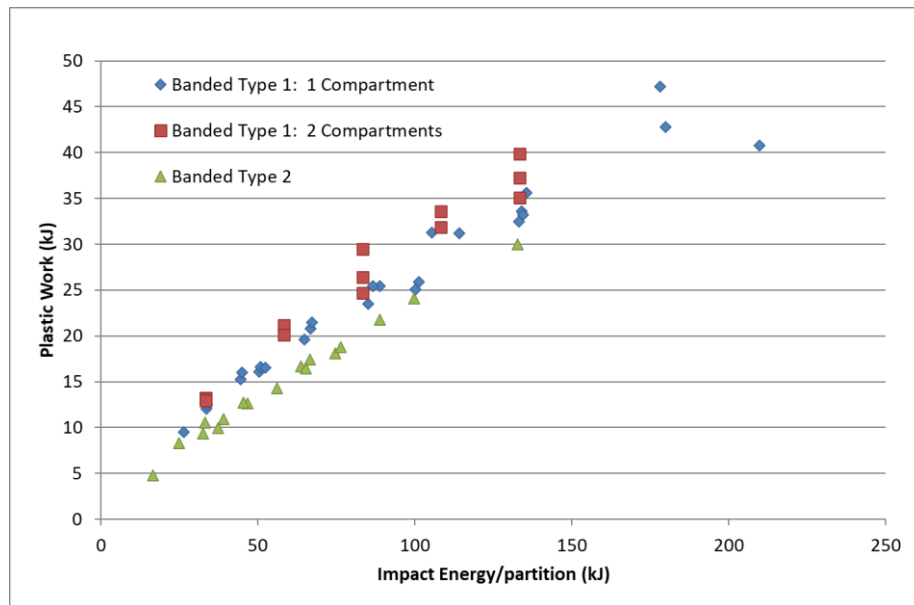


Figure C-18: Plastic work in the partition dishes for different sets of models, banded type 1 and banded type 2 designs

Pressure force

The pressure force is plotted against static pressure multiplied by velocity as this gives the best fit for plotting relationships. The static pressure is the product of the liquid density, liquid depth, and acceleration due to gravity. The actual pressure is likely to be a function of the static pressure and the rate of deceleration on impact, which is likely to be related to the impact velocity and the stiffness of the tanker.

As the banded type 2 design is stiffer, and therefore deceleration at any given impact velocity is likely to be higher, the pressure forces are higher for the banded type 2 design, as can be seen in Figure C-19.

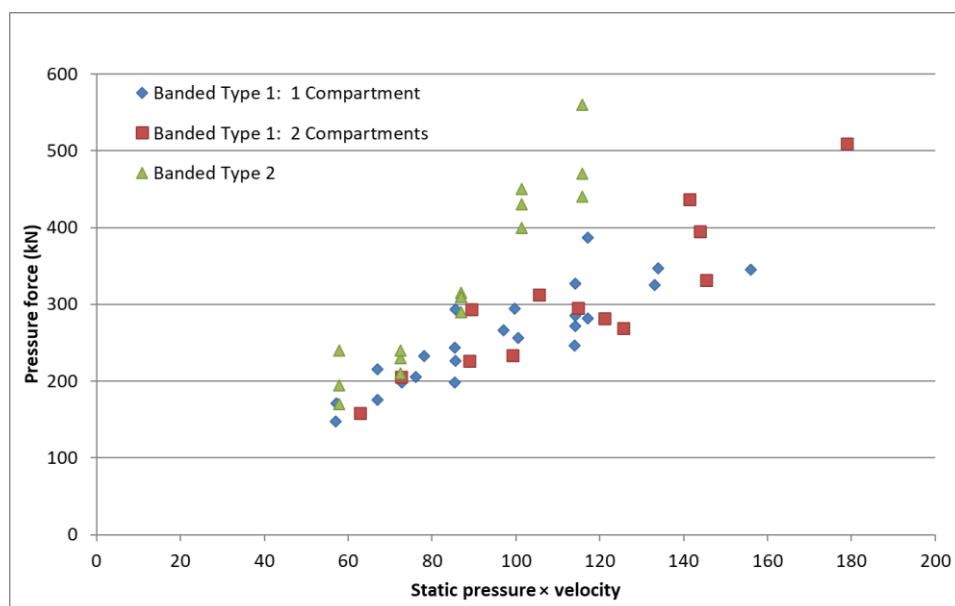


Figure C-19: Pressure forces acting on the partition dishes vs static pressure \times impact velocity for different sets of models, banded type 1 and banded type 2 designs

Omega (deflection based)

The omega values are shown in Figure C20, plotted against deflection.

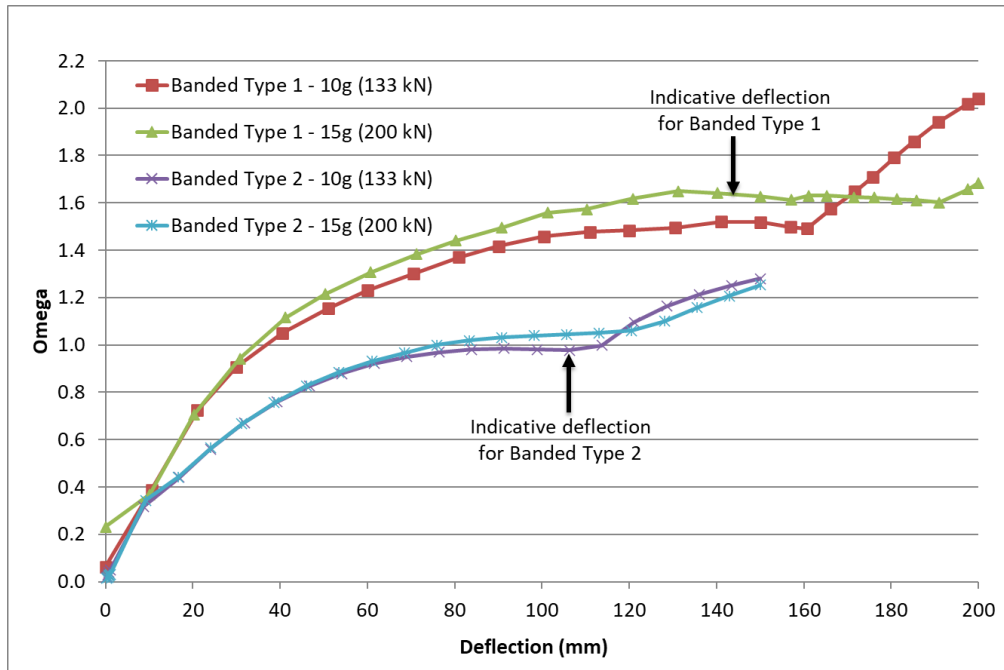


Figure C-20 Omega values vs deflection, banded type 1 and banded type 2 designs

The coupling pressure forces selected (133 kN and 200 kN) are slightly above and below the coupling force that was obtained from the banded type 1 benchmark topple model (156 kN). The acceleration values (10g and 15g) were the input values in the solid model of the banded type 2 subsection that gave these coupling pressure force.

The omega values for the banded type 2 design are clearly lower than the banded type 1 design. As mentioned earlier, an omega value of 1 indicates the forming limit has been reached, although the forming limit curve used is conservative as previously mentioned.

For both banded type 1 and banded type 2 designs, omega values plateau before starting to rise again, which corresponds to the change in location of the maximum omega value from the inside surface to the outside surface of the dish as previously explained. However, for both designs, this point occurs at higher levels of deflection than would be likely to occur under the impact conditions representing a topple.

These results also show that changes in the pressure force between 133 kN and 200 kN have little effect on the omega values.

As the deflections occurring for a given impact energy vary between the banded type 1 and banded type 2 designs, the omega values have been replotted against impact energy per partition in Figure C21. The conversion between deflection and impact energy has been done using a power law best fit through points on the deflection/impact energy plot for the banded type 1 and banded type 2 designs (i.e. from the data shown in Figure C-17).

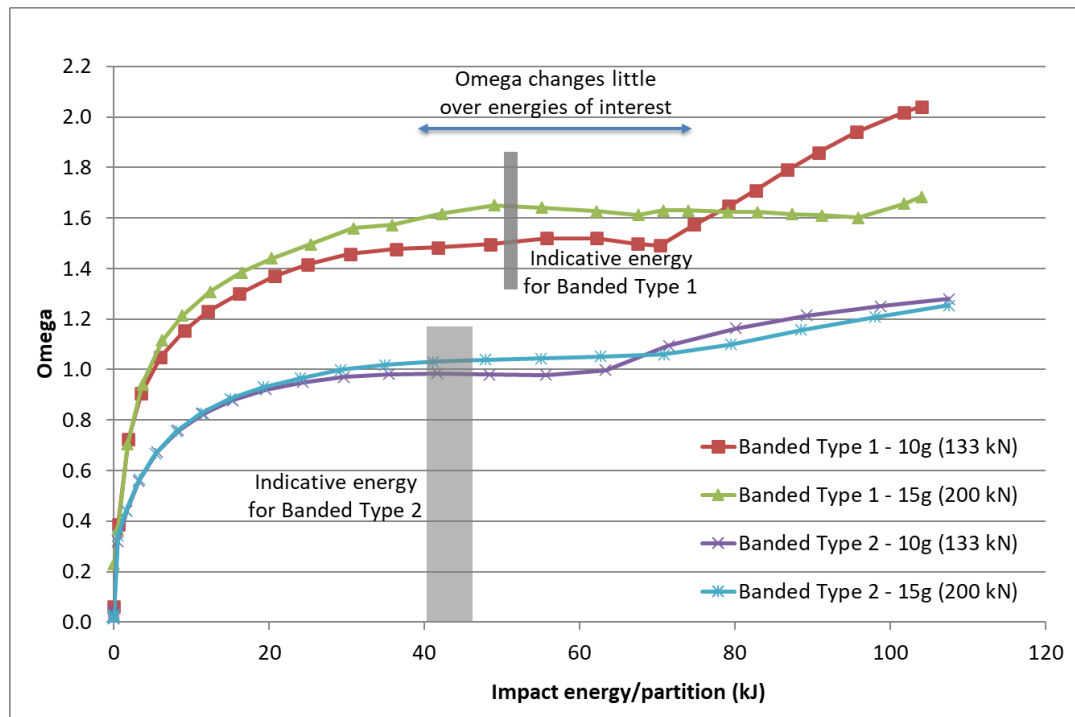


Figure C-21: Omega values vs impact energy per partition for banded type 1 and banded type 2 designs

C.2.5 Topple test comparisons

Although there is currently no benchmark model for the banded type 2 subsection, the topple test carried out on the banded type 2 tanker in the previous project can be used for comparison. The width of the tankers after the rollover impacts are shown in the laser scan results in Figure C-22.

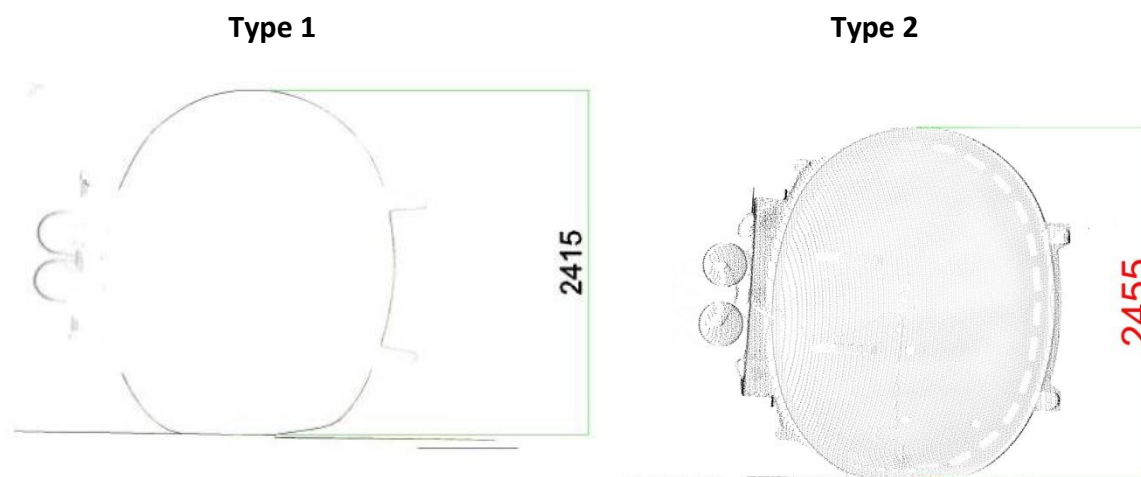


Figure C-22: Comparison of laser scans of the banded type 1 (left) and banded type 2 (right) tankers after topple test

In the previous work on topple tests, two banded type 1 tankers (“A” and “B”) and one banded type 2 tanker were toppled.

The rear width of the tankers before and after the impact are as follows:

- Banded type 1 A: 2530 mm – 2430 mm = **100 mm**
- Banded type 1 B: 2522 mm – 2415 mm = **107 mm**
- Banded type 2: 2545 mm – 2455 mm = **90 mm**

The deformation is the difference between the two values and is shown **in bold**.

These results show that the banded type 2 tanker deformed less than the banded type 1 tankers ($\approx 15\%$ lower). Other observations of the damage from the topple tests were that the width of the flat, impacted section was approximately 20% lower on the banded type 2 tanker than the banded type 1 tankers. Also, the peak pressures that were measured were higher in the banded type 2.

However, the drop test subsection models suggested a larger difference between banded type 1 and banded type 2 designs (approximately 30% less deflection) than the above results. Possible reasons for this difference are as follows:

- Uncertainty about banded type 2 energy
- Drop vs topple
- Final deformation vs peak deformation
- Effect of single compartment vs complete tanker

However, the comparisons between the models for the banded type 2 and banded type 1 designs did show that omega values are lower for the same deflection for the banded type 2 design. Also, the design of the extrusion is stiffer in the banded type 2 design, which means there is less deflection on impact for a given impact energy. The results agreed well with previous TWI results and HSE topple (basic comparison).

Key Points

The key points from the previous sections on steps 1 to 3 results are as follows:

- Good correlations were achieved for the banded type 1 (single and double compartments) and the banded type 2 design.
- For both designs, it was found that deflections and plastic work depended mainly on impact energy. Varying the length of the subsection had less effect.
- The pressure forces depended mainly on impact velocity.
- For an equivalent impact energy, the banded type 2 design had less deflection and higher pressure forces than the banded type 1 design.
- The omega value was lower for the banded type 2 subsection for an equivalent deflection.

C.2.6 *Stuffed tanker design, and further work on number of compartments*

Step 4 included modelling work on a single-compartment stuffed subsection design, and further comparisons between the banded type 2 and banded type 1 designs with a three-compartment banded type 1 model.

C.2.6.1 Three-compartment banded type 1 model and 8 and 10-banded banded type 1 tankers

Further banded type 1 models using three compartments were developed and compared with the 8 and 10-banded banded type 1 tankers. The list of model-runs for the three-compartment banded type 1 subsection are shown in Table C5.

Table C-5: Number of model-runs carried out for the Step 4 three compartment subsection work

RUN	Length of Compartments (m)	Fluid Density (kg/litre)	Fill Proportion	Impact Velocity (m/s)	Impact energy (kJ)
1	1.85	1.00	0.83	4.0	162
2	1.85	1.00	0.83	4.5	205
3	2.93	1.00	0.83	4.0	251
4	2.93	1.00	0.83	3.0	141
5	1.85	1.00	0.83	1.0	10
6	1.85	1.00	0.83	2.0	41

C.2.6.2 Banded type 1 – Number of compartments

The results for deflection as a function of impact energy are shown in Figure C-23.

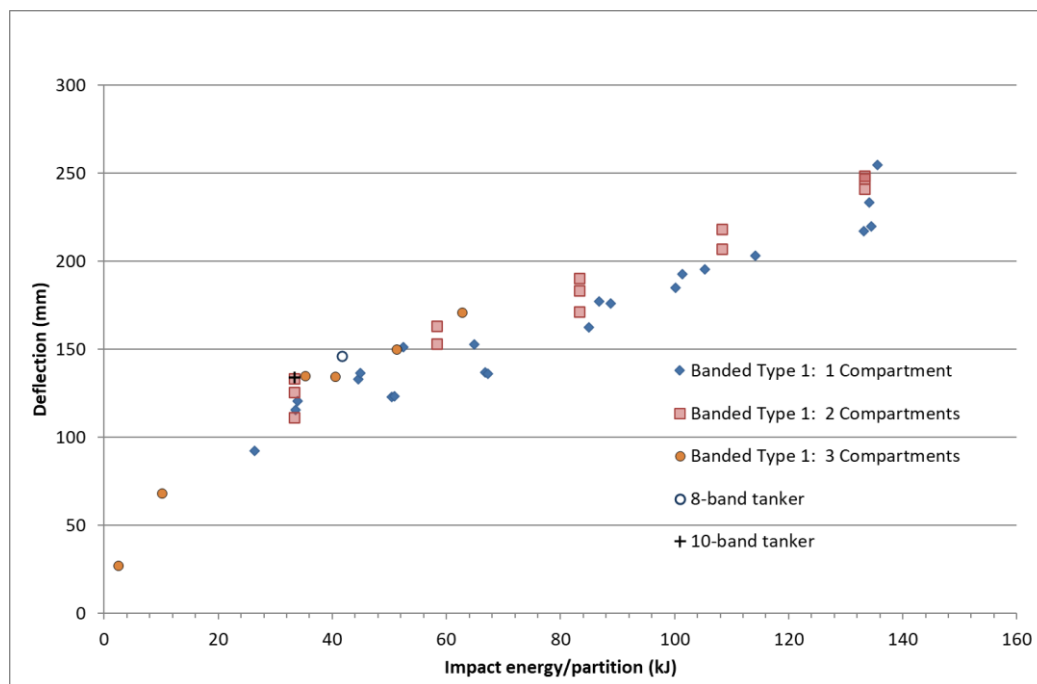


Figure C-23: The effect of number of compartments (and therefore partitions) on deflection for banded type 1 tanker designs

Figure C-23 shows that, when the data from all models is considered together, the results follow a clear trend. The 3-compartment model results fit well with the 1- and 2-compartment models, and the difference between 8- and 10-banded type 1 tankers show a similar trend accounting for the fact that the 10-banded tanker has a lower impact energy/partition.

C.2.6.3 *Banded type 1 – comparison to banded type 2*

Figure C24 shows a comparison of the maximum deflection as a function of the energy per partition from the banded type 2 single compartment model, and the banded type 1 results for the one, two and three compartment models.

The combined banded type 1 results for all numbers of compartments including complete tankers show good correlation ($R^2 = 0.94$), although the banded type 1 results show lower R^2 value compared to banded type 2 as there was a greater variability in what was modelled (number of compartments, lengths, fill levels etc.) so some 'scatter' is to be expected. However, there is a clear difference between tanker designs as mentioned in the previous section. That said, it is noted that the ability to achieve an R^2 score greater than 0.9 when considering only a single variable (impact energy) highlights the importance of impact energy as the key factor.

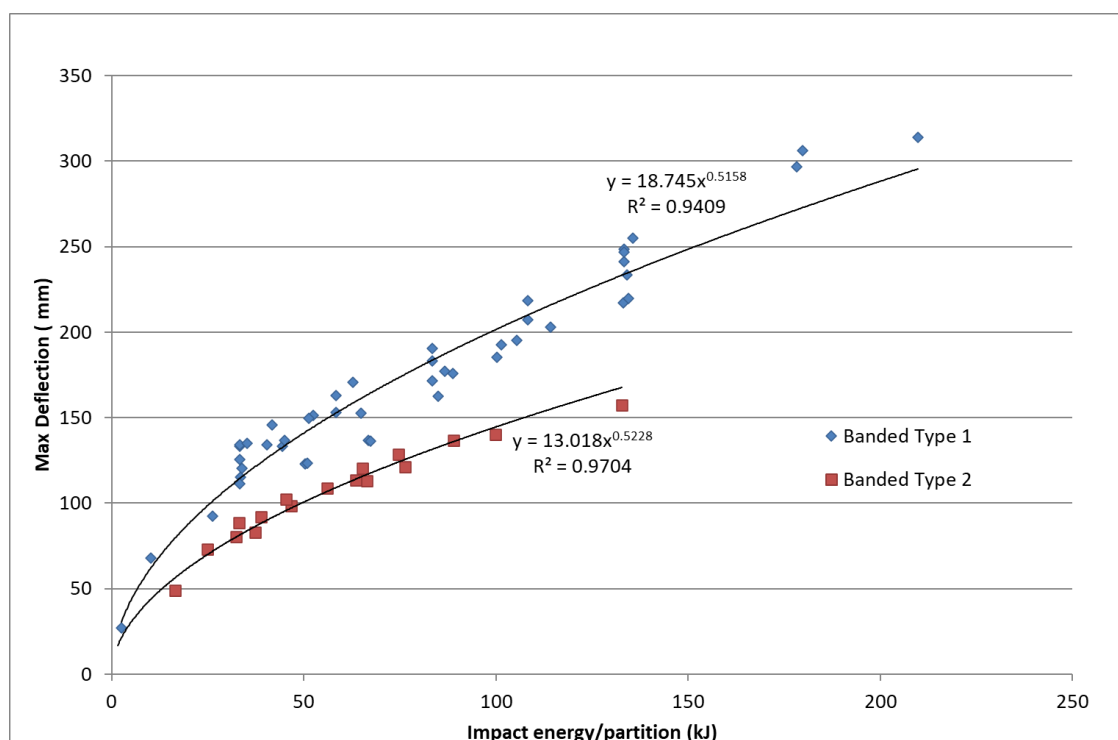


Figure C-24: Comparison of deflections between banded type 2 design and banded type 1 design (banded type 1 data including all numbers of compartments modelled)

C.2.6.4 *Plastic work in each partition of complete tanker – 8 and 10 banded banded type 1 tankers*

Table C-6 lists the plastic work absorbed by each partition in 8- and 10-banded banded type 1 tankers dropped with an impact velocity of 4.5 m/s. Note that these energies represent

the plastic work done in each of the partition dishes **only**. It does not include energy absorbed in the extrusion or the shell.

The initial kinetic energy of the complete tanker was approximately 330 kJ in each case. However, Table C-6 shows that the energy absorbed by individual bands was higher in the 8-banded tanker due to higher levels of deflection.

The plastic work at Band A is lower than the other bands, and there is an increase in plastic work per band towards the rear. However, the last three dishes- have similar energy values to each other in both tankers. These differences could be due to geometry as the front compartment has a lower cross-sectional area as it is above the tractor unit. Therefore, the energy per unit length will be lower in this region, which will influence the amount of energy absorbed. Also, the geometry of the band and dish are different at the front of the tanker – this also may be affecting the results. The lower deformation of band A would suggest that the impact energy was also lower in this region. As the previous work showed failures in the front-end bulkhead in two of the topple test tankers, lower plastic work in the dish may not equate to a lower risk of failure.

Table C-6 Plastic work absorbed by each band in complete tanker drop models¹⁹

Location Band	10 Band (kJ)	8 Band (kJ)
A (Front)	7.7	8.5
B	10.1	11.0
C	12.2	14.0
C/D	9.0	
D	12.6	14.5
E	13.6	15.9
E/F	13.4	
F	14.4	16.6
G	15.1	16.8
H (Rear)	14.7	16.0
Total	122.6	113.3

By comparing the total energy absorbed by the partitions (bottom of Table C-6) with the initial kinetic energy (330 kJ), then about 2/3 of the energy is unaccounted for. The unaccounted energy will be absorbed, stored, and dissipated in the following ways:

¹⁹ For information, in the topple benchmark model the deflections were as follows: Band A – 85 mm; Band D – 105 mm; Band H – 140 mm.

- deformations of other components such as the shell;
- friction;
- elastic deformation
- energy transferred to the water (which is likely to be the larger proportion).

C.2.6.5 *Stuffed Design*

The stuffed tanker design is quite different to the banded type 1 and banded type 2 designs. Banded tankers use extruded bands and the shell (which is in sections) and partitions are welded to these extrusions. External supports are also connected to the bands: these supports connect the tanker to the trailer chassis. Stuffed tanker designs have a continuous shell, and partitions are placed within the shell (stuffed) and then welded in position. External supports are welded to saddles on the shell.

Figure C-25 shows the different designs for the two banded tankers, and a stuffed design.



Figure C-25: Differences in the joint designs for banded type 1 and banded type 2 banded designs, and a stuffed design

Figures C-26 and C-27 show manufacturer's drawing for a stuffed design.

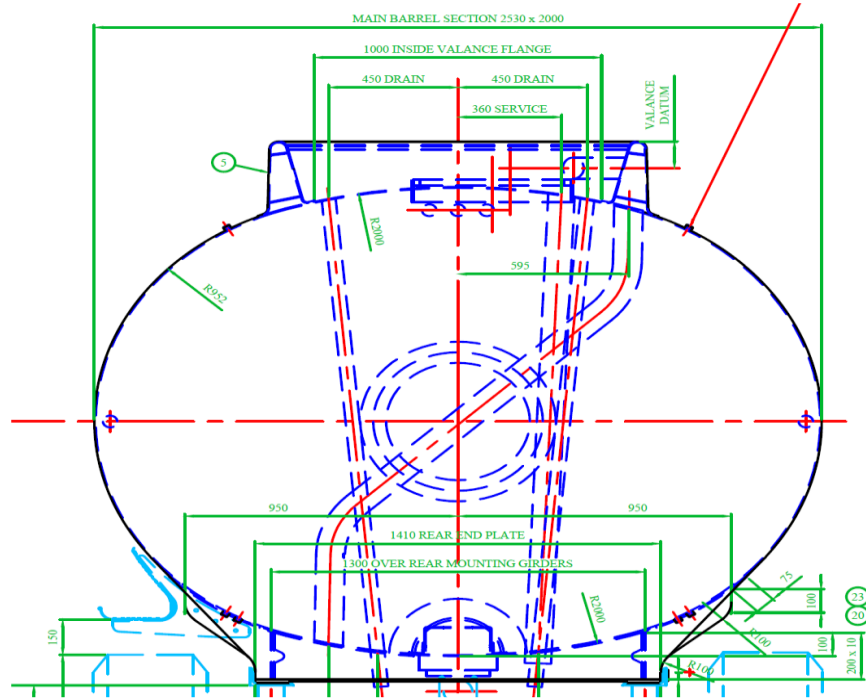


Figure C-26 Manufacturer's design drawing for profile of stuffed tanker

Manufacturer

BS EN 13094

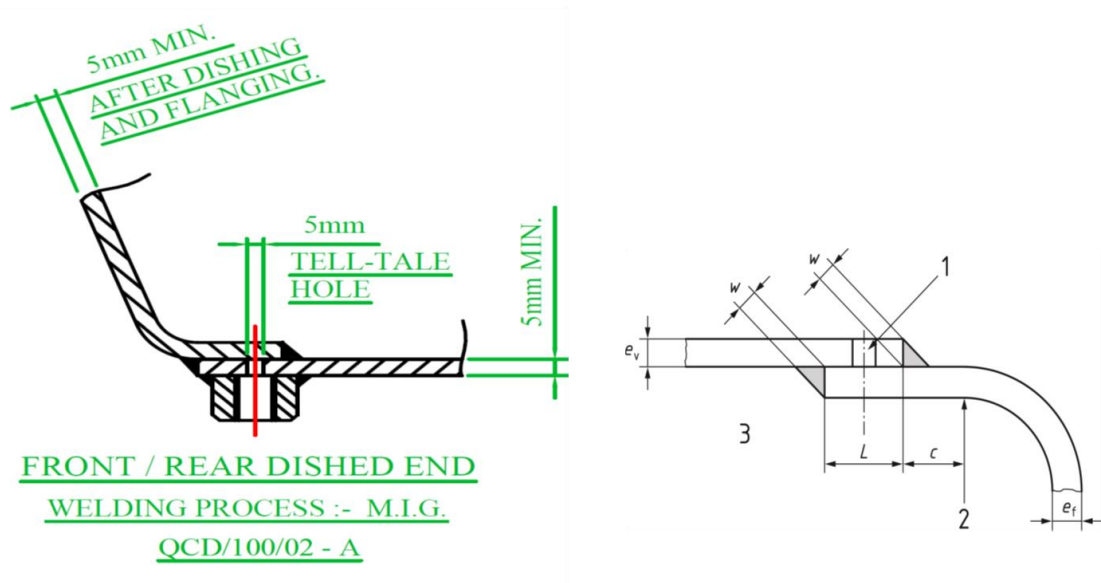


Figure C-27 Joint detail for stuffed design from manufacturer (left) and generic stuffed design from BS EN 13094 (right)

Figure C-28 shows the stuffed design as modelled.

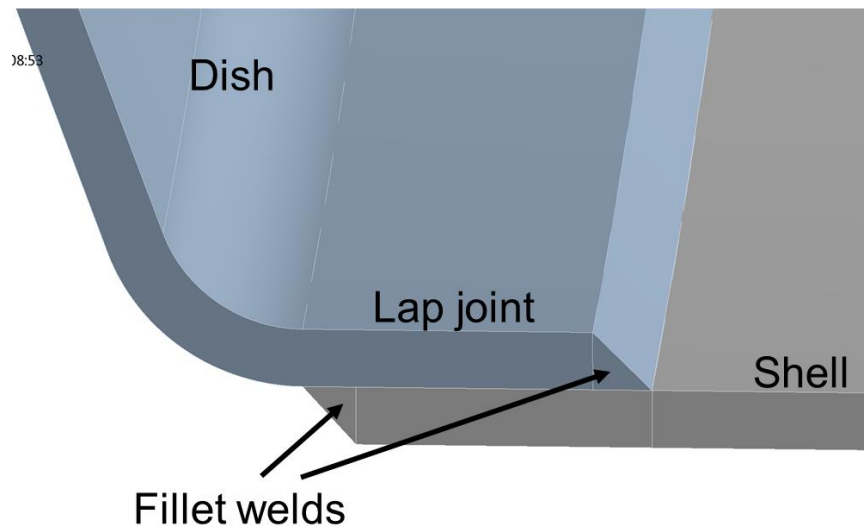


Figure C-28: Stuffed design – Joint detail as modelled

The list of model runs is given in Table C-7.

Table C-7: Number of model-runs carried out for the Step 4 stuffed tanker work

RUN	Length of Compartment (m)	Fluid Density (kg/litre)	Fill Proportion	Impact Velocity (m/s)	Impact energy (kJ)
1	1.770	1.00	0.83	2.000	14.6
2	1.770	1.00	0.83	6.000	58.5
3	1.770	1.00	0.83	4.000	131.6
4	2.053	1.00	0.83	6.000	149.4
5	2.244	1.00	0.83	4.852	105.4
6	2.244	1.00	0.83	2.852	36.4
7	2.719	1.00	0.83	3.703	72.7
8	2.885	1.00	0.83	2.000	22.4
9	3.027	1.00	0.83	4.258	105.8
10	3.359	1.00	0.83	2.852	52.2
11	4.000	1.00	0.83	2.000	30.0
12	4.000	1.00	0.83	2.515	47.5

Figure C-29 shows the deflection as a function of impact energy (converted to energy per partition) for the single-compartment stuffed design.

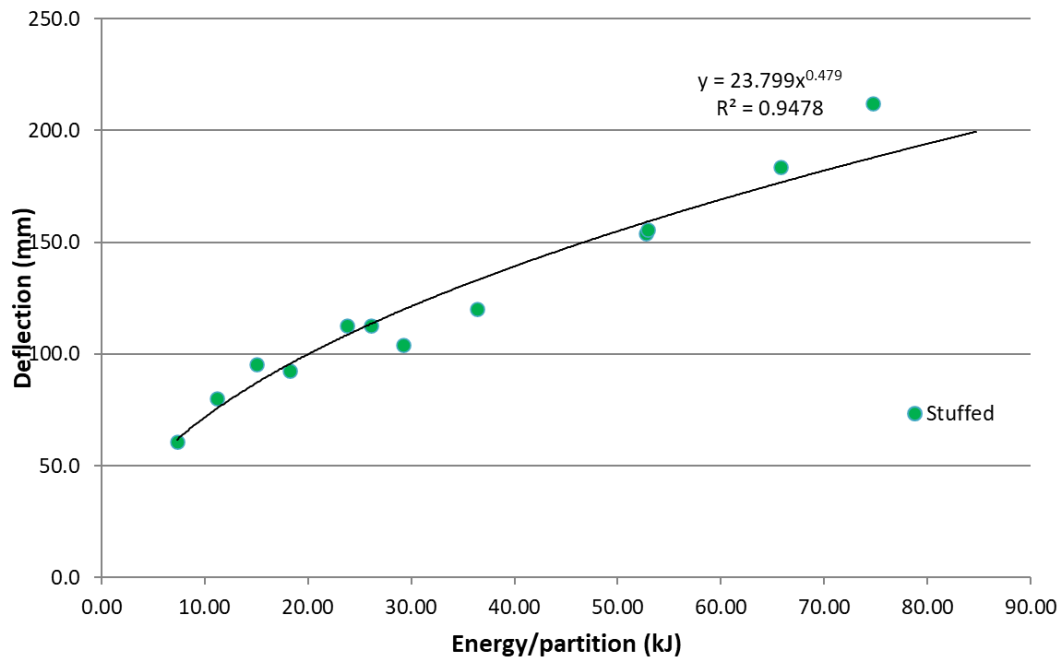


Figure C-29: Deflection vs impact energy per partition for the stuffed design with a power law best fit line

Figure C-30 compares the deflections for the stuffed subsection with the previous results for the banded subsections.

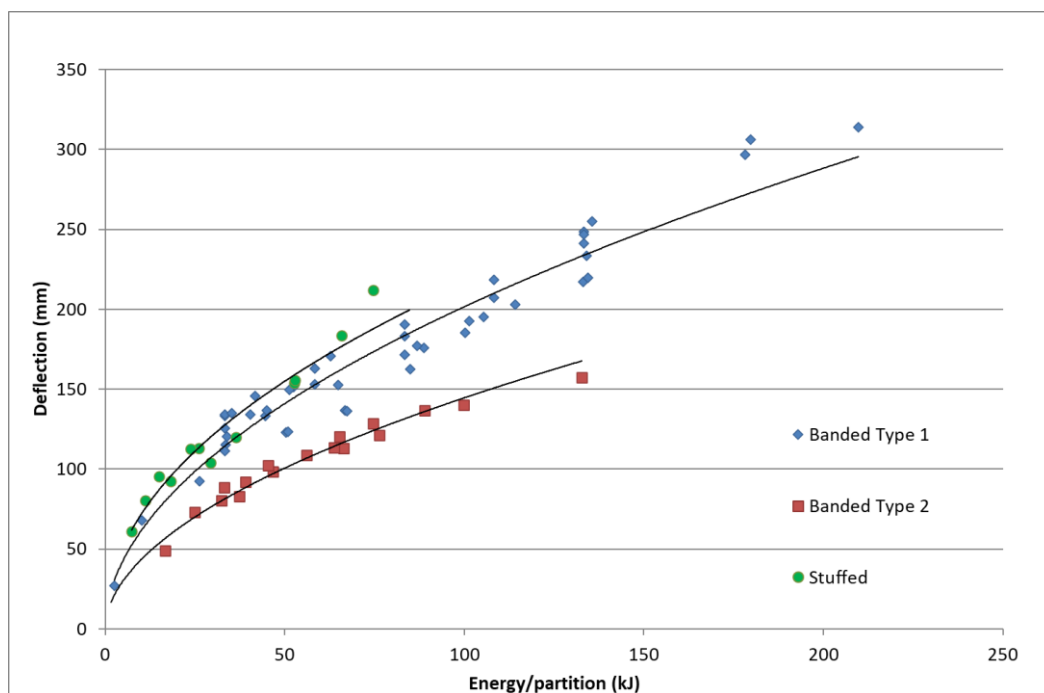


Figure C-30: Comparison of deflection results for banded type 1, banded type 2 and stuffed designs

The results show that the deflections for the stuffed subsection are slightly higher than the two 'banded' subsections. This is thought to be due to the joint lifting off the impact

surface, which occurred to a much greater extent in the stuffed subsections than it did for the banded subsections. Figure C-31 shows this effect in the model.

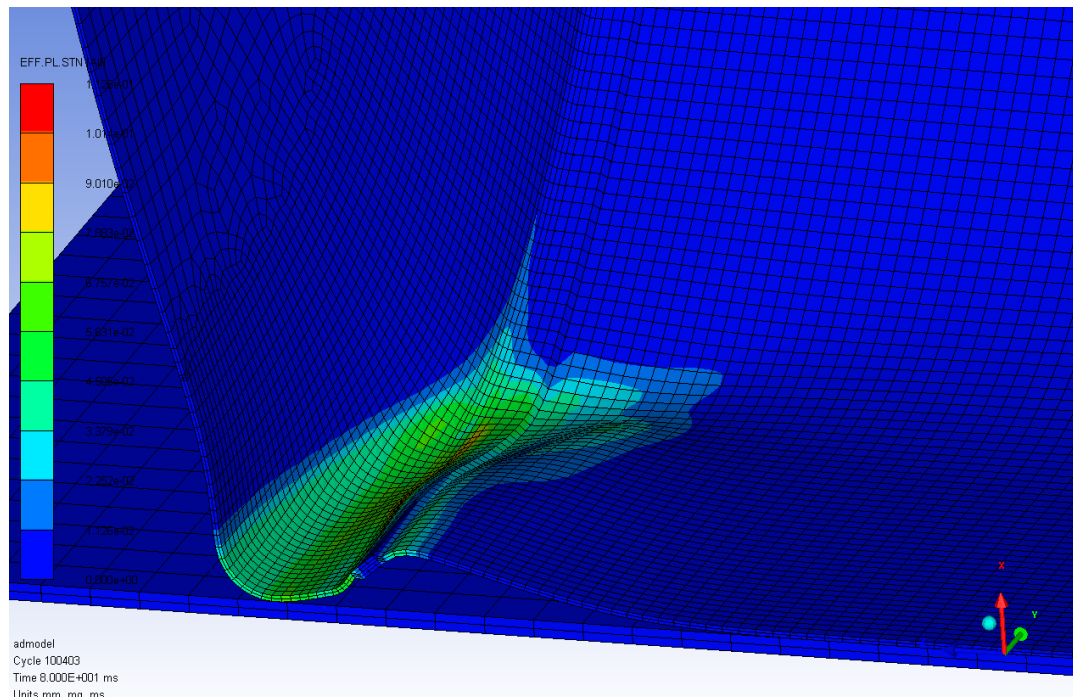


Figure C-31: Stuffed design – Joint lifting off the surface during impact.

In the previous topple test project, the *proof of concept* test was carried out on a stuffed tanker. Photographs from the test were reviewed to see if this effect was present. Figure C-32 shows evidence that the shell at the joint did bend into the tanker during the test.



Figure C-32: The proof of concept test tanker from the original work after topple test showing joint bending into the tanker, as seen in the stuffed model

Figure C-33 shows values of the omega forming limit for the three tanker designs plotted against deflection from the solid models.

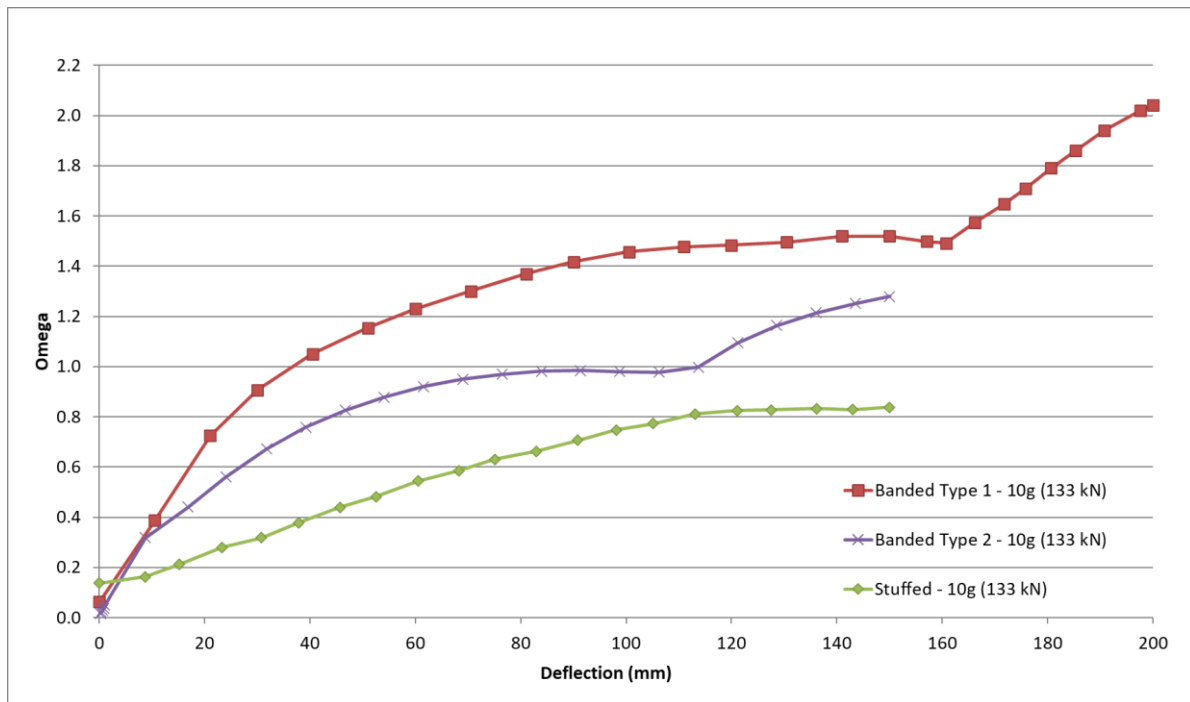


Figure C-33 Omega values compared between banded type 1, banded type 2 and stuffed designs.

Figure C-34 shows the same omega values plotted against impact energy per partition.

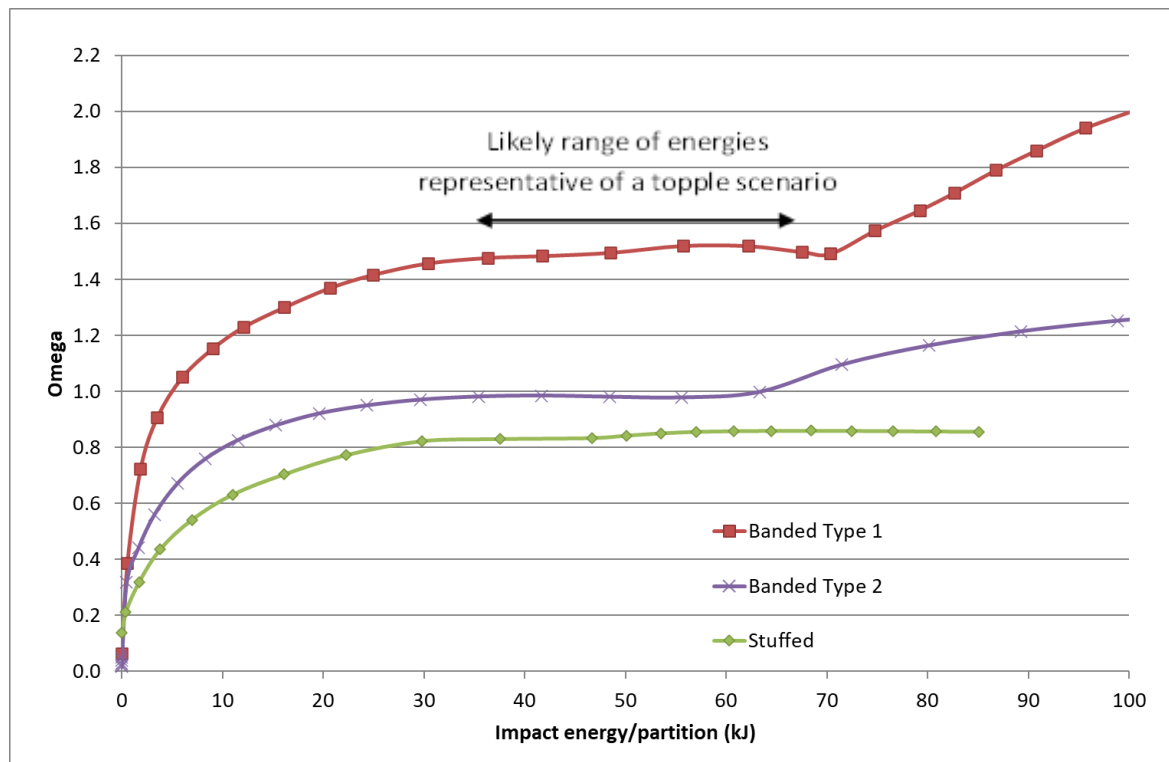


Figure C-34: Omega values vs energy/partition for banded type 1, banded type 2 and stuffed designs

C.3 Conclusions

This section considers the key conclusions from the modelling work described in this appendix. The work has considered one, two and three-compartment subsections with the banded type 1 banded design; a one-compartment subsection with the banded type 2 banded design; and a one-compartment subsection with the stuffed design.

The results in this work have shown that a banded type 1 subsection model can be specified and modelled in a drop test to give a reasonably equivalent level of damage to the 8-banded banded type 1 benchmark tanker (petrol topple) from the previous work. This required that response parameters, in particular the impact energy absorbed by a partition (related to its plastic work and deflection) and local strain conditions (omega value) were known.

Therefore, although there are uncertainties that need to be considered, there is no conclusive evidence at this stage that a subsection test would not be a suitable test method.

Important parameters

Step 1, modelling single compartments of banded type 1 tankers, investigated the effect of four *numerical factors*: fill level; liquid density; impact velocity; and compartment length, on the structural response. It was found that the most important factor was a combination of numeric factors: the impact energy. For example, it made little difference to the structural response (e.g. deflection) if the energy was obtained by a low density liquid moving at higher velocity; or a higher density liquid moving more slowly. The length of the compartment was found to have a small effect on the response.

During step 1, it was found that the fluid structure interaction (FSI) models using shell elements could not accurately obtain the local strains necessary to calculate omega values. Omega is a factor used to describe the strain condition in the material in relation to the forming limit – a value of 1 indicates that the strains are on the forming limit (i.e. it's about to break). Therefore, to resolve this issue, the omega values were calculated with a separate static model using solid elements. However, as this model did not include the interaction with the liquid, the pressures were calculated from the coupling forces from the FSI models. This does illustrate that construction of FE models capable of predicting whether or not failure is likely to occur in a topple or drop type event is quite a difficult and involved process which requires a high level of expertise.

Number of compartments

Once models were created with different numbers of compartments, it was clear that the important test parameter was the impact energy per partition, rather than just the impact energy. Therefore, adding a partition to a single compartment would require an increase in impact energy of approximately 50% (moving from two partitions to three) to achieve the same deflection. The subsection models for the banded type 1 tankers showed that the impact responses followed the same trends for one, two and three compartment subsection models, and also for the complete banded type 1 tanker models with eight and ten bands.

Different tanker designs

Three different tanker designs were modelled: banded type 1 banded design; banded type 2 banded design; and a stuffed design. The main difference between the two 'banded' designs is that the banded type 1 design has the dish welded directly to the extrusion band. In the

banded type 2 design, there is an upstand on the extrusion band on which the dish is attached.

The stuffed design has a continuous shell with the partitions welded into the tube using a lap joint.

The main conclusions from the different design studies are as follows:

- There is good correlation between deflection and impact energy per partition for each of the designs.
- The response of the stuffed joint design followed a similar trend to the banded designs. All three designs showed a power-law relationship²⁰ between the deflection and impact energy per partition.
- The deflection of the banded type 2 design was lower than for the banded type 1 design for a given impact energy per partition, whilst the deflection of the stuffed design was slightly higher than the banded type 1 design.
- For the banded designs (banded type 1 and banded type 2), Omega values were fairly constant within the range of 30 kJ to 70 kJ (per partition), which is the range of interest as this is comparable to the energy absorbed per partition in a topple test. However, there is a clear difference between the designs, with the banded type 1 design having a much higher value of Omega, indicating likely failure. A lower omega value indicates a lower risk of failure.
- The results suggest that, at least for these banded designs, a subsection drop test could differentiate between 'good' and 'bad' joint designs across a wide range of impact energies based on a comparison of the predicted omega values for each joint design.
- Omega values were lowest for the stuffed design and showed little change in value at impact energies above 30 kJ/partition (like the banded tankers).
- Results agreed with previous work from TWI (confirming lower omega values for the stuffed design).

For the stuffed design, results agreed with the '*proof-of-concept*' test from the topple test project (i.e. no failure of the rear end despite higher loading; although the '*proof of concept*' stuffed tanker did fail in the front bulkhead, and also the internal partitions failed). Also, the modelled response in this work showed that the stuffed joint lifted off the impact surface. This effect was also observed in the '*proof of concept*' test on the stuffed tanker from the previous project. However, note that the impact energy in the *proof-of-concept* was higher than would be experienced in the real-world because the tanker was filled with an equivalent volume of water which is more dense and thus heavier than a real-world fuel load.

²⁰ In a 'power-law' relationship, one variable changes as a power of the other variable (e.g. by considering a square in terms of the length of one side, if the length is doubled, the area is increased by a factor of four).

Appendix D Review of potential abrasion test methods

D.1 The helmet abrasion resistance test (ref UN, 2021)

Two types of test were considered from this regulation: the helmet chinstrap test; and the helmet test (which contained three methods).

D.1.1 Chinstrap test

This test is in clause 7.11 and Annex 8 Figure 5 of (UN, 2021). Clause 7.11 is explained below.

Brief Summary

The chin strap is moved back and forward under tension through the rigid part of the retention system (a 'slider' fitting in most cases) for a fixed number of cycles to generate abrasion, and then it is tested in a tensile testing machine. Providing it achieves a minimum breaking strength, it passes the test. The purpose of the test appears to be more to do with ensuring that the strap is suitably resilient to last for a minimum period of operations (tightened and untightened) under normal use without losing too much strength. It is a lifecycle test rather than an accident test.

Clause 7.11 is shown in full below:

7.11 Test for resistance to abrasion of the chin strap (see Annex 8, Figure 5)

The test shall be performed on every device in which the strap slides through a rigid part of the retention system, with the following exceptions:

(a) Where the micro-slip test, paragraph 7.10., shows that the strap slips less than half the prescribed value; or,

(b) Where the composition of the material used, or the information already available, renders the test superfluous in the judgement of the technical service.

7.11.1. The test rig is similar to that described in paragraph 7.10.1. except that the amplitude of motion is 100 ± 10 mm and the strap passes over a representative surface of the associated adjuster or other strap fitting through an appropriate angle.

7.11.2. Select an arrangement of the apparatus appropriate for the particular design of both the strap and the fitting likely to cause abrasion. Grip one end of the strap in the oscillating clamp, arrange the strap to be threaded through the fitting as designed and hang a weight on the end to tension the strap with a force of 20 ± 1 N. Mount or otherwise steady the fitting in such a position that movement of the oscillating clamp slides the strap through the fitting, in a manner simulating slippage of the fitting on the strap when the helmet is on the head.

7.11.3. Oscillate the clamp for a total of 5,000 cycles at a frequency between 0.5 and 2 Hz.

7.11.4. Mount the abraded strap in a tensile testing machine using clamps which avoid local breakage of the strap, and so that there is a length of 150 ± 15 mm of strap, including the abraded portion, between the clamps. Operate the machine to stretch the strap at a speed of 100 ± 20 mm per minute.

7.11.5. The strap shall withstand a tension of 3 kN without breaking.

There are two methods of setting up the test which are shown in Figure 5 in Annex 8 of Regulation UN Regulation No. 22 (see Figure D-1 below):

Figure 5
Apparatus for testing abrasion of the chin strap

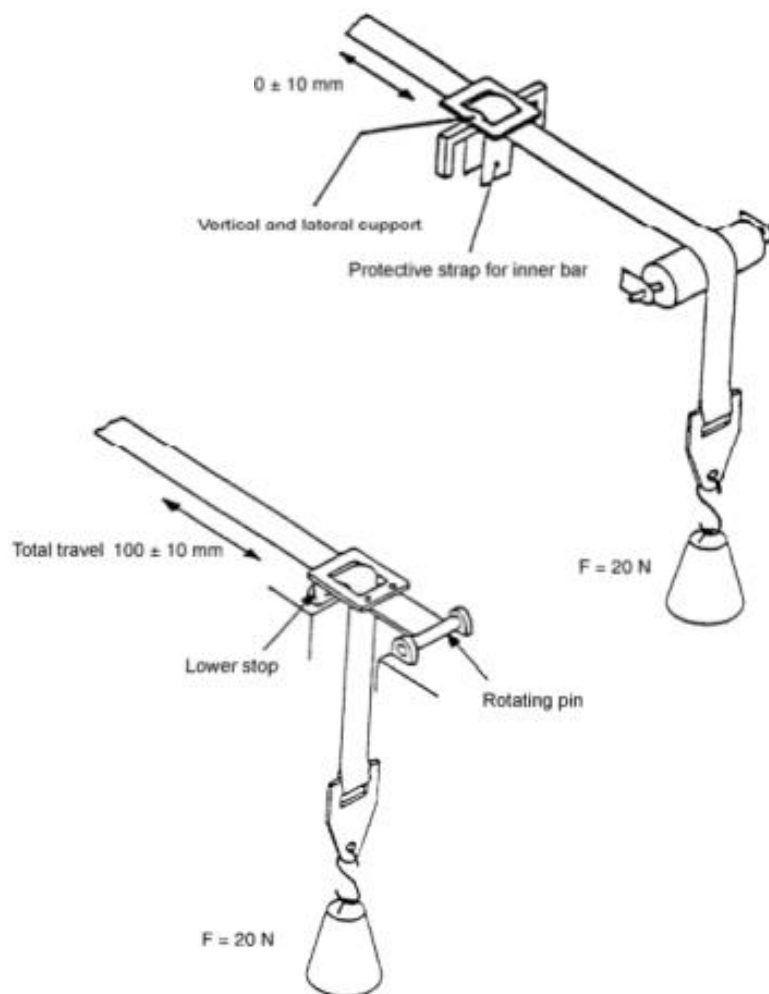


Figure D-1: Apparatus for testing abrasion of the chin strap

This test is designed for assessing the long-term strength of webbing type straps, so is not directly applicable to tanker shell material. However, it could be adapted such that the 'vertical and lateral support' in the figure above become the friction surface. This could be a grinding block or a hand file with a known surface roughness for example. The sample could be attached to a strap at each end with the end of one strap being attached to a weight and the strap at the other end attached to a motor with a reciprocating action so the sample is moved back and forward over the abraded surface. HSE SD have carried out previous work on abrading webbing samples used in fall arrest lanyards. An example test rig (that was built and used) is shown in Figure D-2 below.

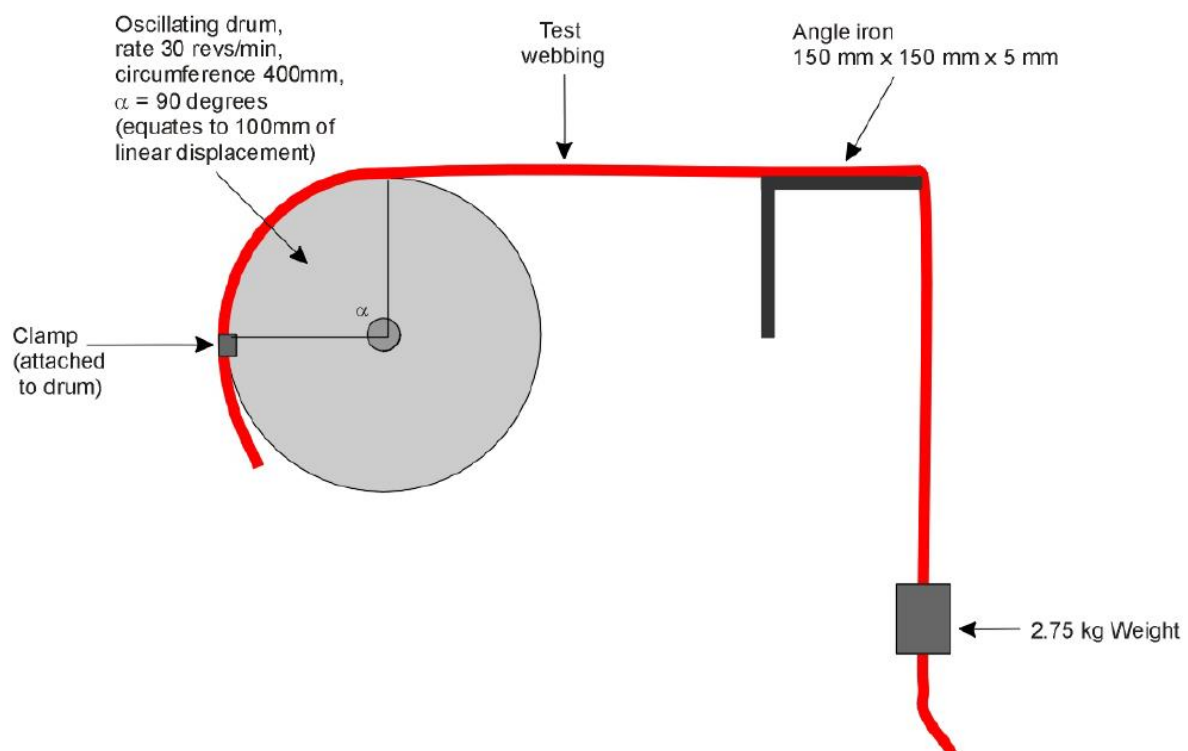


Figure D-2: Example test set up for an abrasion test of webbing

Developing a rig similar to this to carry out abrasion tests of a tanker sample should be straightforward and not require specialist equipment.

Comment As this method requires a reciprocating action, which is different to the sliding action that would occur during a real-life tanker rollover, it was decided that the method did not sufficiently represent the real-world situation. Therefore, it has not been recommended as a test method.

D.1.2 *Helmet Tests*

D.1.2.1 Linear impact –energy absorption test

This test is in clause 7.3 and Annexes 5 and 6 of (UN, 2021).

The helmet is dropped in a guided free fall at a specific impact velocity onto a steel anvil with a circular impact face, and also a kerbstone anvil which has two sides in a similar form to a kerbstone.

Comment As there is no abrasive part to this test, it is not considered relevant for this work.

D.1.2.2 Test for projections and surface friction’.

This test in clause 7.4 and Annex 8 is a helmet test to look at the forces generated at the surface of the helmet when it passes over an abrasive surface. The test considers rotation-inducing forces on the body caused by projections on the helmet and friction against the outer surface of the helmet.

The helmeted headform²¹ is dropped onto an inclined anvil (15° to the vertical) which has a surface of abrasive paper firmly fixed to the anvil. The force along the longitudinal axis of the anvil is measured as shown in Figure D-3. This is method A in clause 7.4.1.

²¹ Headforms are made from metal and vary in size with a mass range of 3 kg to 6 kg depending on the helmet size.

E/ECE/324/Rev.1/Add.21/Rev.5
 E/ECE/TRANS/505/Rev.1/Add.21/Rev.5
 Annex 8

Figure 1b
 Example of a suitable test apparatus for projections and surface friction (method A)

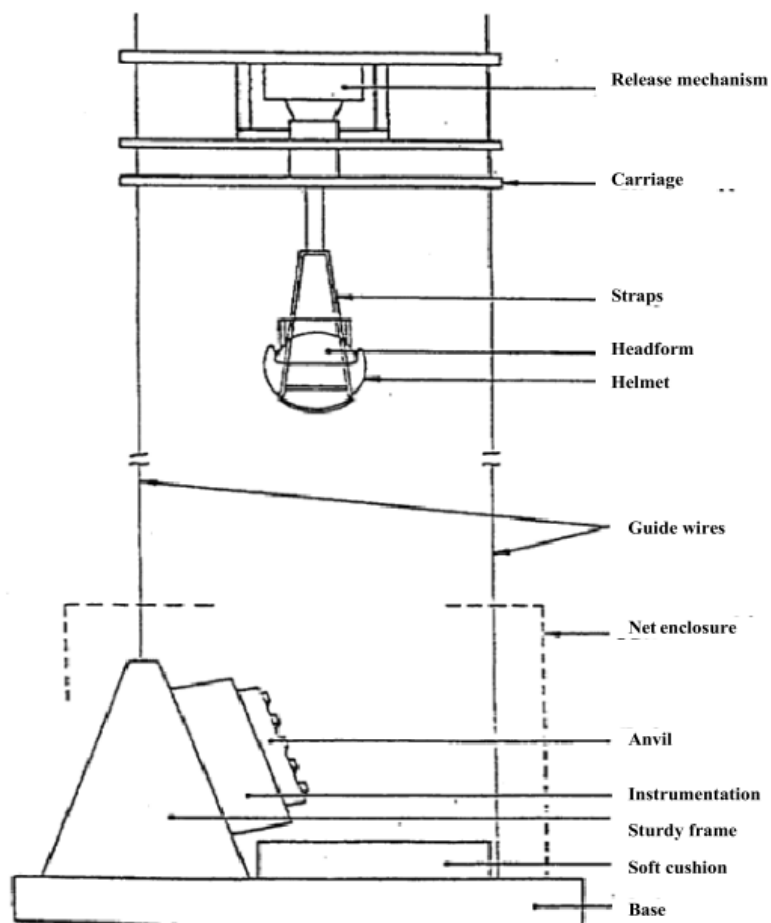


Figure D-3: Helmet test method for projections and surface friction (method A)

Method B in clause 7.4.2 is shown in Figure D-4.

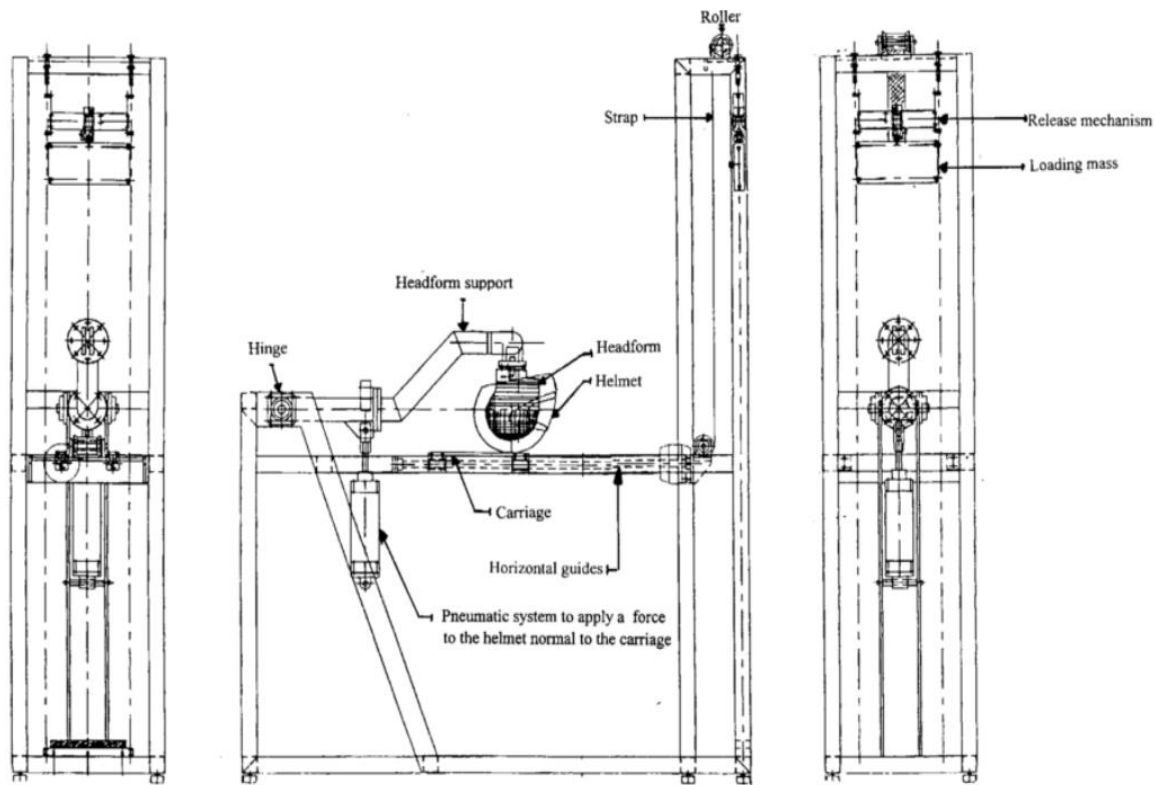


Figure D-4: Helmet test method for projections and surface friction (method B)

From clause 7.4.2.2.1, the test apparatus is as follows:

- (a) A horizontal guided carriage with interchangeable attachments for abrasive paper or a shear edge;*
- (b) A horizontal guide and support for this carriage;*
- (c) A roller with a wire rope or a strap or a similar flexible connection;*
- (d) A lever connecting the headform to the test apparatus with a hinge;*
- (e) An adjustable system supporting the headform;*
- (f) A drop weight to load the lower end support of the wire rope, or a strap, after the weight is released;*
- (g) A system to support a headform and to apply a force to the helmet normal to the carriage.*

A shear impact and abrasion, due to movement of the carriage which is propelled by the movement of the drop weight, acts upon the helmet.

Comment Although both these methods are assessing the helmet response on an abrasive surface. The energy levels in these tests are too low to test a tanker material. For method B, it may be possible to build a much larger rig to carry out abrasion tests on tanker

materials (longer carriages, higher carriage speeds, greater drop heights etc.) but this is unlikely to be practical.

D.2 Standard abrasion tests

D.2.1 Blade-on-block Testing

Blade-on-Block abrasion testing utilizes an object (block) that articulates back and forth on a stationary specimen (blade) while being subjected to a constant normal load. Some examples of behaviour that are characterized by this test are lifespan of coatings and coefficient of friction.

Blade-on-Block testing is useful when a specimen needs non-standard environmental conditions or a higher load force than the Pin-on-Disk test method (described below).

The adaption of the helmet chin strap test described in section D.1.1 is an example of a blade-on-block test.

D.2.2 Taber Abrasion Testing

Taber Abrasion is a quick and inexpensive procedure designed to compare the wear rate and mass-loss of one or more material or coating. A typical Taber abrasion test consists of a disk-shaped specimen that is placed in constant contact with abrasive wheels which produce rub-wear abrasion by contact of the test specimen against the sliding rotation of the two abrading wheels. Predetermined loads, which can be varied between from 250 g to 1000 g, are applied, and a specified number of cycles are undertaken to determine wear. The most commonly used standards for these tests are: ASTM D4060, ASTM F1978-12 and MIL-A-8625. It is a quick and simple way to measure wear resistance and offer sufficient comparable data. It gives a side-by-side comparison of several materials or coatings, so you can evaluate which material has better wear resistance under simulated, accelerated wear conditions (Figure D-5). Loads of

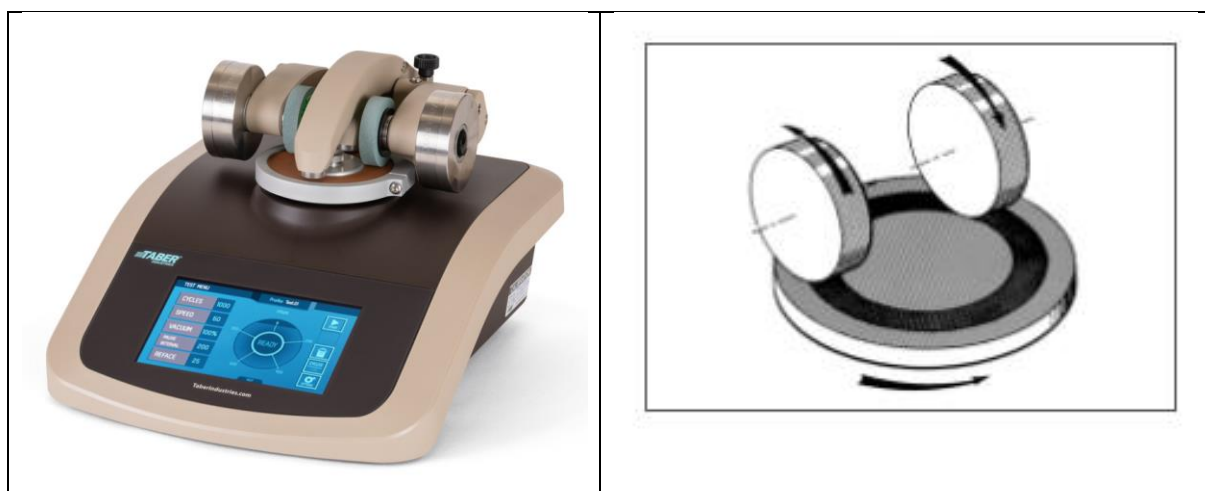


Figure D-5: Taber abrasion testing (photos courtesy of Taber Industries)

Comment This method is not suitable because test setup aimed at much lower loads and amount of wear than required to represent abrasion of tanker.

D.2.3 *Pin-on-Disk wear testing*

Pin-on-Disk wear testing involves abrading two materials – one material is machined into a pin, the other into a disk – to determine a variety of wear properties.

The sample (like a record needle) is pushed up against a horizontal wheel that rotates like a record turntable.

This test is used for comparing tribological properties of a pair of materials including frictional force coefficients and wear rates of your specific material pair.

A test material, e.g. a steel ball or a pin, is moved on a circular path over the specimen with a defined standard force (approx. 5 N). During the test, the friction force or the coefficient of friction is measured continuously. The wear is subsequently determined by measuring the wear track and the abrasion of the test specimen (Figure D-6).

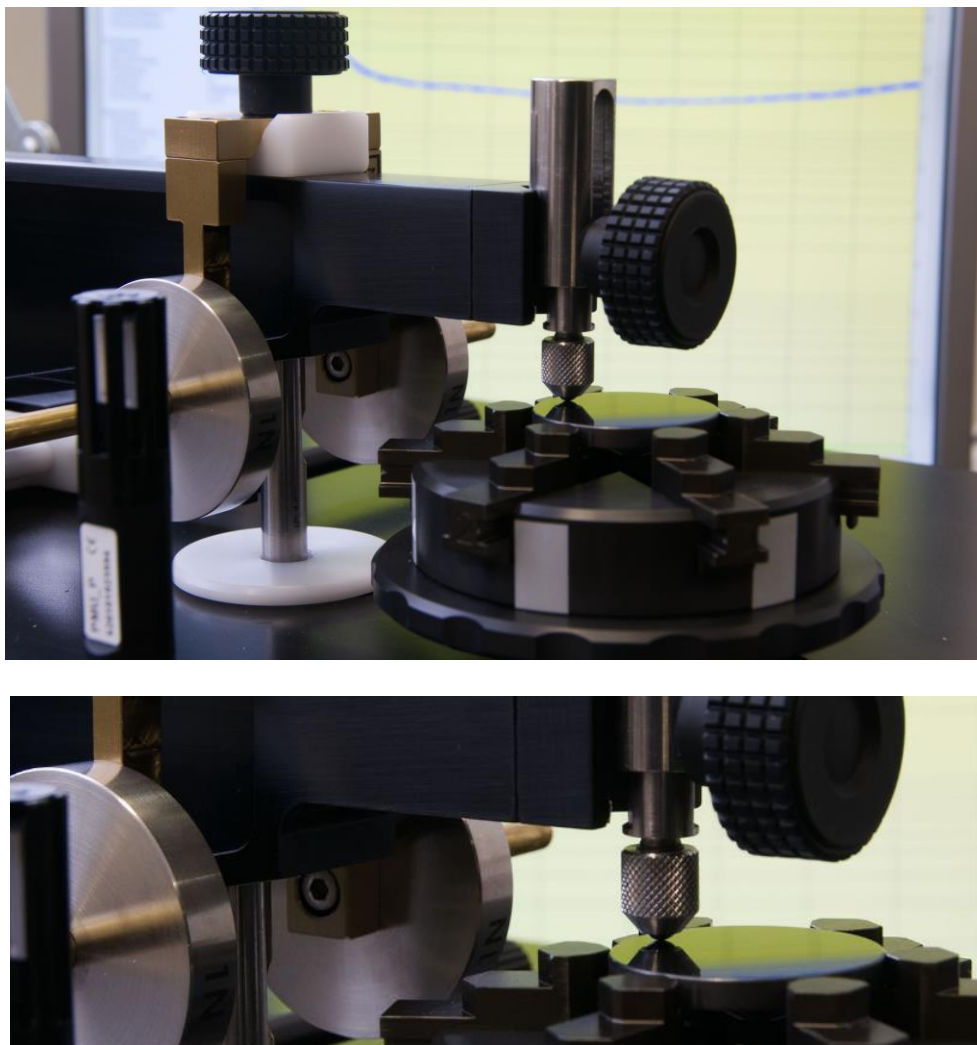


Figure D-6: Test set up for pin-on-disk wear testing (image courtesy of Fraunhofer IST)

This test is widely used when trying to select an appropriate material couple for medical, manufacturing, or other applications. Pin-on-Disk can also be conducted at elevated

temperatures or in submerged environments to simulate “real life” wear conditions more accurately. Typical specifications include ASTM G99-05, ASTM G132, ISO 18535, DIN 50324-07.

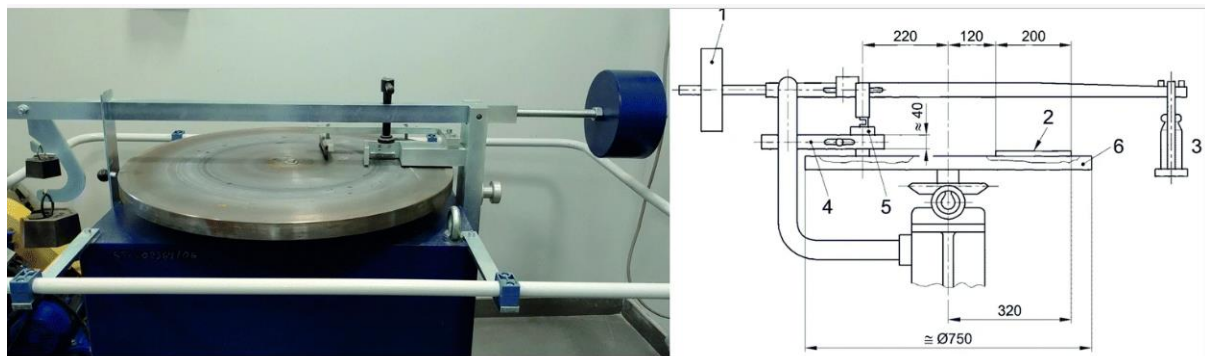
Comment This method is not suitable as the sample sizes will be too small and are the wrong shape. Also, we are not interested in testing a pair of materials.

D.2.4 Block-on-disk abrasive wear test

This is similar to the pin-on-disk method, but the sample is block-shaped rather than pin-shaped so it is larger.

This test is used in the construction industry to test the abrasion of paving slabs. Abrasive material is poured onto the disk in a controlled way during the test.

One example is the Bohme abrasion test apparatus (Figure D-7), where a 100 mm x 70 mm stone sample is pushed against a rotating disk with a force of about 30 kgf. Abrasive material is added during the test.



Bohme abrasion test apparatus. 1—counterweight, 2—test track, 3—loading weight, 4—sample holder, 5—sample, 6—rotating disc (EN 14157 2017)

Figure D-7: Bohme abrasion test apparatus (from Strzalkowski et al, 2020)

Comment This standard test method could possibly be adapted for a tanker abrasion test. The abrasive surface would need to be specified and a suitable abrasive wheel that can be fitted to the Bohme turntable would need to be designed. It is in principle similar to a grinding wheel type method but likely more complicated to redesign a rig because it may be necessary to go back to the manufacturers of the machine and ask for some fundamental re-designs. On this basis it was decided to take the grinding wheel method forward and not recommend this method at this stage. However, note that this decision could be re-considered.

D.3 Grinding polishing methods

Grinding of metals and glass are often carried out on vertical axis grinding/polishing machines such as the one in Figure D-8 below.

However, these machines are designed for fine finishing of materials such as hand-preparation of samples in metallography, and glass and gem samples. Therefore, they are

not considered suitable as the wheels will not be abrasive enough and a system of applying a repeatable load to the sample would need to be designed.



Figure D-8: Example of a grinding and polishing machine (photo courtesy of HSE SD)

Another option would be to abrade the sample on a universal grinding machine with a horizontal axis grinding wheel. The sample could be fitted between head and tail stocks and then the central part of the sample abraded against the grinding wheel (moved back and forth for a set number of cycles). This would still require a method of applying a controlled load against the grinding wheel which may be difficult to repeat accurately (Figure D-9) on a proprietary grinding machine.

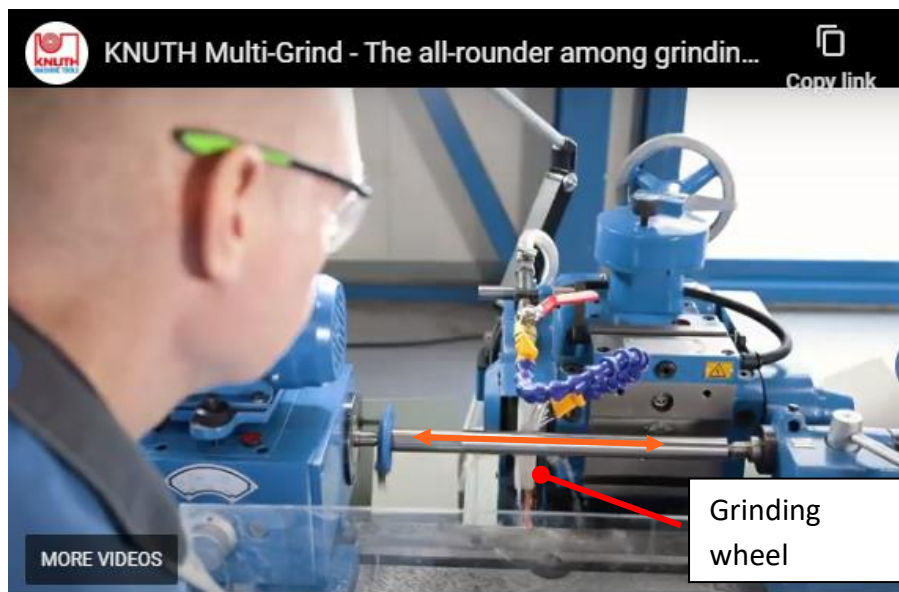


Figure D-9: Universal grinding machine (photo courtesy of Knuth)

Comment This method would be difficult to achieve reliable repeatable results without a method of ensuring that the load the sample applies to the grinding wheel can be measured accurately and is controlled and repeatable. Therefore, this method is not recommended.

Appendix E Calculations for the Outline Technical Code

This appendix describes the work done to establish the appropriate impact energy for the drop test method specified in the Outline Technical Code. This includes modelling a subsection drop test based on the rear two compartments of a banded type 1 tanker and verification of the method by comparing the results to the model of a complete tanker.

E.1 Impact energy of complete tanker topple

The first step in deciding the impact energy for a subsection drop test is to evaluate the impact energy that we are trying to replicate, i.e. a complete tanker topple with a full load of petrol.

Tanker designs may vary. If the only variation was length (with the associated change in overall mass), a standard rotational velocity at impact could be set, and the impact energy would vary by overall mass.

However, to allow further variations to the design which could result in a change to the rotational velocity at impact (such as a change in the height of the CoG), a different method of establishing the impact energy is needed.

For a topple event, where the initial state at the point of topple is stationary (no kinetic energy), all the kinetic energy at impact will arise from the change in potential energy. As the mass stays the same, the only parameter that needs to be evaluated is the change in height of the CoG.

As the topple point will be when the CoG is directly above the pivot line (the outer edge of the wheels), the height of the CoG will be equal to the distance of the CoG from the pivot line, as shown in the following diagram. The angle of the tipping point does not need to be known.

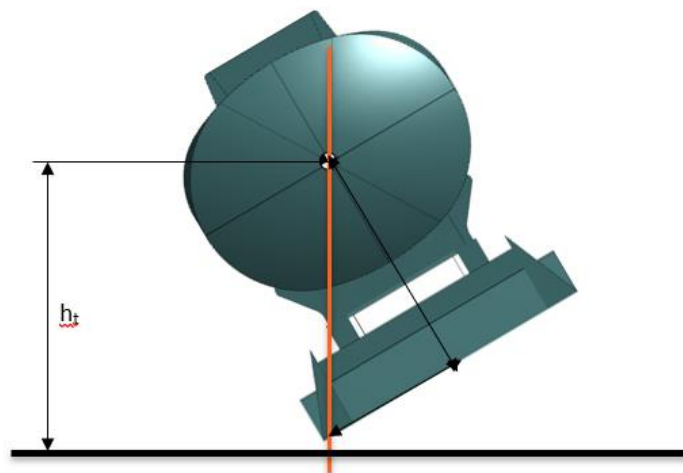


Figure E-1: Tanker at the tipping point with the centre of gravity (CoG) over the edge of the wheels

The masses and heights of the centres of gravity when upright for the different components of a tanker are shown in Table E-1.

These figures are based on the complete tanker model for a banded type 1 tanker that was tested. This will be slightly different to a real-world case, as some modifications were made for the test, such as stripping off unnecessary ancillary parts, replacing the wheels with steel supports and replacing the tractor unit with a steel support (referred to here as the 5th wheel).

The contribution each component makes to the overall change in potential energy is also listed. By far the largest contribution is from the petrol load (94%). The tanker shell makes up 5% if the partitions and ends are included. The table shows a contribution of 1.7% for other structural components and ancillary equipment. This includes manways and the valance²². In the original model, some of these parts were thickened slightly to increase the weight of the modelled tanker to more closely match the physical tanker tested. Therefore, this may be an overestimate.

The other parts (supports and running gear) have low centres of gravity, so their contributions are smaller or even negative. A negative contribution arises when the CoG is higher when in the impact position than it is at the topple position.

Table E-1: Components of a tanker with centres of gravity based on model of banded type 1 tanker

Component	Mass (tonnes)	Height of CoG (m)	Potential Energy Contribution (%)
Tanker shell	1.1	2.1	3.4
Partitions	0.5	2.1	1.6
Additional structural components (top openings etc)	0.5	2.4	1.7
Rear supports (longitudinal members supporting the rear compartments and brackets)	0.2	0.9	0.4
Front supports (tanker supports under the front compartment and brackets)	0.1	1.3	0.5
Running gear (suspension, axles and wheels)	1.9	0.5	-0.9
5 th wheel support (structure used for the topple test)	1.1	0.5	-0.5
Petrol	31.4	2.1	94.1

²² The valance is the covering around the top openings of the tanker. It may be referred to as the 'comb' or 'rollover rails'

The approach used here should be accessible for manufacturers when designing a new tanker without the need for complex modelling. CoG information should be available from computer drawings or could be estimated with reasonable accuracy. If the details of the supports or running gear were not known at the time of assessment, these could be ignored for this calculation, as their contribution is small and likely to be negative (although the overall height of the tanker must be known accurately). Therefore, leaving them out of the calculation would be likely to result in an increase in impact energy, and therefore be conservative.

The most difficult component to assess accurately would be the liquid load, partly due to the shape but mainly due to the shape changing as the tanker moves. Using an assumption that the compartments are full of liquid for the purposes of determining the location of the CoG would avoid this problem. The resulting increase in potential energy due to a slightly higher CoG would be small (approximately 5%) if the actual liquid mass was used (i.e., although the CoG is calculated on the basis of a completely full compartment, the mass would still represent the actual fill level).

To check this method, the change in potential energy calculated using this approach was compared to model results. In Phase 2 of this project, a simple model of a single compartment was run from the topple point to the impact point. The resulting liquid motion at the point of impact was complex due to movement of the liquid relative to the tanker. As it was not possible to accurately map the movement from this model directly to the complete tanker model, a number of simplifications were tried. The most accurate method was to represent the movement as a combination of a linear motion and a rotation about the centre of the tanker compartments. This work is explained in the phase 2 report.

Using this motion for the full tanker with a full load of petrol resulted in kinetic energy of 370 kJ at impact. Calculating the change in potential energy using an accurate estimation of the CoG of the petrol resulted in the same 370 kJ figure. This confirms that this approach is valid to assess the impact energy. Using the simplified centre of gravity approach resulted in a change in potential energy of 387 kJ, an increase of approximately 5%.

E.2 Allowance for differences between topple and drop

The proposed subsection test in the technical code will be a vertical drop test rather than a topple. This is likely to be a more practical test method and would make adjusting the impact energy more straight forward.

However, in a vertical drop, all the initial motion is in the vertical direction, which is stopped by the ground. In a topple, as the tanker and contents are rotating, there is a horizontal component to the motion at the point of impact. Therefore, not all the initial kinetic energy will be stopped on impact. As a result, for the same initial impact energy, more deformation occurs in a drop test than in a topple. This must be accounted for.

From the complete tanker topple model with a full petrol load, the kinetic energy remaining after impact was approximately 100 kJ, i.e. 27% of the initial impact energy.

In the parametric study, many models were run of banded type 1 subsection drop tests consisting of one to three compartments. These had different compartment length, fill levels, liquid densities and impact energies. When plotting deflections against impact energy

per partition, all the points could be fitted to a power law curve. Although there was some scatter, an R^2 value of 0.94 was obtained, which indicated good correlation.

The complete tanker topple test (benchmark topple unadjusted) had an average deflection that lay somewhat below this power law curve, as shown in Figure E-2.

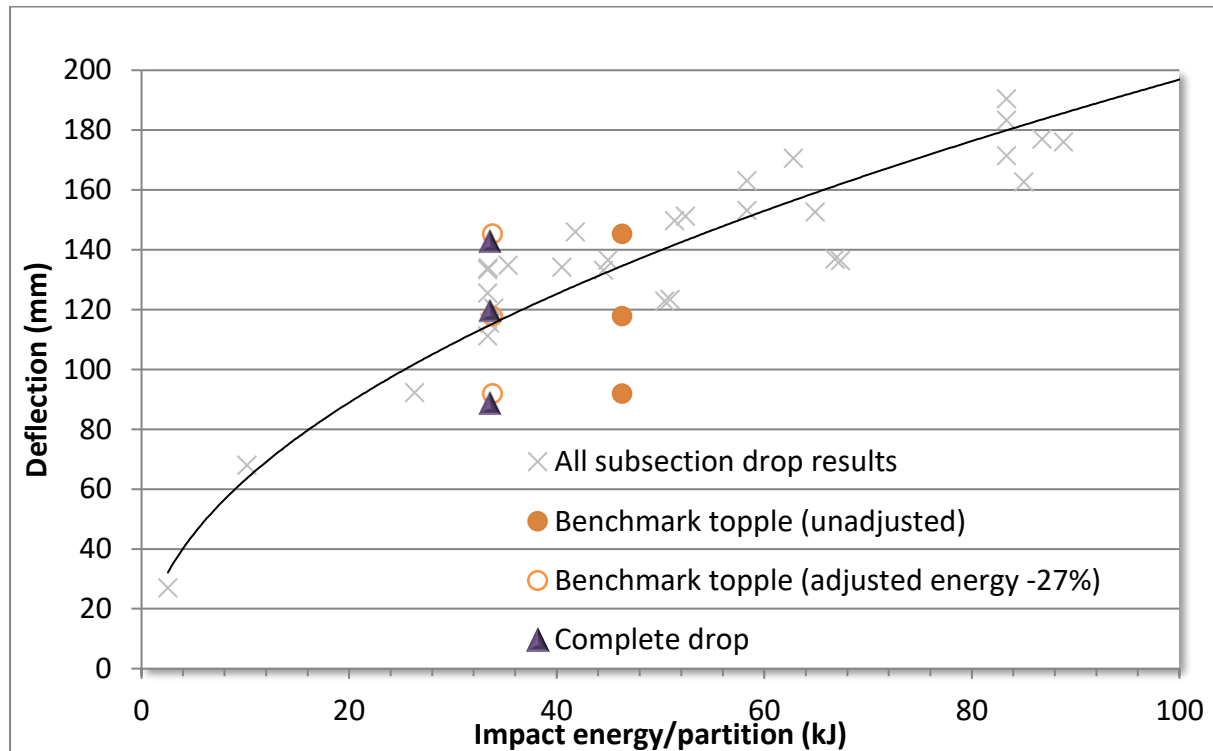


Figure E-2: Benchmark topple results (with adjustment) and drop model results compared to subsection model results

Three points are shown for the topple, presenting the maximum deflection, the average and the minimum deflection. For a complete tanker, the deflection varies along the length, as discussed in the previous section.

As 27% of the kinetic energy at the point of impact remains as kinetic energy in the topple case, reducing the impact energy by 27% to account for this gives the adjusted results in Figure E2. With this adjustment, the average deflection for the topple lies very slightly above the curve.

As a final check, a complete tanker was modelled in a vertical drop scenario, with the impact energy equal to the adjusted topple energy. The average deflection was very close to the topple average deflection, showing that the impact energy adjustment had correctly removed the difference between topple and vertical drop scenarios for this tanker.

The adjustment is to allow for the differences between topple test and vertical drop test that arise due to the difference in the direction of the motion. Therefore, the adjustment should be reasonably independent of tanker design. One aspect that might have an impact on the differences between the two impacts would be a significant change to the height of the CoG. However, large changes are unlikely as the CoG will be kept low for stability reasons, but the extent to which it can be lowered is limited due to the running gear.

E.3 Allowance for differences along the length of the tanker

For complete tankers, the deflections are higher at the rear as discussed in the previous section. This trend was observed in the banded type 1 tanker physical tests and models (both topple and drop), and also the physical tests of banded type 2 and stuffed designs.

Reasons for this are due to differences in the distribution of the mass and stiffening elements (partitions and ends). If all the stiffness and energy absorption characteristics for the partitions are known, then the pattern of deflections may be predicted for a topple or a vertical drop. However, it is unlikely that all the necessary information would be known, and there is some uncertainty about the physics underpinning the calculations.

Therefore, in the technical code, for the test option, where the approval is based on a subsection drop without the need for any finite element analysis, a standard adjustment is needed to take the variation in deflection along the length into account (see option 3b in the Outline Technical Code).

For the benchmark topple model (banded type 1 with a full petrol load), the maximum deflection at the rear was approximately 25% higher than the average deflection. However, to get 25% higher deflection, a bigger increase in energy is needed.

The equation for the power law curve, that fits the data for deflections against impact energy per partition, is approximately:

$$d = 20.3E^{0.5}$$

where d is deflection and E is impact energy per partition. Therefore, to get a 25% increase in deflection, an increase in energy of 56% is needed ($1.25^2 = 1.56$). This factor has been used to obtain the impact energy per partition for the subsection drop test in option 3b in the Outline Technical Code

The banded type 2 and stuffed designs had different deflection/energy relationships, but the exponent of the best fit curve was still approximately 0.5.

This factor would be appropriate for the banded type 1 tanker that has been modelled. No other design has been modelled as a complete tanker, so the variation along the length of the tanker is not known for other designs. Also, the data set for banded type 1 tankers is limited. Therefore, an additional uncertainty factor is appropriate. This could be avoided by modelling a complete tanker topple, as in Option 3a of the Outline Technical Code.

E.4 Calculating the subsection drop test impact energy and drop height (worked example)

For the banded type 1 tanker modelled, when in a horizontal position, the height of the CoG was 1934 mm when using the actual 95% fill level, or 1963 mm if assuming the compartments were 100% full of petrol. The horizontal distance from the pivot line (outer edge of the wheels) to the CoG was 1279 mm.

Therefore, the height of the CoG at the tipping point was:

$$h_t = \sqrt{(1924 \text{ mm})^2 + (1279 \text{ mm})^2}$$

$$h_t = 2311 \text{ mm}$$

The total mass of the tanker (including the petrol load and all the supporting structures) was 36 768 kg. At the impact point, h_i , the height of the CoG can be assumed to be half the width of the tanker, which is 1275 mm. Therefore, the change in potential energy is:

$$\begin{aligned}\Delta PE &= (h_t - h_i) \times g \times m \\ \Delta PE &= (2.311 \text{ m} - 1.275 \text{ m}) \times 9.81 \text{ m}\cdot\text{s}^{-2} \times 36768 \text{ kg} \\ \Delta PE &= 374 \text{ kJ}\end{aligned}$$

This value obtained was slightly higher than the kinetic energy at the point of impact for the benchmark topple model (370 kJ), although the difference is only 1%.

Using the assumption that the CoG of the petrol is the same as the CoG of the compartments (i.e. assuming the compartments are completely full for the purposes of defining the CoG, but not mass), results in the combined CoG of the tanker being 29 mm higher when the tanker is upright. This translates to an increase in potential energy up to 385 kJ, an increase of 3% compared to the 95% fill level calculation.

For the purposes of this illustration, the kinetic energy at impact for the benchmark model will be used for further calculations (370 kJ). This is so that a true comparison of the subsection drop model can be made against the benchmark topple. It is acknowledged that using the potential energy calculation is likely to result in kinetic energy values a few percent higher.

Next, the kinetic energy for the topple is reduced by 27% to take into account the differences between a topple test with rotation, and a straight vertical drop test.

Topple to drop adjustment:

$$\begin{aligned}KE_{Drop} &= 0.73 \times KE_{Topple} \\ KE_{Drop} &= 0.73 \times 370 \text{ kJ} \\ KE_{Drop} &= 270 \text{ kJ}\end{aligned}$$

The impact energy per partition is then calculated, using the number of partitions/ends in the complete tanker. Stiffening elements that are significantly different to the standard partitions, such as stiffening rings, should not be included. Surge plates, provided they are of a similar design to partitions, should be included provided that the hole in them does not exceed a certain proportion of the cross-sectional area (30% of the area is currently suggested²³). The number of partitions, n_{tot} , in the benchmark model is 8 (including the ends and a surge plate in the front compartment).

$$\begin{aligned}KE_{part} &= \frac{KE_{Drop}}{n_{tot}} \\ KE_{part} &= \frac{270 \text{ kJ}}{8} \\ KE_{part} &= 33.8 \text{ kJ}\end{aligned}$$

The adjustment for variation in deflection along the length must then be made. For the benchmark topple, the rear end of the tanker deflected by approximately 25% more than

²³ Further modelling could be carried out if needed to get a more accurate understanding of how surge plates with larger holes reduce the stiffness of the tanker

the average of the other partitions. To get the change in energy required, this adjustment should be squared.

$$\begin{aligned} KE_{partAdj} &= KE_{part} \times 1.25^2 \\ KE_{partAdj} &= 33.8 \text{ kJ} \times 1.56 \\ KE_{partAdj} &= 52.7 \text{ kJ} \end{aligned}$$

If option 3a is used, where the variation along the length is assessed for the actual tanker using the finite element method, the 25% value can be replaced with the value obtained from the assessment. However, the new value should still be squared.

The subsection to test will be performed on the rear two compartments unless modelling suggests other compartment will have a higher deflection (option 3a). The number of partitions/ends in the subsection to be tested, n_{sub} , is therefore likely to be 3, unless either of the compartments being tested would normally contain a surge plate. These should only be included if they were included in the total number of partitions for the complete tanker. Any stiffening elements not counted should not be included.

Total impact energy for test subsection:

$$\begin{aligned} KE_{Test} &= KE_{partAdj} \times n_{sub} \\ KE_{Test} &= 52.7 \text{ kJ} \times 3 \\ KE_{Test} &= 158 \text{ kJ} \end{aligned}$$

The mass of subsection with a water fill to 95% capacity, m_{sub} , was 13 702 kg.

The impact velocity can then be calculated:

$$\begin{aligned} v &= \sqrt{\frac{2 \times KE_{Test}}{m_{sub}}} \\ v &= \sqrt{\frac{2 \times 158 \text{ kJ}}{13702 \text{ kg}}} \\ v &= 4.8 \text{ m/s} \end{aligned}$$

From the impact velocity, the drop height, H , can be calculated:

$$\begin{aligned} H &= \frac{v^2}{2 \times g} \\ H &= \frac{(4.8 \text{ m/s})^2}{2 \times 9.81 \text{ m/s}^2} \\ H &= 1.17 \text{ m} \end{aligned}$$

However, H can also be calculated from:

$$\begin{aligned} H &= KE_{test} / (m_{sub} \times g) \\ H &= 158 \times 10^3 / (13702 \times 9.81 \text{ m/s}^2) \\ H &= 1.17 \text{ m} \end{aligned}$$

Therefore, the impact velocity does not need to be calculated to obtain the drop height.

E.5 Subsection drop test modelling using real-world tanker dimensions

E.5.1 Geometry

The geometry for the subsection model was based on the rear two compartments of the banded type 1 tanker, as shown in Figure E-3. The lengths of the compartments were 1.85 m for the rear compartment (compartment 6) and 1.31 m for the second-to-last compartment (compartment 5). In this subsection, both of the end dishes are orientated so that they bow-out of the compartments (convex from the outside), so will be loaded in the normal way. It is uncertain at this stage how significant it would be on the impact response if one of the subsection ends was reversed²⁴.

As for the earlier two-compartment subsection models, one plane of symmetry was used (the vertical plane along the length of the tanker).

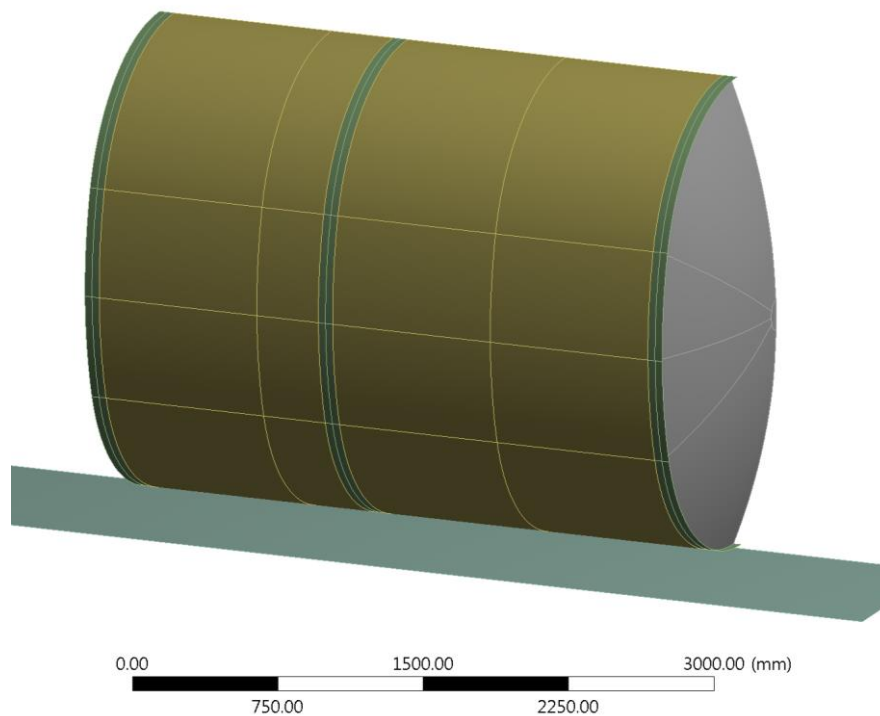


Figure E-3: Rear two compartments modelled for the subsection drop test

E.6 Test parameters

The parameters used for the model of the drop test were as follows:

²⁴ If one of the end subsection partitions is in reversed orientation (i.e. concave when looking from the outside of the subsection), then this may alter the response dynamics slightly. At this stage it is uncertain how this will change the overall deflections and forming limit. Although the changes may not be significant, it may require further 'tuning' of the drop height to maintain equivalence between the complete tanker and subsection.

- The fill level used was 95%. This gave the maximum mass for the size of the compartments tested, meaning that the drop height could be kept to a minimum while achieving the required impact energy.
- The liquid used was water with a density of 1000 kg/m^3 . Use of any other liquid would be either prohibitively expensive or have safety and/or environmental concerns.
- The impact speed was 4.8 m/s as calculated in Section E.4, to give an impact energy of 52.7 kJ per partition from a total mass of $13\,702 \text{ kg}$.

E.7 Results

Unlike the previous two-compartment subsection models, there was a variation in deflection values between the two ends. All other models had compartments of the same length, and although there was a slight variation between the end and central partitions in some models, there was not a significant difference between the two ends.

The aim of the exercise was to model a subsection representing the rear two compartments of a tanker that would match the proposed subsection drop test. The target was to achieve deflections in this subsection model that match the maximum deflection in the benchmark petrol topple model (145 mm). The average deflection value from the subsection model was 141 mm , with maximum and minimum values of 149 mm and 133 mm , as shown in Figure E-4. Therefore, while the average deflection was slightly below the target value (by 2.8%), the maximum deflection was slightly above the target value (2.3%).

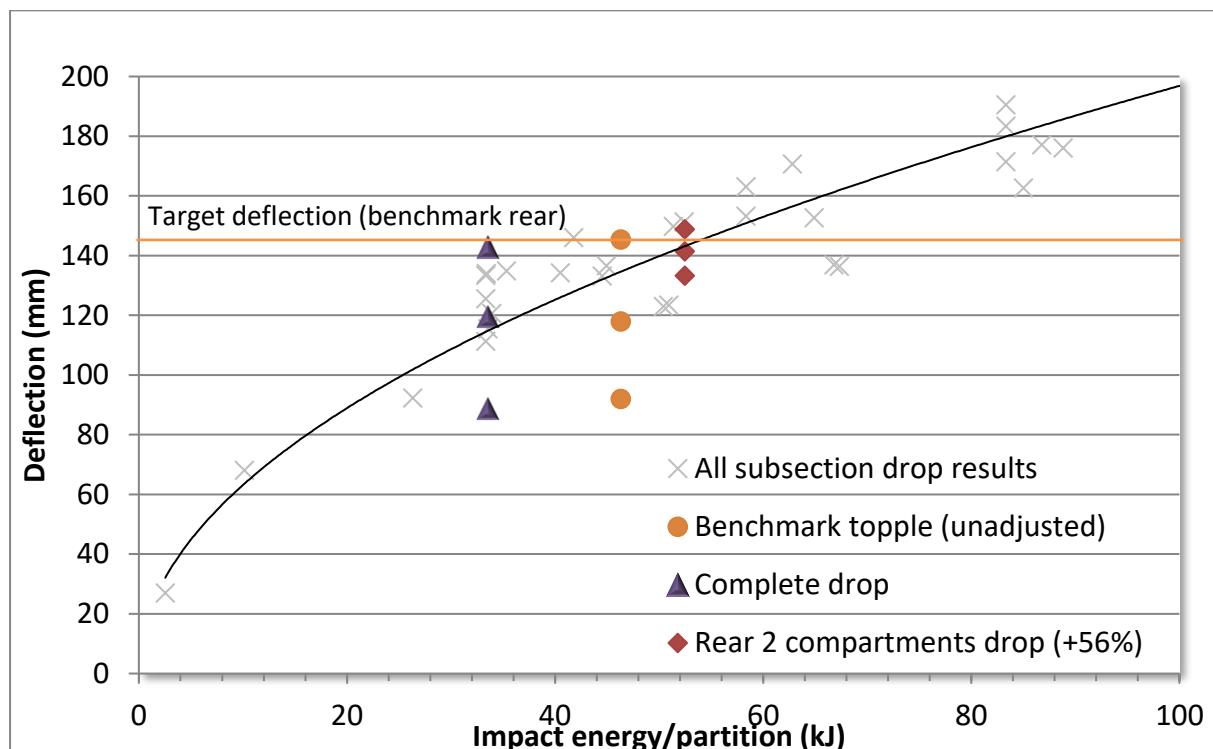


Figure E-4: Results from the subsection model compared to benchmark topple and complete tanker drop models

The above results are based on the impact energy of the benchmark topple model (370 kJ). Calculating the energy for the subsection drop test based on change of potential energy and a simplified CoG for the petrol would result in the impact energy for the drop test being about 4% higher than the value used. With the deflection being proportional to the square root of the impact energy, this would be likely to result in a deflection about 2% higher, bringing the average deflection very close to the target value. However, the highest deflection would then be about 5% higher than the target value.

It should be remembered that the main failure mechanism, which is exceeding the forming limit of the material, is fairly independent of the deflection over the range of interest as shown in Figure E-5.

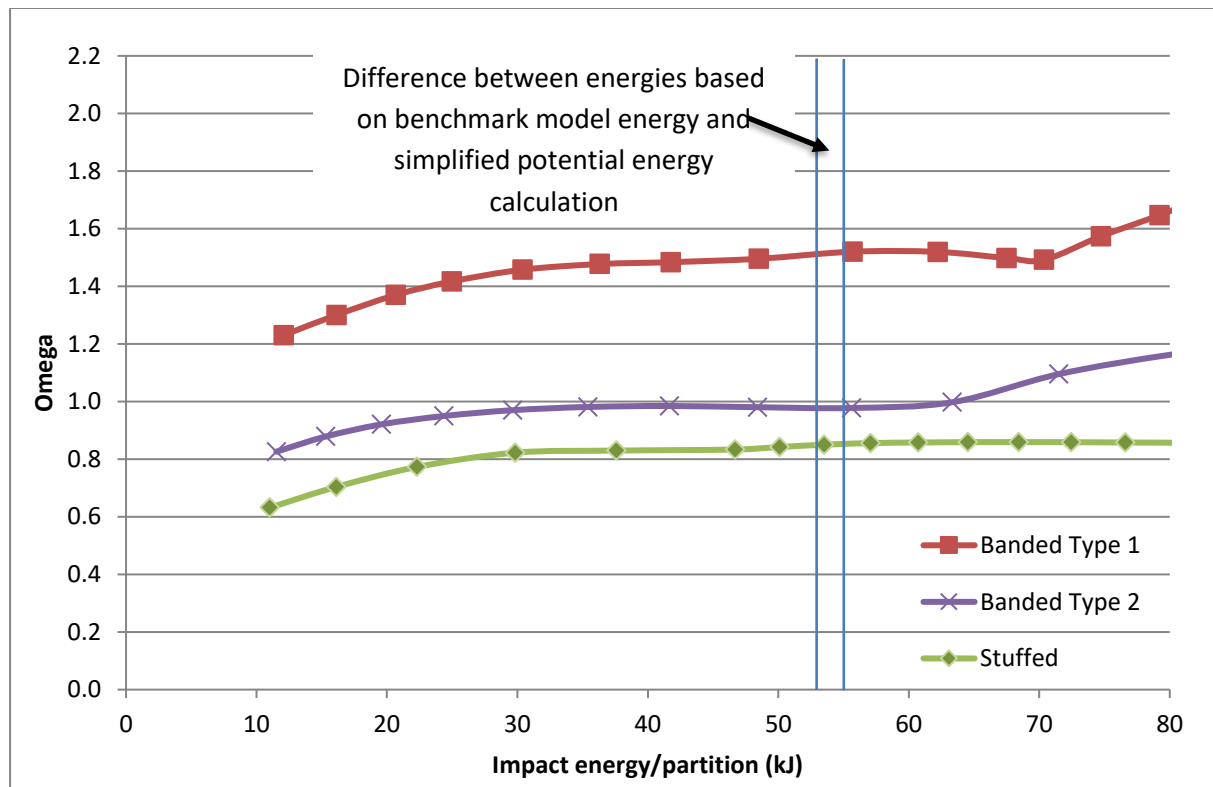


Figure E-5: Effect of methods of estimating the impact energy per partition on omega (forming limit) values

Therefore, an increase of 5% in the maximum deflection is not likely to alter the results of the test.

Appendix F Additional information: development of performance-based tests – frontal impact

F.1 Loading methods

Three different loading methods were used: Static (displacement controlled), quasi-static loading (force controlled), and a dynamic model applying an acceleration to the kingpin location while adding mass to the rear compartments of the tanker. In order to greatly increase the solution speed, none of these models contained fluid. These three loading methods were chosen to model the different tanker responses that may arise when the load is applied in different ways. This will help to establish what degree of complexity would be needed to accurately model tanker performance. Adding the fluid load would enable a more realistic dynamic response to be obtained but would increase the solution time disproportionately at this initial stage of research (probably a ten-fold increase in time).

The boundary conditions at the kingpin were applied at a line that was near the top of the kingpin and flush with the skid plate. Therefore, this would slightly underestimate the moment produced by these forces if the force application point on the kingpin was slightly lower than this. Consequently, the responses predicted in this work may be slight underestimates of the actual response. In the model, there was no restraint to limit the amount of rotation of the kingpin unit and front compartment.

F.1.1 Static Models (displacement-controlled loading)

A static loading model was run, using the standard implicit Ansys solver. The loading was achieved by incrementally applying a displacement to the kingpin location up to a maximum of 40 mm (compared to a force in the quasi-static model) while restraining the rear of the tanker. Therefore, the full force was transmitted through every compartment, which is not entirely realistic. Displacement controlled loading was chosen for this model as it can be difficult to get a finite element model to converge on a solution when using load control, especially in highly non-linear cases

F.1.2 Quasi-static Models (force-controlled loading)

In order to apply the loading in a more realistic way, with the forces acting on each bulkhead, the model had to be run with force controlled loading as the relative displacements of each bulkhead would not be known. It can be difficult to get finite element models to converge on a solution if a yield or buckling point is reached as large displacements may occur for a small increase in load (assuming any load increase is possible at all). Therefore, while still using the implicit solver, transient loading was used, which takes into account inertial effects that can sometimes help to obtain converged solutions around sudden changes in stiffness. For the quasi-static force models, the kingpin location was constrained in all directions and forces were increased at a constant rate on each bulkhead in relation to the capacity of the compartment behind that bulkhead, up to the equivalent of 4 *g* over 1 second.

F.1.3 Dynamic Models (solid mass in tanker)

For the dynamic models, the tanker's rear compartments were filled with a solid with the same density as water in the rear four compartments. Solid was used as this greatly simplified the set up and run times for these initial models. The front two compartments were left empty as filling these with a solid would prevent the shell from deforming as it had in the incident. Therefore, the full mass of the tanker (32 tonnes) was somewhat less than the maximum (37 tonnes).

With the tanker stationary, a step-change acceleration was applied in a direction towards the rear of the tanker along the same line as the quasi-static model for 0.1 s. The velocity after 0.1 s²⁵ was then maintained for an additional 0.1 s (total run time was 0.2 s for both models). Therefore, the 2 g acceleration case was equivalent to a change in velocity of approximately 2 m/s (7 km/h) over a time of 0.1 s.²⁶

The models were then re-run with a 3 g acceleration, again over a time of 0.1 s (3 m/s change in velocity). By keeping the time constant, it makes comparison of the load/time traces easier. Shortening the time to keep the overall change in velocity the same would keep the energy change the same but would reduce the length of time for the effect of that acceleration to become established.

F.2 Results

F.2.1 Static models

The results from the static models, loaded using controlled displacement, show a clear buckling point, as shown in Figure F-1. From the initial gradients of the force-displacement lines, it is clear that the 'short unit' model has a lower stiffness before buckling (as the gradient is lower than the gradient for the 'long unit') and a lower buckling load.

Using this loading method, the short unit model reached a maximum load of 713 kN, which is slightly below the 725 kN load that would be equivalent to a 2 g acceleration of a fully laden tanker.

²⁵ The tanker is starting at 'zero' velocity and being accelerated in a rearward direction. This is simulating a deceleration on impact (i.e. the deceleration event is happening in reverse)

²⁶ The kinematic equation in this case is $v = at$ where $a = 2 g = 20 \text{ m/s}^2$ (approx.), $t = 0.1 \text{ s}$. So $v = 20 \times 0.1 = 2 \text{ m/s}$

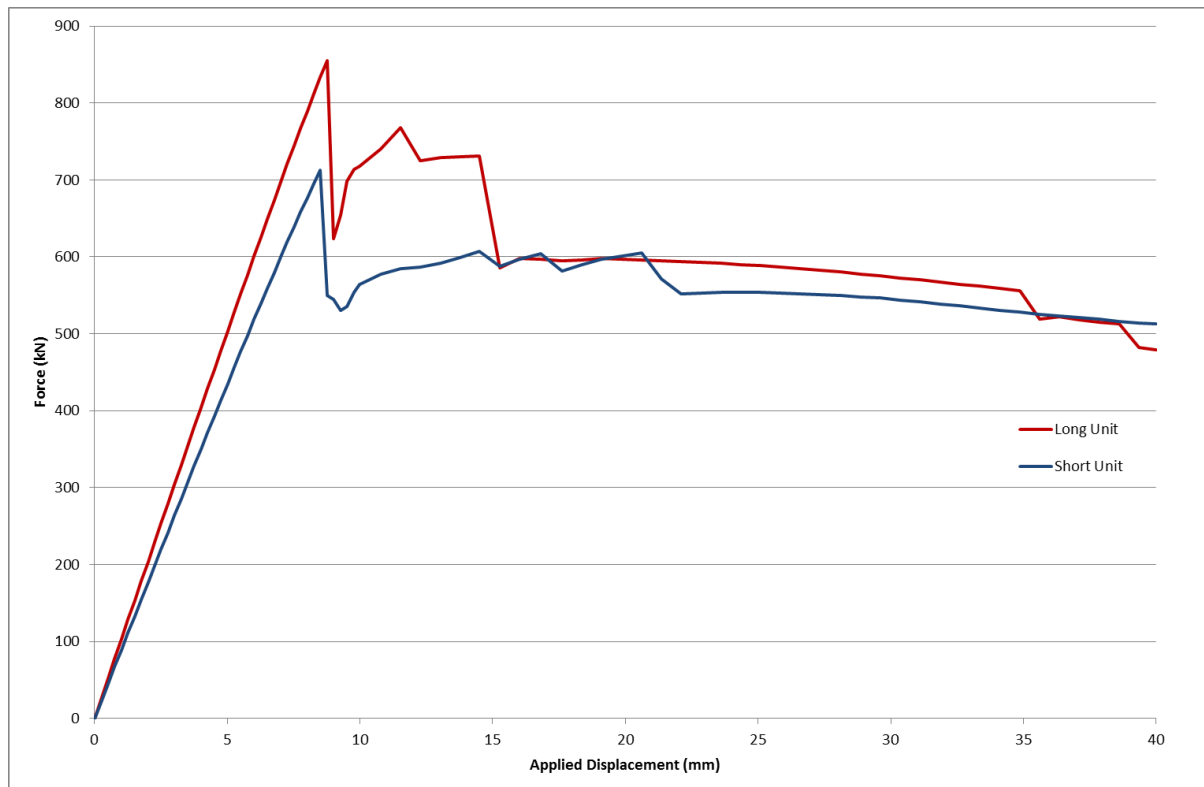


Figure F-1: Results for the static (displacement controlled) model

F.2.2 Quasi-static Models

For the quasi-static models, with the load applied to the bulkheads and increasing at a constant rate, the results are shown in terms of reaction at the kingpin in Figure F-2. The results are shown against time, rather than displacement, as the displacement will vary along the tanker, and the kingpin is being held static. Therefore, the gradient of the loading line is dependent on the loading rate only (same in both models) rather than indicating stiffness as in the static model.

As for the static model, the buckling point is clear from the force plots. As this model is defined by forces applied on the bulkheads, a drop in reaction force at the kingpin is only possible due to the consideration of inertia, and a fairly rapid loading.

As for the static model, the longer kingpin unit can sustain a higher load than the shorter kingpin unit. In both cases, the maximum loads at the kingpin were higher than for the static case, likely to be due to the fact that not all of the load was being transferred through the critical compartments. There is also a larger difference between the two types of kingpin support than was observed in the static model; again likely to be due to the different levels of force in the critical compartments.

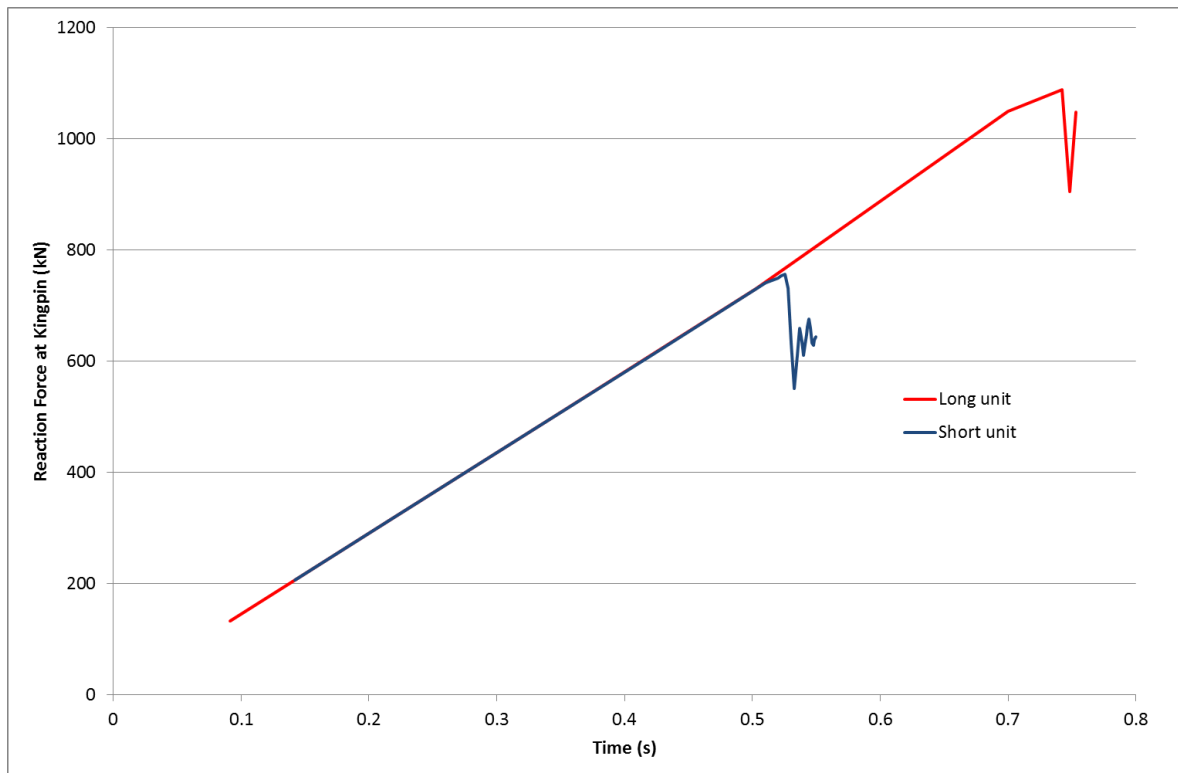


Figure F-2: Results for the quasi-static (load controlled) model

F.2.3 *Dynamic Models (solid mass in tanker)*

As mentioned earlier, the dynamic results were achieved by applying a step-change acceleration towards the rear of the tanker along the same line as the quasi-static model for 0.1 s. The velocity after 0.1 s was then maintained for an additional 0.1 s (total run time was 0.2 s for both models).

Figure 10 shows model results for a 3 g deceleration on both the long kingpin (top) and shorter kingpin (bottom) units. In both cases, the tanker has buckled in the compartment behind the end of the kingpin support. Although buckling of the tanker has occurred in all the models, failure criteria have not been included so a more detailed engineering assessment would need to be made to determine if rupture was likely to occur.

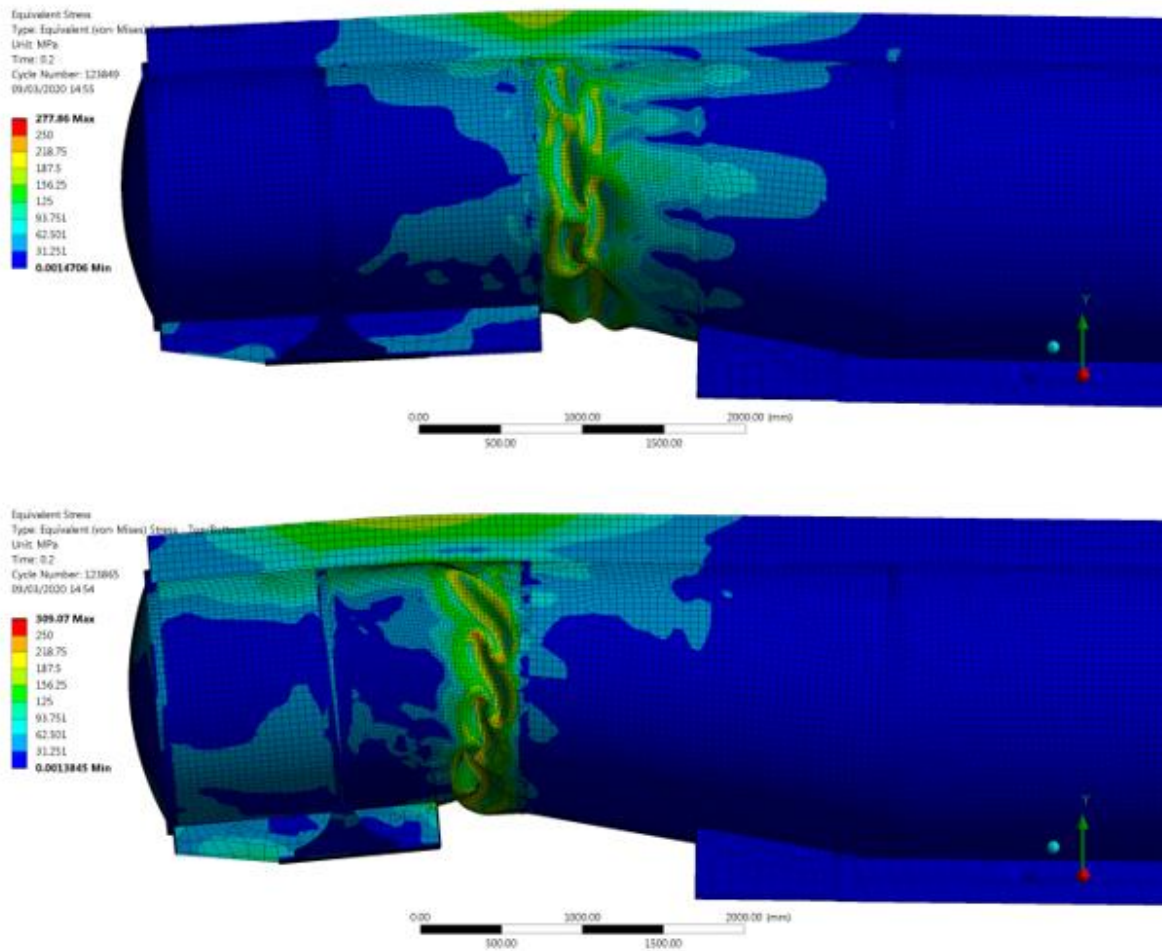


Figure F-3: Deformation of the tanker under 3 g deceleration (dynamic model)

The 'long unit' model reaches a higher maximum load before buckling than the 'short unit' model, as shown in Figure F-4. With the mass of the tanker with load being modelled as 32 tonnes, a steady 2 g acceleration would result in a force of approximately 630 kN. A steady 3 g acceleration would result in a force of 940 kN.

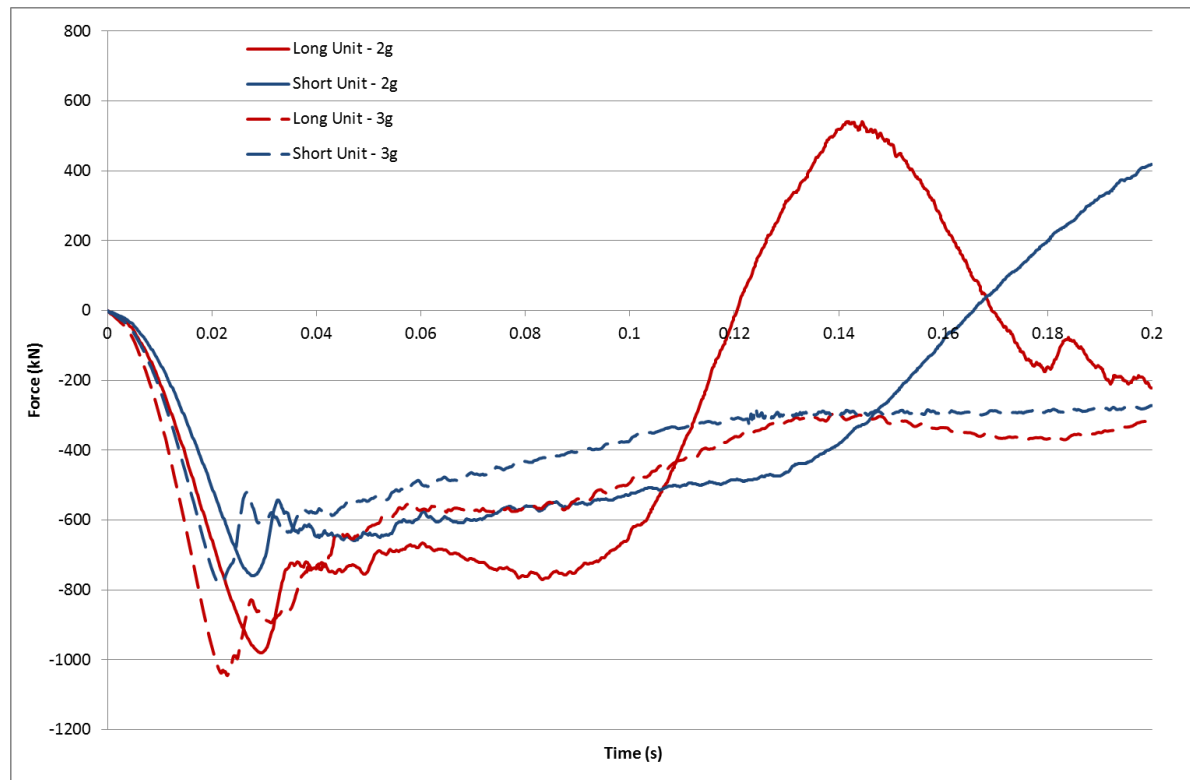


Figure F-4: Force results for the dynamic model

For the 2 *g* case, both the long unit and the short unit models reached loads in excess of the theoretical force that a 2 *g* acceleration would generate for a 32 tonne mass. In a dynamic impact situation, the maximum force depends not only on the applied acceleration, but also the stiffnesses of the bodies and loads. In these models, the flexibility of the tanker sections allowed the rear of the tanker to compress slightly before it accelerated significantly. This leads to the rear accelerating later but more quickly than the applied acceleration to the kingpin, resulting in higher loads. For the 2 *g* cases for both models, the elastic compression of the tanker sections resulted in the rear of the tanker rebounding slightly at the end of the impact, resulting in some tensile (positive) forces.

Increasing the acceleration to 3 *g* resulted in very small increases in the maximum load achieved for both tanker models. For the 3 *g* models, the forces did not become tensile within the time that the model was run.

Appendix G Outline Technical Code for rollover resilience

G.1 Introduction

G.1.1 Purpose

The purpose of this technical code is to describe ‘performance based’ test methods which can be used to demonstrate the resilience (resistance to rupture, abrasion, and penetration) of metallic²⁷ gravity discharge fuel tankers in a rollover accident scenario.

It is expected that, when the design and construction of such tanks deviates from the ADR²⁸ requirements such as those referred to in ADR regulation 6.8.2.7, this code could be used to demonstrate that the tank would maintain an ADR-equivalent performance in a rollover type accident for approval purposes.

Therefore, the acceptance requirements for the test methods, where defined, were set to demonstrate a performance level which is equivalent, as far as practical, to the performance of a tanker which currently meets the ADR requirements.

G.1.2 Current Status

Development and validation²⁹ of the topple and abrasion test methods in this code are not yet complete. Test methods are only described in outline as they require further development. Therefore, this code is currently a consultative document to be used by industry stakeholders as a guide to potential approval options at this stage.

G.1.3 Scope

This code only considers rollover resilience (resistance to rupture, abrasion and penetration) of metallic gravity-discharge petroleum fuel tankers in a rollover accident. Frontal, side and rear impacts are not considered within this scope. However, the structural parts of the tanker that make it resilient to rollover accidents will also help make it resilient to side impacts. Therefore, a separate side impact test may not be necessary.

G.1.4 Overview

A typical rollover accident may consist of three main distinct events which are assessed in three separate tests in this Code:

²⁷ Steel or aluminium

²⁸ Agreement concerning the international carriage of dangerous goods by road (ADR) 2023
<https://unece.org/transport/standards/transport/dangerous-goods/adr-2023-agreement-concerning-international-carriage>

²⁹ Validation is defined along with other terms in the Glossary in section G.4

- Topple test³⁰;
- Abrasion test; and
- Penetration test.

Accident analysis 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. This will be associated with high normal forces for a short period during the impact phase and the initial period of sliding; and then with lower normal forces for a longer period during the main sliding phase (see Knight and Dodd, 2020). Before the tanker comes to rest it may strike an object and this could penetrate the tanker shell.

However, the sequence of events of a rollover accident can vary. For example, the tanker could topple onto an object (e.g. a road barrier or street furniture) that could penetrate the tanker, which may then be followed by a period of sliding. To ensure a complete and robust assessment, each of the three distinct events involved in a rollover accident; i.e. topple and potential rupture; abrasion related to sliding; and penetration related to striking an object, are assessed using separate test methods.

Topple test methods are presented in Section G-2, and abrasion and penetration methods are in Sections G-3.1 and G-3.2, respectively.

For the topple test, four options have been provided to allow designers to choose an acceptance method based on their preference. If designers prefer the focus to be on a test-only approach (no modelling), then options 1, 3b, or 4 (joints only) can be followed. If they prefer to carry out finite element modelling, then options 2 or 3a can be followed. The key stages of the test process are shown in Figure G-1. As the tests are independent of each other, they can be carried out in any order.

³⁰ 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.

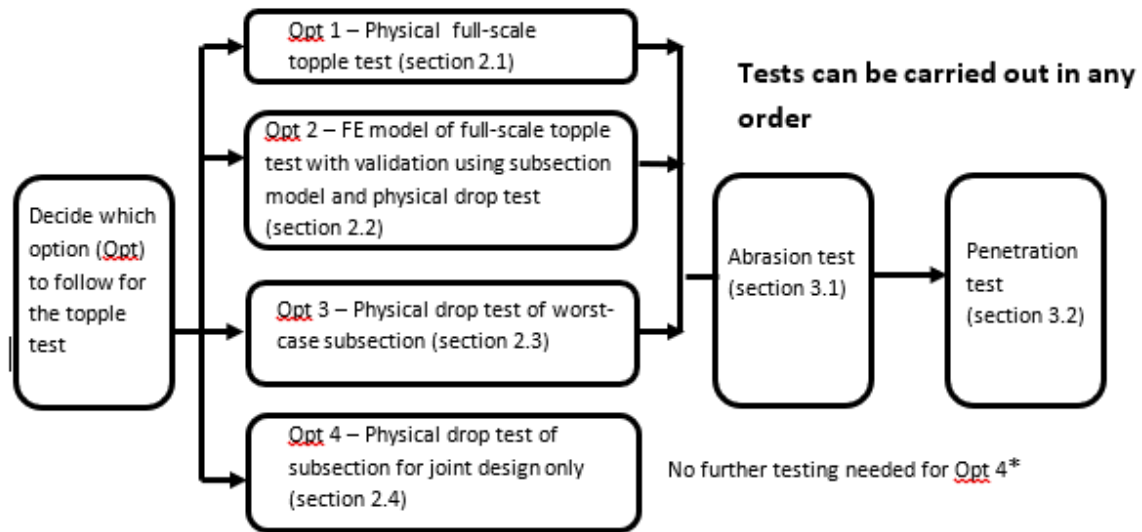


Figure G-1: Flowchart showing the test process

* No further testing is required for option 4 as this option is only being used to assess joints.

G.2 Topple Impact Performance Testing

Research has identified various methods to demonstrate a tanker's performance (i.e. its fuel containment integrity) in a 90° topple and impact (Edwards et al. 2023). These methods give rise to the four options described in Section G.1. They vary in their approach, the limitations of their applicability, and their content in terms of physical testing and modelling. Depending on exactly how the proposed alternative tank design deviates from the ADR requirements, some options may be more appropriate and/or cost effective than others for its approval via this Technical Code.

A brief description of each of the test options, which includes limitations of their applicability and the basis of acceptance requirements, is shown in Table G-1.

Table G-1: Summary of 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 in Appendix 1)	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 in Appendix 3)	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 in Appendix 5)	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 subsection 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 (Appendix 4) and will be adjusted to allow for variations in deflection along the length of the tanker, which can be determined using: <ul style="list-style-type: none"> • A finite element model of a complete tanker (Option 3a) to find the worst case compartment(s) as described above; or • a <i>standard length adjustment factor</i> as described in step 7. in Appendix 4 (Option 3b). 	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: <ul style="list-style-type: none"> • 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.

Option No.	Title	Description of Approach	Applicability Limitations	Basis of Acceptance
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).

Worked examples that show, as far as possible, how each of options 1 to 3 could be performed are provided in Appendix 1 (option 1), Appendix 2 (option 2), and Appendix 5 (option 3).

G.2.1 Option 1 – Physical full-scale topple test

The main stages in carrying out a topple test on a full-size tanker are as follows:

1. **Preparation** – the tanker should be representative of a tanker in road condition, filled to the maximum permitted level with its normal fuel load. However, a representative level of water to give equivalent impact energy to the fuel load is used for the payload in this Code (see Appendix 1.2). All auxiliary tanker components that interfere with the impact should be removed.
2. **Test** – representative of a simple 90° topple of a complete tanker onto a flat, unyielding surface and resulting impact. The test shall be designed such that it can be performed in a repeatable and reproducible manner.
3. **Acceptance** – the tanker will be accepted if it can be demonstrated to the satisfaction of the Appointed Inspection Body that visual inspection shows that there are no leaks to the external environment, and the tanker passes a leakproofness test in accordance with clause 5.8 of BS EN 12972:2018 *Tanks for transport of dangerous goods - Testing, inspection and marking of metallic tanks*.

A worked example of a full-scale topple test can be found in a report of ‘fuel tanker’ related research performed at HSE’s Science and Research Centre (Atkin C *et al.* 2015). An abridged version of this example is given in Appendix 1. This appendix also contains an explanation of why a petrol load results in a topple impact with a greater energy than for a water load of equivalent mass; namely the centre of gravity height is higher for the petrol load.

G.2.2 Option 2 – FE model of full-scale topple test with validation using subsection model physical drop test

The main stages for this option are as follows:

1. **Build and analyse a full-scale topple model** – using finite element methods, model a full-scale topple; the model shall contain a method to predict failure.

This model is used to identify the compartment most likely to fail (i.e. worst case one) and also the equivalent drop test impact conditions.

2. **Build a two-compartment subsection drop test model** - the two-compartment subsection to be modelled and tested must be agreed with the Competent Authority. The full-scale model should be adapted appropriately to create the subsection model. This subsection model will then be used to validate the full-scale topple model in step 1. The drop should be representative of the impact experienced in a full-scale topple.

3. **Test** – perform a physical drop test on a two-compartment subsection onto a flat, unyielding surface. Measure macro and local parameters to validate the subsection drop model, and by induction/inference the full-scale topple model for macro and local (failure) parameters.

4. **Acceptance** – this is based on three parts as follows:

- the two-compartment subsection does not leak after the drop test and it passes a leakproofness test.

- validation of the subsection model with the two-compartment subsection drop test; and
- verification of the full tanker model by the subsection model (verification and validation are defined in the Glossary in Section G-4).

Acceptance will be given by an Appointed Inspection Body (AIB).

Validation of macro and local model behaviour/predictions are needed. For macro behaviour validation, the minimum parameter is deflection; for local behaviour, the minimum parameter is a leak test. Strain measurements in low and high strain areas in the subsection drop test could also be used to provide further confidence in validation. For example, to give sufficient confidence to enable use of the model to facilitate approval for a limited range of tank designs. Also, it may provide useful information if the model fails to meet the validation criteria and requires further development.

An outline of the steps within each stage are as follows:

1.1) Prepare a finite element model of the complete tanker. The model should account for all parts of the shell and partitions and should consider:

- modelling the fluid structure interaction;
- the mesh type and its refinement;
- material properties;
- including supports, frame, running gear and other equipment as well as the shell and partitions where relevant;
- local strain conditions and the likelihood of failure. To achieve this, a method using the forming limit is recommended. To obtain this level of detail, an additional solid model may be needed to complement a two-dimensional shell model. Further work on forming limit methods as applied to petroleum tankers can be found in London and Smith (2016).
- the highest payload (e.g. petrol, fuel oil etc.) that the tanker has been designed to carry should be used, and the tanker velocity at impact should be as close as practical to what the impact velocity would be in a real-life tanker topple from the point of instability.

The method to determine the impact velocity shall be agreed between the Competent Authority and the tanker manufacturer/designer and shall be consistent with the method used to determine the subsection drop height. A method to consider is the calculation described in Appendix 4 to calculate an impact energy, then convert it to a single rotational speed. To do this, the second moment of area of the tanker and contents about the pivot line needs to be known. This can be obtained from the finite element model.

1.2) Analyse the model and assess the results.

2.1) From the full-scale model results, identify the compartment that is predicted to be the most likely to fail. However, a two-compartment subsection will be agreed with the Competent Authority. This two-compartment subsection will be used for the subsection

model and for the subsection drop test. This model should be able to predict macro parameters (e.g. overall deflection), and local parameters (e.g. strain) for validation purposes.

2.2) Create a subsection model of the two-compartment subsection selected. The subsection model must be created from the full tanker model. The mesh size must give similar converged results to the full-size tanker and the same material parameters must be used. However, there is no need to do detailed modelling of the forming limit in the subsection model.

2.3) Model the two-compartment subsection with a water load in a drop test. The water fill level in the model should not be less than the equivalent mass of a full fuel load (mass of water \geq fuel mass at maximum fill level). See Appendix 4 for calculating the drop height and water-fill level. The impact energy should be the same energy/partition as that specific two-compartment subsection from the full tanker model (prior to the 'topple-to-drop' adjustments described in Appendix 4).

2.4) Compare the deflection results from the subsection model with the complete tanker model (differences should be within $\pm 3\%$)

3.1) Manufacture a two-compartment subsection for testing and consider methods of lifting for the drop test (see Appendix 3).

3.2) Carry out a drop test of the same subsection with the same water-fill level used in the model in step 2.3 (see Appendix 4 for outline drop test method).

4.1) Carry out a visual check for leaks.

4.2) If no leaks are observed, carry out a leakproofness test in accordance with clause 5.8 BS EN 12972:2018 *Tanks for transport of dangerous goods – Testing, inspection and marking of metallic tanks*.

4.3) Carry out measurements of the deflection of the subsection after the drop test

4.4) Compare the measured deflections from the subsection drop test with those predicted in the subsection model.

The predicted deflections from the model should not exceed the measured deflections by 10% or be less than the measured deflections by 6% ($\pm 10\%/-6\%$). **The principal aim is not to just to have good agreement between model data and test data, it is to have a safe tanker in rollover accident scenarios which is why these asymmetric tolerance values have been selected. If test data was measuring 10% higher deflections than the model, although that might be deemed reasonable agreement, it suggests that the real tanker may have higher peak strains than the modelled tanker. As 'acceptance' is given on the basis of the model of the complete tanker, this difference is regarded as too large. However up to 10% greater values for predicted deflection is allowed as this is conservative.**

Any lids or covers that form tank openings (e.g. top openings and bottom openings) and relief valves should be fully fitted on the test subsection.

Appendix 2 gives further information on finite element modelling methods and shows a worked example of the option 2 method.

G.2.3 Option 3 - Physical drop test of worst-case subsection

This option is based on the identification of the worst-case tanker compartment i.e. the one that has the highest risk of failing. This worst-case compartment is then assessed in a two-compartment drop test. The height of the drop 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 (Appendix 4) and will be adjusted to allow for variations in energy absorption (observed in terms of deflection) along the length of the tanker (this variation in deflection was found to occur during the research to prepare this code (see Edwards et al., 2023)).

The main stages for this option are as follows:

1. **Preparation** – identify the two-compartment subsection to test. This would normally be the subsection with the highest deflection response after topple impact on a full-scale tanker where the partition joint design is the same for all partitions. This subsection can be identified by using either of two methods:
 - a) **Option 3a*** – a finite element topple model of a full-scale tanker to assess the deflection at each partition/surge plate to find the deflections that are the highest; or
 - b) **Option 3b** – an assumption that it is the rear two-compartments (Note that this has been found to be the case in supporting research for a banded type 1³¹ tanker which had the same type of joint design for all partitions and a ‘non-swept end’ type design).
2. **Preparation** – calculate the test parameters for a topple-equivalent two-compartment subsection drop test (see Appendices 4 and 5).
3. **Test** – perform a two-compartment subsection drop test from the height calculated in stage 2. above (outline test methods are shown in Appendix 3). The water fill level should be not less than the equivalent mass of a full fuel load (mass of water \geq fuel mass at maximum fill level).
4. **Acceptance** – the tanker will be accepted if it can be demonstrated to the satisfaction of the Appointed Inspection Body that visual inspection shows that there are no leaks to the external environment, and the tanker passes the leakproofness test in accordance with clause 5.8 EN 12972:2018. *Tanks for transport of dangerous goods - Testing, inspection and marking of metallic tanks.*

* In option 3a, the subsection test result does not need to validate the full-size tanker finite element model as it does in option 2. The purpose of the model in this case is just to predict the deflection along the length of the tanker (which will vary). No subsection model is required. Therefore, the modelling required in option 3a is simpler than what is required for option 2.

³¹ Banded type 1 and banded type 2 tankers are defined in the glossary.

Any lids or covers that form tank openings (e.g. top openings and bottom openings) and relief valves should be fully fitted on the test subsection.

A worked example for option 3 is shown in Appendix 5.

G.2.4 Option 4 – Physical drop test of subsection for joint design only

Option 4 is the same as Option 3b except the impact energy/partition will be fixed at 50 kJ/partition*, and it will be to demonstrate resilience of the tank for different types of circumferential joint designs between band and shell, and between partition and band/shell only (where relevant).

*** 50 kJ/partition is still to be finalised with further revisions of the code when additional tanker designs are considered.**

The test energy for the worked example for option 3 (Appendix 5), based on an 8-banded tanker, is 55 kJ/partition. It would have been 44 kJ/partition if it had been based on a 10-banded tanker. Therefore, the impact energy required to test for joint design only may be slightly less onerous than for the complete design in option 3 in some cases, and more onerous in others. However, the compartment size and strengthening element requirements of ADR are still required in Option 4. If preferred, the method shown in Appendix 4 (with worked example in Appendix 5) can be used to calculate a more accurate impact energy/partition.

Acceptance is based on there being no external leaks which are due to the failure of the joints, and the joints pass a leakproofness test in accordance with clause 5.8 EN 12972:2018. *Tanks for transport of dangerous goods - Testing, inspection and marking of metallic tanks.*

In this test, blanks can be used at tank openings if preferred. However, relief valves should be fully fitted on the test subsection.

In agreement with the Competent Authority, it may be possible to satisfy option 4 with model predictions. London and Smith (2016) have already detailed a finite element (FE) modelling method to quantitatively assess the mechanical performance and susceptibility of joints to failure. This could be used to demonstrate resilience of a tank for different types of joint designs. Pass / fail criteria for this method are based on the use of a Forming Limit Diagram (FLD) to predict the onset of tearing and rupture. The authors validated the method against experimental measurements obtained from three full-scale topple tests covering three distinct tanker designs. They then used it to assess the performance of three types of joint designs within Annex D of BS EN 13094:2015 in topple impacts for the case of end dishes. The FLD assessments predicted good performance for two of them but tearing and rupture for the design in Figure D.14(a) (clause D.2.2.5.4 in BS EN 13094:2015). On this basis, the removal of this design from BS EN 13094 was recommended and subsequently implemented in the 2020 revision of the standard. This design is referred to as 'banded type 1' in this code and in London and Smith (2016).

This illustrates that, as an alternative to a physical drop test, the FE method detailed by London and Smith could be used to demonstrate resilience of the tank for different types of joint design, provided the FE analysis requirements recommended are met and appropriate material data can be sourced, in particular for the Forming Limit Diagram (FLD). Indeed,

these provisions and the validity of the method for different joints and tanker cross-sections could be extended by comparison with results from physical subsection drop tests.

G.3 Abrasion and Penetration Testing

G.3.1 Abrasion testing

G.3.1.1 Introduction

There are currently no abrasion requirements in BS EN 13094. However, during a typical tanker rollover accident, the tanker will slide along the ground after rolling onto its side. This sliding action causes the side of the tanker to abrade as it passes along the ground. Previous research has identified that abrasion of the tank shell may be a factor causing tanks to leak in rollover accidents, though possibly only at higher speeds (Knight and Dodd, 2020). This suggests that the existing design requirements in ADR regulations result in materials with sufficient thickness to achieve a suitable level of abrasion resistance.

An abrasion test must be able to:

- abrade a sample in a reliable, repeatable way;
- control the contact force between sample and abrasion surface so that it is repeatable;
- be representative of sliding during a rollover tanker accident; and
- assess the loss of material to ensure that this is not too great when compared with the loss of material for an ADR-compliant sample.

Two potential abrasion test methods that may meet the above criteria are proposed. With further work, one of these methods could be developed into an appropriate test.

As an equivalent level of resilience to the current ADR-compliant tankers is an aim of this technical code, the abrasion test must be designed to ensure that the shells and welds of current metallic tankers that meet the requirements of ADR and BS EN 13094 will pass this test. The pass/fail criterion should be adjusted to ensure that this is the case.

There will need to be a pass/fail criteria on abrasion rate (overall loss of material). If the abrasion rate of the new sample is higher than an ADR compliant sample (e.g. it loses 50% of its thickness during the test when the ADR-compliant sample only loses 10% of its thickness) then this may indicate that the material is unsuitable as it is losing thickness in the abrasion test (and therefore strength) at a higher rate than an ADR-compliant tanker sample.

The test samples will include a jointed sample and a non-jointed (shell only) sample from the tanker body.

The two types of test method proposed for further development are: a grinding wheel method – section G.3.1.2; and a method using a tyre test machine – section G.3.1.3.

G.3.1.2 Grinding wheel method

Figure G-2 shows an outline sketch of an abrasion test with a grinding wheel.

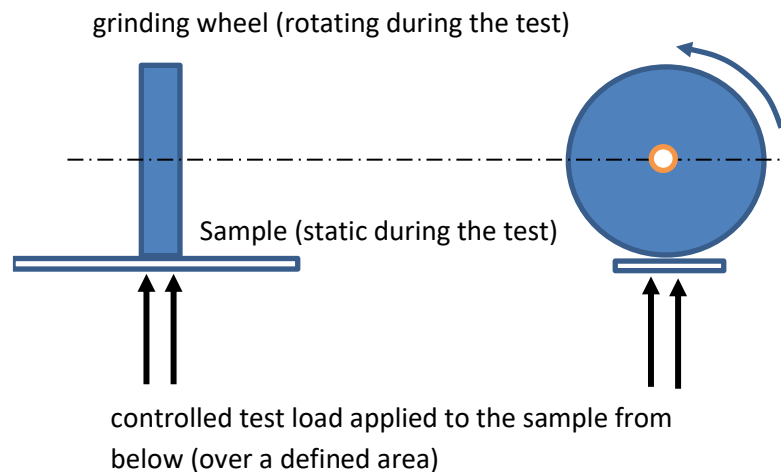


Figure G-2: Sample in a grinding wheel abrasion test

The grinding wheel rotates to abrade the sample and a controlled load is applied to the sample from below, for example, from a screw jack with a load cell attached.

A jig will need to be designed to keep the sample in approximately the same position throughout the test and resist the lateral loads.

This method will allow the force between grinding wheel and sample to be accurately controlled.

The change in thickness of the test sample due to abrasion will be compared with the change of thickness of a benchmark ADR-compliant tank material, and a pass/fail performance limit will be decided.

Potential disadvantages

The grinding wheel could become clogged with filings which alters its abrasive properties. Therefore, the grinding wheel may need to be re-dressed regularly to the same level of abrasiveness.

The type of grinding wheel to use would need to be specified for the test.

Note that dressing a grinding wheel reduces its diameter. For a reduced diameter grinding wheel the tangential speed of its circumferential surface is lower. Therefore, for a smaller diameter wheel, it would be as though the tanker was sliding along the ground at a lower speed.

Therefore, when the grinding wheel diameter has reduced by a certain amount, the wheel may need to be replaced. Increasing the rotational speed of the grinding wheel may be an option, but this may be outside the manufacturer's acceptable operational parameters for the wheel.

Work required to further develop the test method

The following tasks should be considered to further develop the method. As this is essentially a bench-top test with commonly available equipment, it is assumed that a

manufacturer or test house could acquire the equipment to carry out this test without undue cost.

1. Develop the test method to decide on the most suitable ways to hold and load the sample in a controlled way. This will require a careful risk assessment to assess all the safety issues that are relevant to working with grinding wheels.
2. Discuss the following with grinding wheel suppliers: grit sizes³², hardness values, and how often grinding wheels normally need to be re-dressed (e.g. if finer wheels need re-dressing more frequently, then a coarser grade wheel may be more suitable etc.).
3. Discuss with the grinding wheel supplier what the maximum acceptable radial load is that can be applied to the grinding wheel and what speeds the grinding wheel has been designed for.
4. Review grinding wheel options and determine a grinding wheel with a defined grit size to use.
5. Design and build a prototype test rig.
6. In the test method, note that the diameter of the grinding wheel is a significant parameter, and this will reduce as the wheel wears. Therefore, the wheel diameter may need to be used in calculating the number of rotations that should be carried out for the test. This is to ensure that all tests have the 'equivalent' distance of sliding.
7. Carry out tests with several samples from an ADR-compliant tanker to assess the reliability and repeatability of the method.
8. Review the method and advise on any improvements that need to be made. It is likely that a second series of tests on a revised design may be needed before the test method can be finalised.
9. Research work on rollover accidents referenced in Knight and Dodd (2020) has shown that, at impact, the normal forces will be high. Hence the frictional force and associated abrasion rate will also be high during the impact phase compared with the subsequent 'slide-to-rest'. An abrasion test should take this into consideration when determining suitable test loads and a time period for the test.

G.3.1.3 Abrasion test method using a tyre testing machine

This test method is based on adapting a current test machine used for tyre durability testing.

Examples of the equipment are shown in Figure G-3 (a tyre rolling resistance test) and Figure G-4 (a tyre endurance test).

³² Grit size typically runs from coarse (16 - 24 grit), medium (36 - 60 grit) and fine (80-120 grit). Superfine grits run from 150 and higher. Grinding wheels usually will be between 24 and 100 grit. A coarse grit is normally used for fast, aggressive stock removal and finer grits for less stock removal but better surface finish.

Tire Endurance Test Machine



Figure G-3: Tyre endurance test machine (photo reproduced by kind permission of www.tmsi-usa.com/51.html#page2 USA)

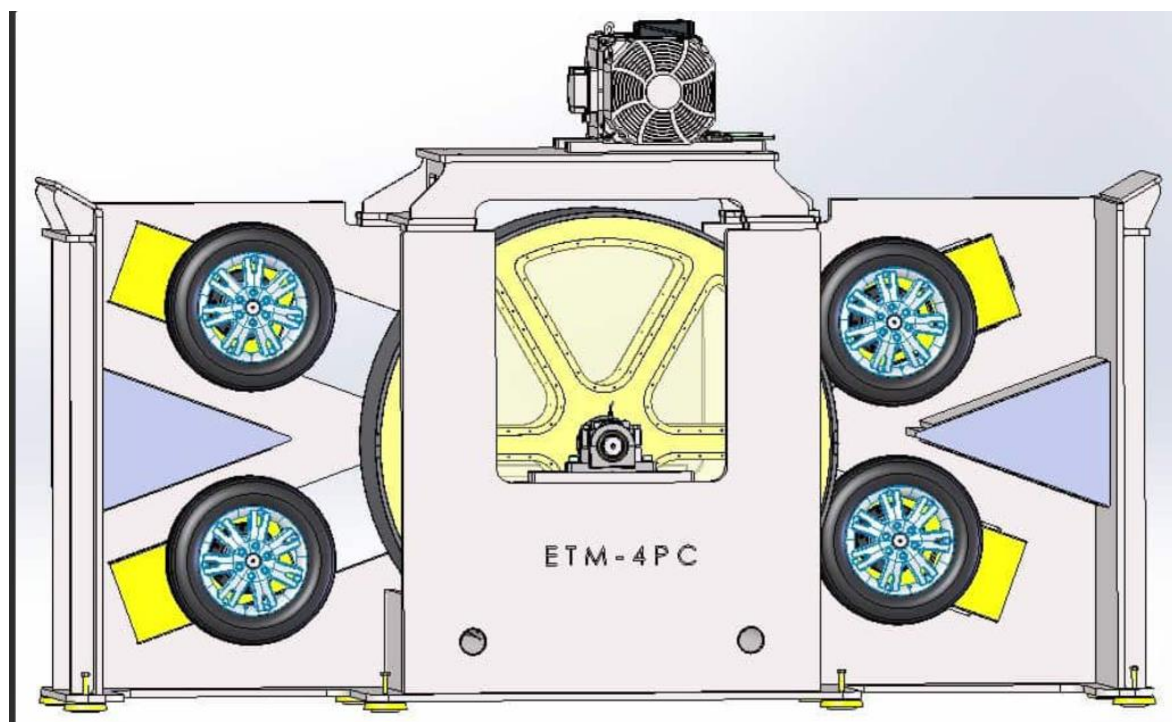


Figure G-4: Endurance and high speed test machine (photo reproduced by kind permission of www.inmess.de/en/endurance-and-high-speed-testing-machine-etm/ Bremen, Germany)

There are other types of tyre test machines that place the tyre on top of a belt rather than against a drum. A suitable abrasive surface would need to be determined for the drum or belt.

The sample would need to be attached to a jig that can apply the sample to the drum surface with a controlled static force. The drum will rotate to abrade the sample and the type of abrasive surface on the drum will need to be specified.

After the test, the sample should be subject to a tensile test and have a minimum tensile strength which will need to be determined.

Potential disadvantages

The abrasive surface may become clogged with filings which alters its abrasive properties as before. Therefore, the abrasive surface may need to be replaced regularly to the same level of abrasiveness each time. However, as the abrasive surface is that much longer than it is for the grinding wheel method, it will happen to a lesser extent. Also, the tests may need to be booked into specialist facilities (rather than using a test rig that could be developed in-house) and a specialist jig will need to be designed to hold the sample as the tyre mounts will not be immediately usable for the small test samples required in this work. Also, the owners of the facilities (mainly tyre manufacturers) may choose not to make the test facilities available for this work. There may be some concerns about the sparks that are likely to be emitted in a tank shell abrasion test. Therefore, bespoke test facilities may need to be designed from a tyre testing facility specifically for these test purposes.

We are not aware of any manufacturers of tyre endurance or tyre rolling resistance test machinery in the UK at present.

Work required to further develop the test method

1. Discuss with tyre machine manufacturers and users to assess how practical it is to adapt a machine for an abrasion test (consider both drum and belt machines).
2. Write an early draft test protocol to show the key stages required in the test.
3. Carry out visits to see machines in use and start to put ideas together for how a machine could be adapted, and what would be the most suitable type of machine.
4. Carry out an assessment of how practical this method will be (e.g. will there be sufficient providers of test services in the UK for manufacturers to go to? What effect would it have for tanker manufacturers if a provider of the test service chose to withdraw from the market? etc.) and assess what the potential cost could be to manufacturers (in time and price) if this method was to be used.
5. Research work on rollover accidents referenced in Knight and Dodd (2020) has shown that, at impact, the normal forces will be high. Hence the frictional force and associated abrasion rate will also be high during the impact phase compared with the subsequent 'slide-to-rest'. An abrasion test should take this into consideration when determining suitable test loads and a time period for the test.

At the moment the preferred method is the grinding wheel method due its simplicity and lower cost whilst not appearing to be any less accurate than the tyre test method.

G.3.2 Penetration Test

There is currently a test procedure in EN 13094 for the measurement of specific resilience of tanker shell material (i.e. its resistance to penetration). This requires a 500 mm x 500 mm sample sheet of the material used for the tanker shell.

This specific resilience test, in accordance with BS EN 13094 Annex B, shall be used as a penetration test in this test code.

EN 13094 specifies that for tank shells with a non-circular or elliptical cross-section, the shell should offer a specific resilience at least equal to that of a shell constructed in reference steel (mild steel) of a thickness of 5 mm for a shell diameter not exceeding 1.8 m; or 6 mm for a shell diameter exceeding 1.8 m - see clause 6.8.2.2 (c).

This test and the current pass/fail criteria as noted in EN 13094 are used for this test.

Work required to assist the development of the test method

The pass/fail criteria are still to be determined, so the above criteria is only a suggestion at the moment. A number of tests would need to be carried out to give a better indication of where the pass/fail criteria should be.

Also, to ensure all manufacturers are working to the same reference values of specific resilience, the test data for the ADR-compliant tanker could be carried out and be provided in the Code. Therefore, the manufacturer knows what specific resilience value they have to meet.

G.4 Glossary of terms

Acceptance	a term used to confirm that the tanker has passed the requirements of the test methods described in this technical code.
Approval	a term used to determine if the tanker is suitable for use. This will account for all the other requirements, not included in this technical code, that the tanker must also satisfy.
Banded type 1 tanker	this is the banded tanker with the joint design that was in Figure D.14a in EN13094:2015 (see below). This was removed from the 2020 version of the standard
Banded type 2 tanker	this is the banded tanker with the joint design shown in Figure D.14 c) in EN13094:2015 (see below). This has been retained in the 2020 version of the standard.

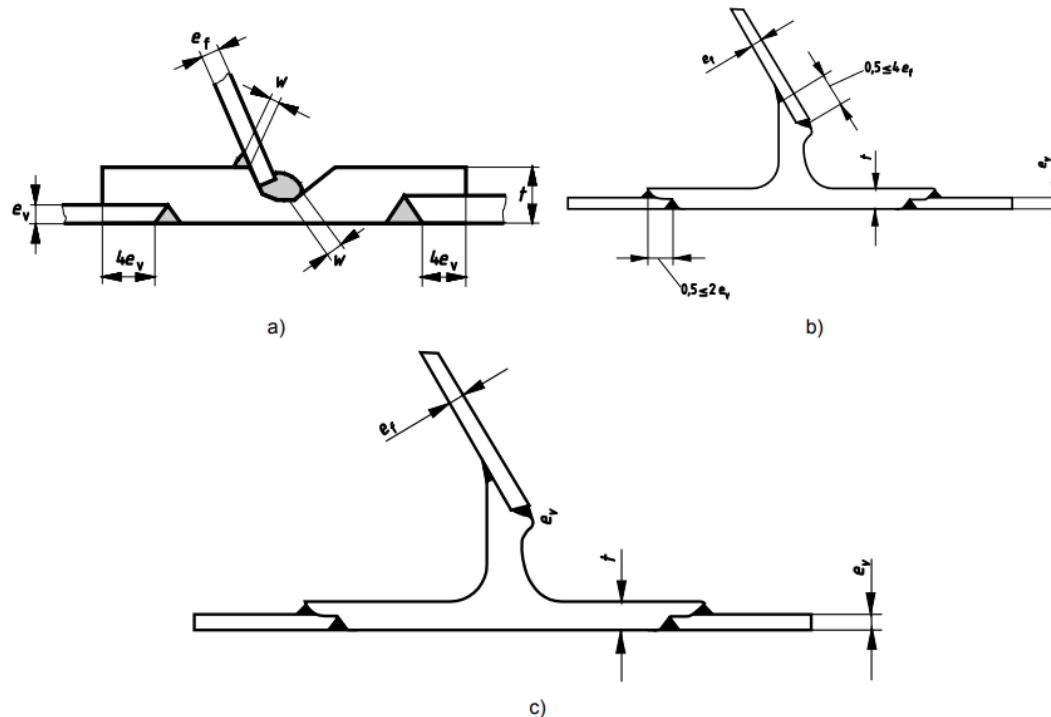


Figure D.14 — Typical partition joints

Validation a method to ensure that the model created is representative of the real-world scenario that it is trying to predict and is within its domain of applicability. A model should be built for a specific purpose or set of objectives and its validity determined for that purpose. In contrast to verification, validation can be a more interpretive process. It involves making assessments of how well the model agrees with the real-world experimental data. If agreement between the model and experimental data is varied, then assessment will need to be made to explain these differences and determine if the model is an accurate representation of the experiment. **In this code, the subsection model is validated by the subsection drop test data.**

Verification a process to check that the modelling code has been correctly set up in response to the proposed specifications. For example, have any mathematical formulae been accurately written into the code? Are the mesh size and the boundary conditions correctly input? **In this code, the full-scale model is verified if there is agreement between the full-scale model and the subsection model.**

G.5 References

Atkin C, Hobbs J, Geary L, and Webb D. (2015) *Technical assessment of petroleum road fuel tankers. Work Package 1 extension – full scale testing of Lakeland tanker AT11-1475*

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Knight I and Dodd M. (2020) *Performance test procedures for petroleum road fuel tankers – Review and analysis of accident data impact conditions and regulations*. Apollo Vehicle Safety. Report for Part A of DfT contract P4/030/0013c

London T, and Smith, S (2016) *Department for Transport Technical Assessment of Petroleum Tankers: Assessment of BS EN 13094 Lap and Partition Joint Designs* TWI Report no. 25272/1/16. Available from https://assets.publishing.service.gov.uk/government/uploads/system/uploads/attachment_data/file/940245/Technical-assessment-of-BS-EN-13094.pdf (accessed 29/05/2023)

Theseus (1995) *Tank vehicles with maximum attainable safety through experimental accident simulation* – final summary report – sponsored by the Federal Ministry of Education, Science, Research and Technology BMBF (Cologne, March 1995)

G.6 Appendices

G.6.1 Appendix 1 – Option 1: example of physical full-scale topple test

Appendix 1.1 shows an example of a previous topple test of a full-size tanker as required in option 1.

Appendix 1.2 explains why a petrol load has greater impact energy than a water load of the same mass in a topple test. This is relevant when deciding the water level for the tanker in the topple test.

G.6.1.1 Appendix 1.1 Topple test of a full-size tanker

Figure A1 1 shows a tanker in position in preparation for a topple test.

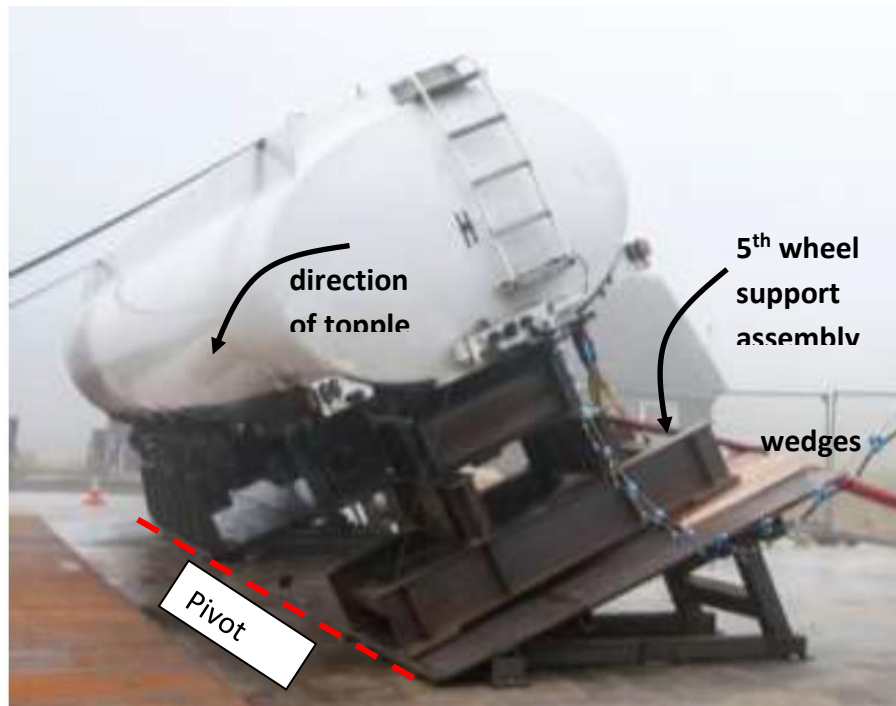


Figure A1.1 Tanker being prepared for a topple test (photo courtesy of HSE Science Division)



Figure A1.2 Tanker being prepared – showing the square flanges in place of wheels on the topple side (photo courtesy of HSE Science Division)

Preparation

1. The tanker should be toppled onto the opposite side to the one which the filling ports are located on.
2. Carry out a pre-test leakproofness test in accordance with clause 5.8 BS EN 12972:2018 *Tanks for transport of dangerous goods – Testing, inspection and marking of metallic tanks* to ensure that the tanker is not leaking. This is optional, but it does confirm that the tanker was not leaking immediately prior to the test.

3. If the tanker has been in operation, ensure the compartments are ventilated, cleaned and checked appropriately.
4. All auxiliary components that could interfere with the impact shall be removed. This may include: brackets; mudguards; flexible hoses; the box containing firefighting equipment etc.
5. If the tanker has been in operation, consider removing the pipework to and from the pump and blocking suction and discharge ports at the pump to prevent any residual fuel spills during the test.
6. Replace the wheels on the 'pivot-line' side (Figure A1.2) with square flanges so the vehicle track (overall width) and axle heights remain the same.
7. Block the suspension at each side on each axle³³.
8. Replace the tractor unit with a 5th wheel support assembly of the same width as the tractor unit (an example made of welded carbon steel beams is shown in Figure A1.1 and A1.2).
9. Install two steel ramps at an angle (for example 25°) to reduce the winching force required. Wedges can be placed under the wheels and the 5th wheel assembly on the upper sides on the ramp (2° wedges are shown in Figure A1.1) to reduce the winching force further if required. **Safety note: these angles are nominal and based on the angles required for previous topple tests of a semi-trailer tanker. This should be assessed for your unique design of tanker before deciding on what angles to use. Too steep an angle could mean the tanker is too close to the point of toppling when placed on the ramp.**
10. Weld a steel lip along the bottom of the ramps at the pivot line to prevent the tanker from sliding down the ramp.
11. The impact pad should be an *unyielding target*³⁴. In this case 'unyielding' means a hard surface where the deflection on the pad at impact must be less than 5% of the deflections on the tanker. A suitable impact pad would be a carbon steel plate 20 mm thick the full length of the tanker, which is firmly anchored to a 200 mm thick fully-supported reinforced concrete ground slab mounted on firm soil or bedrock (the steel plate can be seen more clearly in Figure A1.3).
12. Lift the empty tanker (with 5th wheel assembly fitted) onto the ramps.

³³ 6. and 7. are to ensure there is no secondary movement of the tanker during the topple.

³⁴ The target for drop tests is specified as an essentially unyielding surface. This unyielding surface is a hard surface intended to cause damage to the tanker which would be equivalent to, or greater than, that anticipated for impacts on to actual surfaces or structures which might occur during transportation. The specified target also provides a method for ensuring that analyses and tests can be compared and, if necessary, accurately repeated. So-called 'real' targets, such as soil, soft rock, tarmac and some concrete structures, are less rigid and may cause less damage to a tanker for a given impact velocity.

13. Whilst the tanker is still supported by the crane, restrain the tanker on the upper side to ensure it does not topple prematurely during test preparation.
14. Fill the tanker with water to an 80% fill level³⁵. Note: this will be over-filling by mass. However, this is necessary due the reasons explained in Appendix 1.2.
15. Carry out a visible check for leaks.



Figure A1.3 Tanker being prepared for a topple test (photo courtesy of HSE Science Division)



Figure A1.4 Tanker being prepared for a topple test (showing winching lines) (photo courtesy of HSE Science Division)

³⁵ The fill level for water to be approximately the same energy as petrol is 80% (when compared to a 'standard' petrol fill of 95%)

Test

16. Place two wide, long-length webbing slings over the top of the tanker and attach them to the tanker on the non-impact side – these slings will act as winching lines (see Figure A1.3 and A1.4).
17. At the end of the slings, connect them to further slings, as necessary, and then to two winches on the topple side of the tanker (Figure A1.4). **Ensure the winches are a safe distance from the tanker in case it rolls or slides further after impact. Hand winches with a suitable safe working load (SWL) are adequate.**
18. Carry out the topple test. Ensure the load is applied evenly to each winch – in-line load cells can be used on each line and the winching adjusted to balance the load on each line if needed. The load reduces during the winching and approaches zero as the tanker is at the point of toppling. When the tanker impacts the ground, water will be released from the relief valves on top of the tanker (Figure A1.5). However, this does not constitute a failure unless the water continues to leak through the valves after the tanker has come to a rest (Figure A1.6). Figure A1.7 shows the ‘flat’ on the tanker when it has been lifted back into position.



Figure A1.5 Tanker topple test – at the point of impact (photo courtesy of HSE Science Division)



Figure A1.6 Tanker on its side immediately after a test (photo courtesy of HSE Science Division)



Figure A1.7 Tanker up-righted after a test showing the flats on the impacted side (photo courtesy of HSE Science Division)

Assessment

19. Whilst the tanker is resting on the ground, carry out a visual check for leaks on all parts of the shell, partition joints, and front and rear bulkheads.
20. Lift the tanker to an upright position.
21. Carry out further visual check for leaks.
22. If no leaks are observed, drain the tanker and carry out a leakproofness test on all compartments in accordance with clause 5.8 BS EN 12972:2018.

G.6.1.2 Appendix 1.2 Explanation of why a petrol load has greater energy than a water load of equivalent mass for topple

In option 1, water is required as the liquid medium due to safety and environmental risks that would occur if petrol or other fuels were used. However, water of an equivalent mass to petrol will have less volume and a lower centre of gravity (CoG) than petrol. Previous modelling has shown that this gives a lower deflection from the impact than a fuel load. This is because the distance between the pivot point (or pivot line) and the CoG is greater for a petrol load than it is for a water load. This is explained in the HSE Science Division paper (Hobbs et al. 2022).

A section of the paper is shown below which demonstrates this visually.

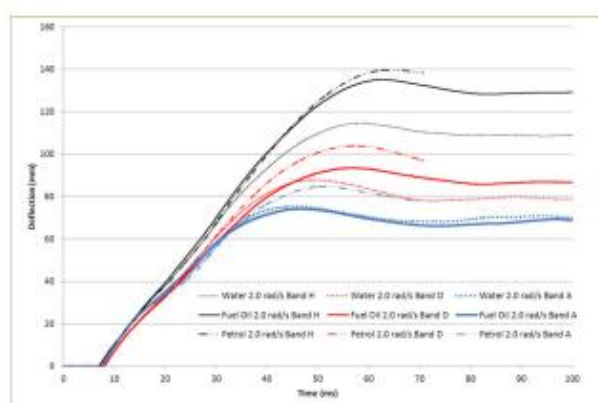


Figure 15. Effect of tanker load (fuel oil/water/petrol) on deflections of the bands.

empty. With the previous models, only the end bulkheads were modelled as coupling surfaces; all other bulkheads and the baffle

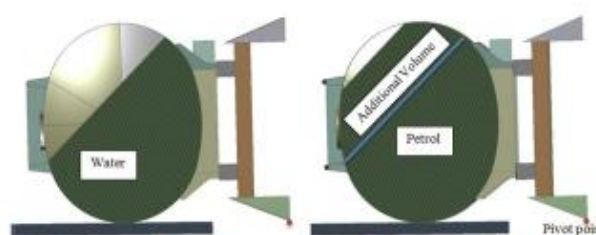


Figure 16. Schematic of water and petrol fills illustrating the additional mass further from the pivot point.

Table 4. Maximum plastic strains occurring in the bulkheads.

Liquid	Maximum equivalent plastic strain (through- thickness average)	Maximum equivalent plastic strain (near inner surface)
Water	20 %	32 %
Fuel oil	25 %	37 %
Petrol	34 %	41 %

Information on petrol topple having greater energy than a water topple (for equivalent mass)

Therefore, the water level needs to be set to give equivalent impact energies as would occur in a petrol topple. To achieve this the mass of water and associated fill level must be slightly higher than it would be for equivalent mass to the petrol.

For a banded type 1 tanker, an equivalent energy to a petrol fill of 95% (by volume) was given by a water fill level of 80% which had an equivalent mass of 113% (i.e. 13% heavier than a 95% fill with petrol). These percentage values are likely to be the same for all tankers of a similar geometry.

G.6.2 Appendix 2 – Option 2: FE modelling methods and example

This Appendix is divided into two parts: the first part contains information related to modelling methods recommended for option 2; and the second part contains an example of how to carry out a complete tanker model, subsection model, and drop test as required for option 2. Guidance on performing a subsection drop test is contained in Appendix 3. Calculation of the impact energy and drop height for the subsection model and drop test is shown in Appendix 4, with a worked example in Appendix 5.

G.6.2.1 Appendix 2.1 Finite element modelling methods

For option 2, from a material point of view, the finite element model should predict the onset of failure (thus a failure criteria) but does not need to predict failure propagation. The software should support constitutive behaviours for materials that at a minimum allow for elastic-plastic behaviour based on the von Mises yield criterion, associative flow rule, and isotropic hardening.

The finite element modelling software package should³⁶:

- a) *be capable of analysing thin shells;*
- b) *be capable of calculating the membrane and bending stresses and principal strains;*
- c) *be capable of calculating shell deflections;*
- d) *automatically generate warnings where elemental shape, aspect ratio or other parameters exceed limits set by the software supplier;*
- e) *be capable of displaying the model in such a way that unintended disconnections between elements are clearly visible;*
- f) *be capable of displaying duplicated elements;*
- g) *have a modelling system capable of modelling fluid structure interactions.*

Appropriate material failure criteria and material property data for the modelling are essential.

For option 2, due to the sensitivity of explicit models to the minimum element size, it may be difficult to achieve a mesh independent solution for the strain results using a shell model and fluid structure interaction. In this case, creating a more detailed model of the joints using solid elements may be necessary. If this approach is taken, the pressure forces due to the liquid should be approximated using pressure forces as appropriate.

Further information on finite element modelling of tankers can be found in London and Smith (2016).

G.6.2.2 Appendix 2.2 Example (FE model of a full-scale topple test)

Determine parameters for the topple model

The first step is to estimate the impact energy for a full tanker impacting the ground from a topple position (i.e. at the point of instability). This can be assumed to be the same as the change in potential energy between the topple position and the impact position.

In this example, the centre of gravity (CoG) of the liquid is taken from the finite element model as this information will be available. This results in a slightly lower assumed impact

³⁶ This is based on Annex 3.3 of BS EN 13094: 2020 *Tanks for the transport of dangerous goods — Metallic gravity-discharge tanks — Design and construction* but with some modifications.

energy than calculated when assuming the CoG of the liquid is at the geometric centre of the compartments (see option 3 worked example in Appendix 5).

For the tanker in a horizontal upright position, the distances of the CoG for the tanker and the liquid load from the outer edges of the wheels are shown in Table A2.1. The outer edges of the wheels are on the pivot line as shown in Figure A1.1.

Table A2.1: Example positions of centre of gravity (CoG) for tanker and load as distance from the pivot line, which corresponds to the outer edge of the wheels (the tanker is horizontal and in the upright position)

	Horizontal distance (mm)	Vertical distance (mm)	Mass (kg)
Empty Tanker (including supports)	1275	1182	5 500
Liquid Load	1275	2068	31 400
Combined (empty tanker and liquid)	1275	1934	36 900

The height of the CoG when the tanker is at the tipping point (h_t in step 4 of Appendix 4) is the vertical distance of the CoG above the pivot line. This is obtained by carrying out a vector addition:

$$h_t = \sqrt{(1934 \text{ mm})^2 + (1275 \text{ mm})^2}$$

$$h_t = 2317 \text{ mm}$$

Potential Energy (PE) is calculated from the mass (m) x acceleration due to gravity (g) x height (h). The change in potential energy (ΔPE , where the mass remains constant as in this case) is calculated from the change in the height of the CoG between the tipping point and the impact point (h_i in step 4 of Appendix 4):

$$\Delta PE = (h_t - h_i) \times g \times m$$

$$\Delta PE = (2.317 \text{ m} - 1.275 \text{ m}) \times 9.81 \text{ m}\cdot\text{s}^{-2} \times 36900 \text{ kg}$$

$$\Delta PE = \mathbf{377 \text{ kJ}}$$

As mentioned above, it is assumed that the change in potential energy ΔPE , is the same as the kinetic energy at impact, KE_{Topple} .

Calculate the rotational velocity to achieve the desired kinetic energy.

If the moment of inertia about the pivot line is known, a rotational velocity can be calculated:

$$\omega = \sqrt{\frac{2 \cdot KE_{\text{Topple}}}{I}}$$

The moment of inertia, I , for the example case is $2.26 \times 10^5 \text{ kg/m}^2$. Therefore, the rotational velocity at the moment of impact is:

$$\omega = \sqrt{\frac{2 \times 377 \text{ kJ}}{2.26 \times 10^5 \text{ kg/m}^2}}$$

$$\omega = 1.83 \text{ rad/s}$$

Model the complete tanker

The complete tanker finite element model is set up to represent the topple. This is shown in Figure A2.1 with a fill to 95% of petrol. The model consists of shell elements to represent the tanker and supports, and a rigid surface to represent the ground. The liquid was represented using Euler elements, with a coupling surface defined as the tanker shell and the end dishes. The internal partitions were not defined as coupling surfaces as the pressures acting on each side would be similar, and additional coupling surfaces would have significantly increased the solution times.

The model was run from just before the moment of impact, with an initial rotational velocity of 1.83 rad/s as calculated above.

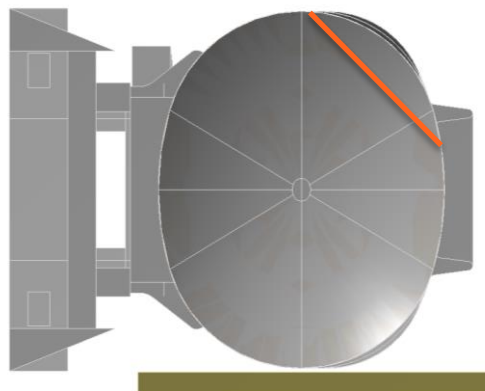


Figure A2.1: Model of complete tanker just before impact

Assess the complete tanker topple results

Deflection

The maximum deflection at each partition/surge plate and the end bulkheads is recorded. The deflection is the reduction in the width of the tanker (i.e. the vertical line through the centre of the tanker in Figure A2.1).

Forming Limit

To assess material failure, the strains in the tanker need to be checked to make sure that they do not exceed the forming limit. As mentioned earlier it may be necessary to use a solid element model to get the local strain data required for this assessment.

The solid element model consists of a quarter of a partition with the associated shell, as shown in Figure A2.2, with planes of symmetry defined on the cut surfaces (the symmetry planes). The load is applied by moving the ground up into the partition. As the solid element model does not include the liquid, the force applied by the liquid need to be applied as a hydrostatic pressure force on the internal surfaces. The magnitude of the force should be obtained from the dynamic shell element model.

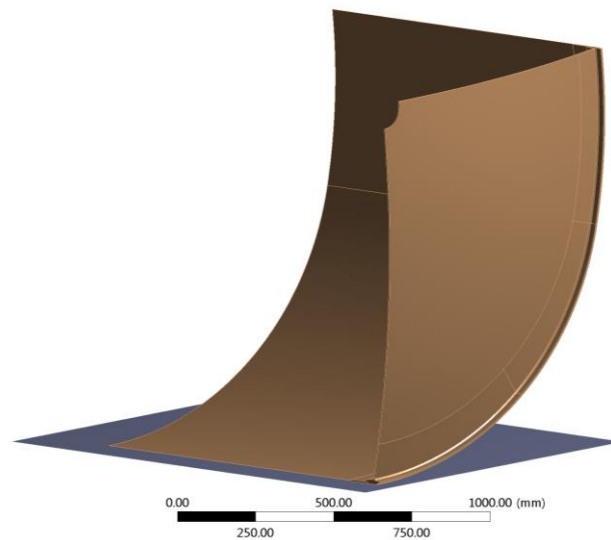


Figure A2.2: Quarter model of a partition using solid elements to obtain local strain information

The compression of the partition should match the deflection recorded in the dynamic shell-element model.

The results for two different designs of banded tanker are shown in Figure A2.3. The forming limit used in this example was a lower bound to published data for EN AW-5182 aluminium alloy³⁷:

$$\varepsilon_{maj} = \begin{cases} -0.54\varepsilon_{min} + 0.20 & \text{if } \varepsilon_{min} < 0 \\ 2.6906\varepsilon_{min}^3 - 2.6198\varepsilon_{min}^2 + 1.1519\varepsilon_{min} & \text{if } \varepsilon_{min} \geq 0 \end{cases}$$

³⁷ For more details, see 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 at: [Technical assessment of BS EN 13094: petroleum tankers \(publishing.service.gov.uk\)](https://www.publishing.service.gov.uk/government/uploads/system/uploads/attachment_data/file/444444/Technical_assessment_of_BS_EN_13094_petroleum_tankers.pdf)

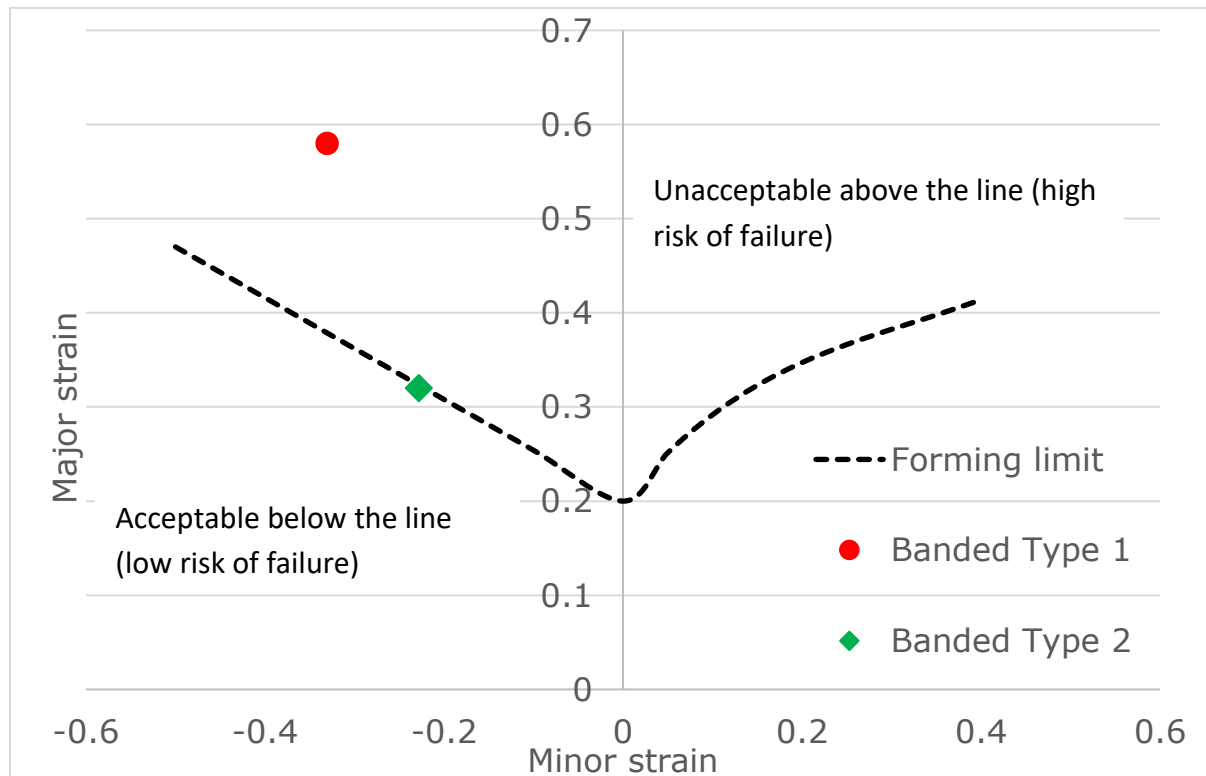


Figure A2.3 Forming limit diagram with results for two types of tanker

For the banded type 1 tanker, the strains in the dished section of the partition near to the weld of the extruded band, exceed the forming limit. In this case, there would be no point in proceeding with the physical tests as there is a very high risk of failure. If the physical drop test confirmed the model results, the tanker would fail the test. And if the physical drop test results varied from the model results, the model would not be validated.

The banded type 2 tanker results lie just below the forming limit. Therefore, for this tanker, the complete tanker finite element model could now be validated using a subsection drop test with associated model as the full-size model results suggest this tanker is likely to pass the subsection drop test and not leak.

The two compartments to be chosen for the test will be agreed with the Competent Authority. In this example, the highest deflection occurred at the rear end of the tanker, so the rear two compartments were chosen for the subsection drop test.

Subsection drop test

The kinetic energy for the topple needs to be adjusted to account for the difference between drop and topple, variation along the length, and the number of compartments. An example of the procedure for this can be found in 'Option 3(a) Finite element analysis' in Appendix 5.4 to obtain the length adjustment factor (f_{LA}). This will require the deflection results obtained at each partition/surge plate from the complete tanker model. The drop height should then be calculated as shown in Appendix 5.5.

G.6.3 Appendix 3 – Options 2, 3 & 4: Subsection drop test method(s)

This appendix shows an example of a method used to carry out a subsection drop test from past research work and provides an outline of potential test methods.

For a drop test, lifting points (lugs) may need to be added to the upper side of the subsection to raise it into position. However, this could possibly be avoided by lifting the tanker from lifting slings that go around the underside of the tanker, this will make the drop test easier, although the straps may interfere with the drop (cushion the impact a little). However, if the strap is a thinner webbing strap, the cushioning effect will be minimal. This is what was used to carry out drop tests at BAM in the Theseus project³⁸ as shown in Figure A3.1 (Figure 6.27 on p212 of the report). Therefore, any future assessment of the test could consider the advantages and disadvantages of this method as it will be simpler than installing lugs (**Note: lorry straps with quick-release ratchets should be avoided for safety reasons, unless they are approved for lifting operations**).

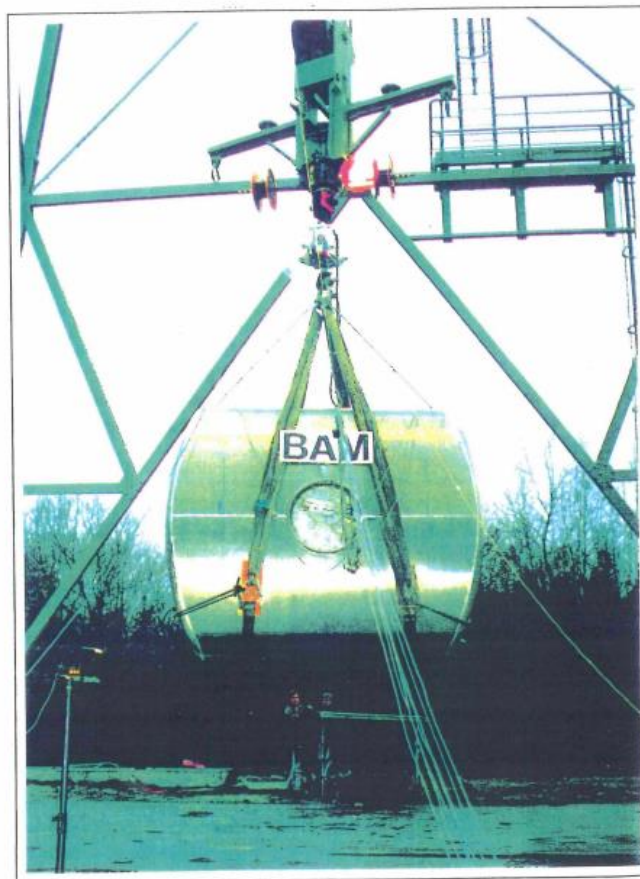


Figure A3.1: Subsection drop test carried out at BAM, Germany

³⁸ Theseus – *Tank vehicles with maximum attainable safety through experimental accident simulation* – final summary report – sponsored by the Federal Ministry of Education, Science, Research and Technology BMBF (Cologne, March 1995)

Two methods of dropping the subsection using lugs on the upper side are shown in outline in Figures A3.2 and A3.3.

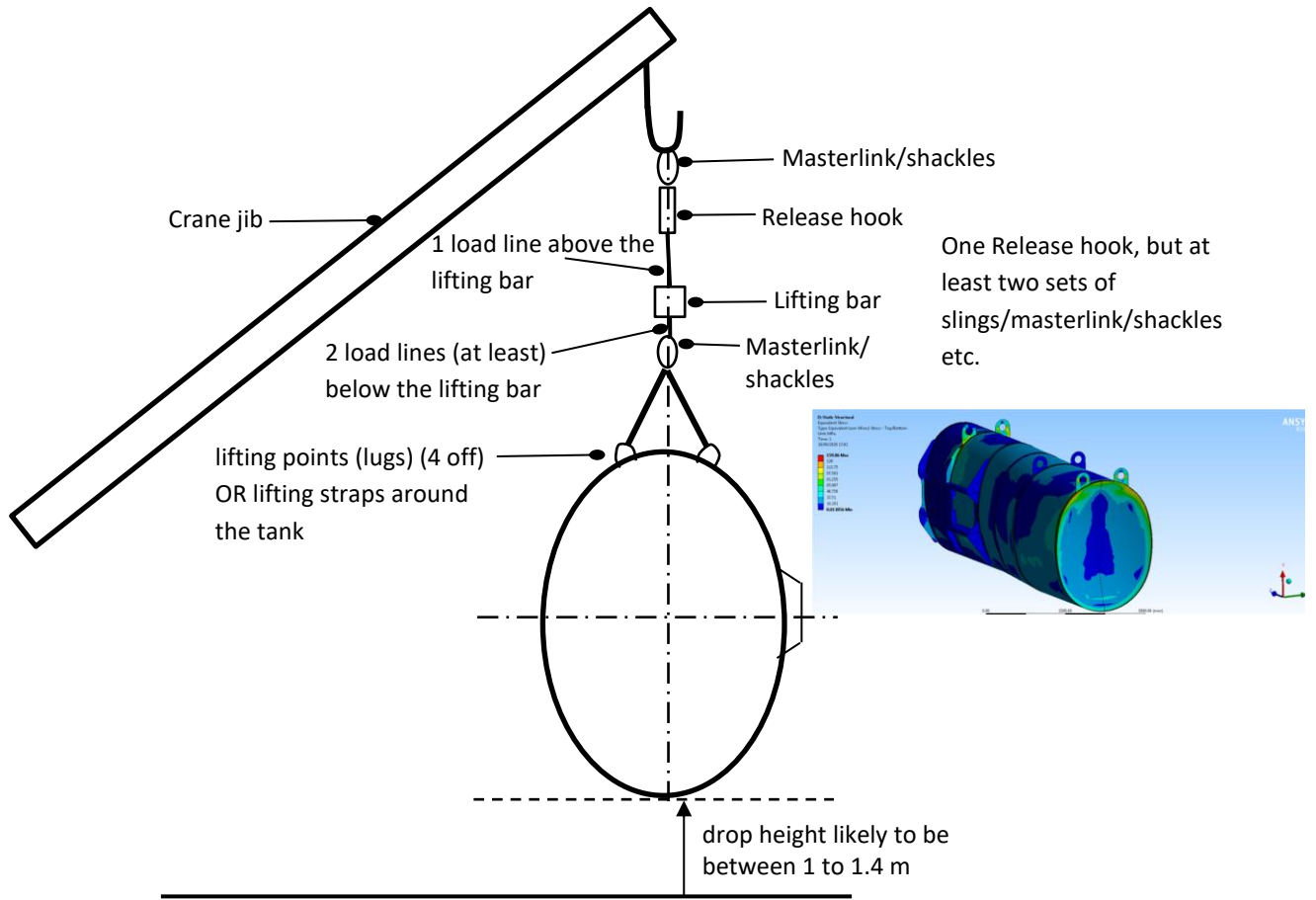


Figure A3.2: Drop from a crane (lifting points on the upper side) (Method 1)

As the subsection mass is likely to be around 10 to 20 tonnes, it may not be practical to drop from a crane jib. Therefore, a fixed frame could be used as shown in Figure A3.3.

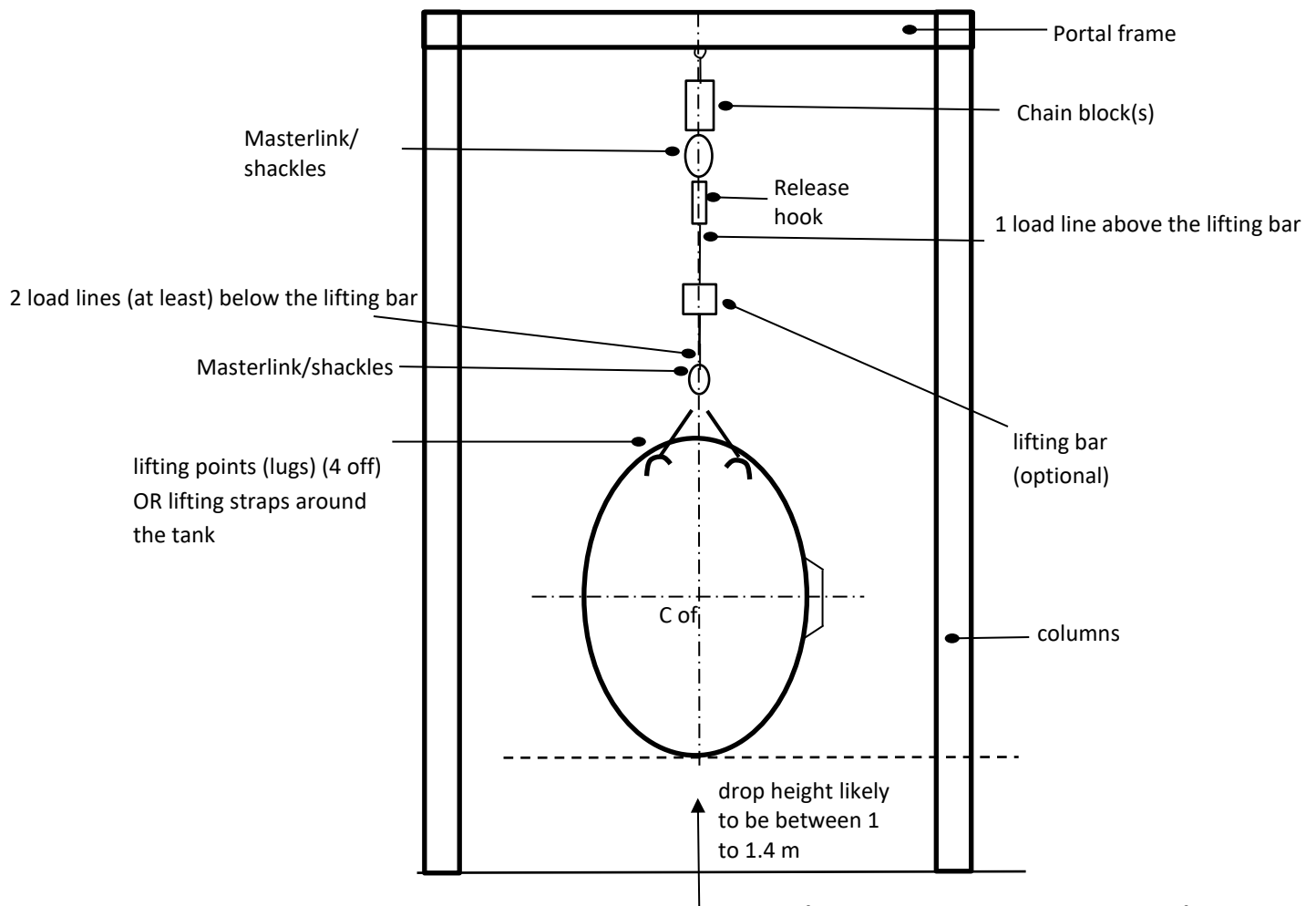


Figure A3.3: Drop from a frame with one release point (lifting points on the upper side) (Method 2)

The lifting bar in Figures A3.2 and A3.3 may not be necessary. It was not used in the tests at BAM (Figure A3.1).

G.6.4 *Appendix 4 – Options 2, 3 & 4: Calculation of the impact energy and drop height for subsection drop test/model*

This appendix provides a detailed description of a method to calculate the impact energy and drop height for a subsection drop test or FE model which is representative of the loading conditions experienced in a topple impact.

1 Calculate the height of the centre of gravity (CoG) of the tanker when upright and horizontal³⁹.

The CoG of the liquid mass can be assumed to be at the centre of the compartment (i.e. assuming a completely full compartment). As the ullage is small, the difference in results will be small (around 5%) and the calculations will be much easier, especially when considering the tipping point.

The mass and CoG of gravity of the running gear can be ignored in the calculation. This will simplify the calculation (especially if it is not known at the time of performance testing).⁴⁰

2 Calculate the height of the CoG when the tanker is at the point of instability

This is the 'tipping point' where the CoG is directly over the outside of the wheels, h_t (see Figure A4.1a).

NOTE A petrol topple is being considered here so the fill level is 95%. If we use the assumption in step 1 (full compartment), the CoG will be very slightly higher than it would be in reality because of the small movement of the petrol due to the change of position of the ullage at the topple point when compared with the tanker in its normal horizontal position. This will give a very slightly higher impact energy. The mass would still be the same.

3 Assume the height of the CoG at the point of impact (h_i) is likely to be half the width of the tanker

The tanker is assumed to be symmetrical about the longitudinal axis h_t (see Figure A4.1b).

4 Calculate the change in potential energy

$$\Delta PE = m \times g \times (h_t - h_i)$$

where m is the total mass of the tanker including the fuel load filled to its maximum fill level

g is acceleration due to gravity (9.81 ms^{-2})

³⁹ As designs could vary, a single set impact energy may not be appropriate. The easiest way to calculate the impact energy is to assume that it is equal to the reduction in potential energy during the drop as mentioned in Appendix 2. By finding the location of the CoG when the tanker is level, the angle of instability (when the CoG is over the pivot line) can be estimated.

⁴⁰ As the running gear is low with the tanker in the upright orientation, it will be higher when in the topple position. Therefore, ignoring the mass of the running gear results in a small increase in the change in potential energy, and therefore a higher impact energy in step 5. The mass of the running gear is included in the worked example in Appendix 5 as it was known when the model was being prepared.

h_t is the height of the CoG above ground level at topple
 h_i is the height of the CoG above ground level at impact.
The subscripts t and i refer to 'tipping' (height of CoG at the tipping point) and 'impact' (the height of the CoG at impact).

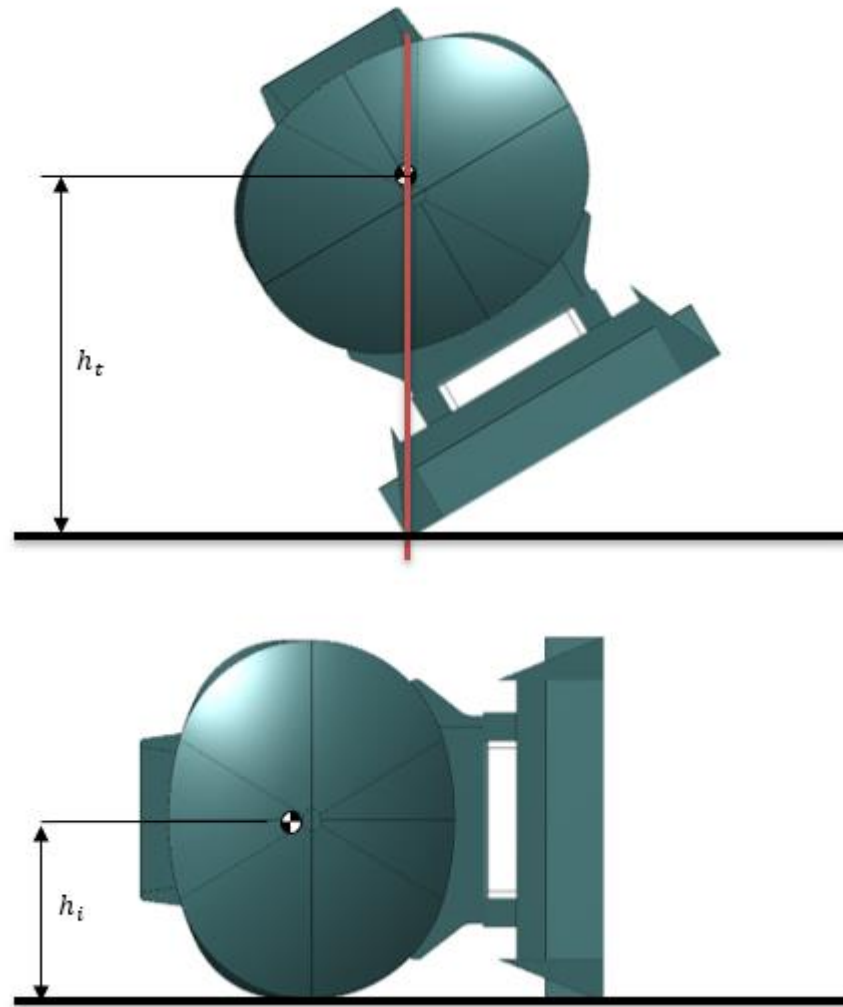


Figure A4.1a (above): Tanker at the point of impact

Figure A4.1b (below): Tanker on its side

5 Make adjustment to allow for difference between a topple and a drop test.

As the topple will involve rotational movement, not all the liquid velocity is in the vertical direction at the point of impact. As the subsection drop test will be a vertical drop, with far more liquid velocity in the vertical direction, much more of the kinetic energy will go into deforming the tanker.

Modelling has shown that a reduction of 27% of the potential energy calculated from a finite element model of a topple test gives equivalent tanker deformation in a model of a vertical drop test.

$$KE_{drop} = 0.73 \cdot \Delta PE$$

(i.e. the kinetic energy for a vertical drop test of a complete tanker that would give similar deflections to that of an equivalent topple.)

6 Calculate the impact energy per partition (KE_{part})

The deflection of the tanker depends on the number of partitions and surge plates within the complete tanker. This should include all ends and partitions. Surge plates should be included where they have a similar form to the partitions, and any holes account for less than 30% of the overall surface area.

Further assessment will be required and agreement with the Competent Authority for what strength addition can be considered for surge plates with a hole which is greater than 30% of the overall cross sectional area.

$$KE_{part} = KE_{drop} / n_{tot}$$

where n_{tot} is the total number of partitions/surge plates.

7 Adjust the impact energy per partition to account for variations in the deflection along the length of the tanker.

From previous research work, when modelling the topple (and drop) of a complete tanker, the deflection varies along the length, with the highest levels of deflection typically occurring at the rear. Therefore, the impact energy for a subsection drop test needs to be adjusted so that the deflections match the most heavily impacted compartments in a full-size tanker.

This is referred to in this Code as the *standard length-adjustment factor*, which is 1.25*, and is used in options 3b and 4 (see worked example in Appendix 5).

*** 1.25 is a conservative value based on results from the banded type 1 tanker (model and topple tests), and the banded type 2 tanker (topple test only) where the factor varied from 1.15 to 1.24.**

To achieve this higher deflection, the impact energy must be increased by a factor which is the square of the standard length-adjustment factor due to the relationship between deflection and impact energy (this is explained in Appendix 5.5). This gives an 'adjusted' value of kinetic energy (KE_{adj}).

$$KE_{adj} = 1.56 \times KE_{part}$$

However, in option 3a the *standard-length adjustment factor* is not used. A complete tanker topple is modelled using finite element to obtain a prediction of the variation in deflection along the length for the specific tanker. To do this, create a model according to guidance for Option 2, but without the requirements for strain analysis (solid modelling). Then calculate the deflection for each partition/end/surge plate to obtain the average (mean) deflection. The deflection is the reduction in the width of the tanker (i.e. the vertical line through the centre of the tanker in Figure A2.1).

This is then used to obtain the *length-adjustment factor* (f_{LA}) which is specific to the design of tanker rather than the standardised value used for option 3b.

The length-adjustment factor, f_{LA} , is the proportion that the highest-deforming partition/end/surge plate is above the average:

$$f_{LA} = \frac{\text{Maximum deflection}}{\text{Average deflection}}$$

The increase in energy is proportion to the square of the deflection increase:

8 Calculate the total energy for the subsection drop test

The water fill level should not be less than the equivalent mass of a full fuel load (mass of water \geq fuel mass at maximum fill level).

The total energy for the drop test will depend upon the number of partitions in the subsection being dropped (n_{sub}). This will include ends, partitions and any surge plates as before. n_{sub} will typically be three for the two-compartment subsection unless surge plates are in the rear two compartments.

$$KE_{test} = KE_{adj} \times n_{sub}$$

9 Multiply by a safety factor (S)

The need for a safety factor is to accommodate possible uncertainties as follows:

option 3a – uncertainties due to the modelling (i.e. the model has under-predicted deflection at the partitions); or

option 3b – uncertainties in the standard length-adjustment factor of 1.25 due to the limited sample of data it has been obtained from, and only from two tanker

designs⁴¹. Further data from other tanker designs could potentially show that the highest partition deflections are higher than the average partition deflection by a factor greater than 1.25. However, as 1.25 is an upper bound for the banded type 1 and type 2 designs it is very unlikely that other tanker designs would have a factor much higher than this (a few percentage points at most is what would be expected).

Therefore, as the factor 1.25 is already an upper bound based on the current data, this will influence the choice of safety factor. At present a safety factor of 1.1 is suggested. This is a typical value for partial safety factors used in structural reliability work where the uncertainties are not thought to be too high. This will be multiplied by the energy value from the previous step. This can be reviewed in future revisions of the code.

10 Calculate the drop height (H)

As the kinetic energy for the test (KE_{test}) is the same as the potential energy for the test (PE_{test}) when the subsection is dropped from the test drop height (H), the drop height can be calculated from the ratio of the total kinetic energy and the total mass of the subsection (m_{sub}) multiplied by acceleration due to gravity (g):

$$H = KE_{test} / (m_{sub} \times g)$$

The tank fill should be of water at an equivalent volume to the fuel volume when the tank is fully loaded.

11 Do drop test onto an unyielding target (see Appendix 3)

The subsection will be dropped onto its side, so the side of the tank impacts the ground as it would in a rollover scenario.

G.6.5 *Appendix 5 – Option 3a/3b: example for a two-compartment subsection drop test*

⁴¹ The banded type 1 factor is from models and topple tests, banded type 2 values are from a topple test only

This appendix contains a worked example of the application of Option 3 based on a banded tanker type design. The example is presented in alignment with the five key stages of the test method process described in section 2.3.

Note that the second stage, ‘calculate the change in potential energy’, is based on the method presented in Appendix 4, and for the fifth stage, ‘drop test’, outline methods for carrying out the test are described in Appendix 3.

G.6.5.1 Appendix 5.1 Preparation – identify the two-compartment subsection to test.

A decision is needed at this stage on whether to carry out finite element modelling to find the two-compartment subsection with the highest deflection (and use the length-adjustment factor); or assume that it is the rear two-compartments and use the standard length-adjustment factor (see Appendix 4). If the tanker being assessed has a swept end or is of a stuffed design, then option 3a will be more suitable. At present, option 3b has not yet been verified for swept ends or stuffed tankers.

G.6.5.2 Appendix 5.2 Preparation – calculate the change in potential energy

In this example, the centre of gravity (CoG) of the liquid is assumed to be at the geometric centre of the compartments. The total mass of the liquid is still assumed to be the maximum allowable load (assumed to be a 95% fill in each compartment). As this leaves a small ullage, the CoG of the liquid will be slightly lower than assumed. However, assuming the CoG is slightly higher is a conservative assumption as the kinetic energy at impact will therefore be slightly higher.

In this example, the horizontal and vertical distances between the CoG for the tanker and liquid load, and the outer edge of the wheels are shown in Table A5.1. The outer edge of the wheels is the pivot line during the topple – see Figure A1.1.

Table A5.1: Distances between the centre of gravity (CoG) for tanker and load, and the pivot line (the tanker is horizontal and in the upright position)

	Horizontal distance (mm)	Vertical distance (mm)	Mass (kg)
Empty Tanker (including supports)	1275	1182	5 500
Liquid Load	1275	2100	31 400
Combined (empty tanker and liquid)	1275	1963	36 900

The height of the CoG when the tanker is at the tipping point (h_t in step 4 of Appendix 4) is obtained by carrying out a vector addition:

$$h_t = \sqrt{(1965 \text{ mm})^2 + (1275 \text{ mm})^2}$$

$$h_t = 2342 \text{ mm}$$

Potential Energy (PE) is calculated from the mass (m) x acceleration due to gravity (g) x height (h). The change in potential energy (ΔPE) is calculated from the change in the height of the CoG between the tipping point and the impact point (h_i in step 4 of Appendix 4):

$$\begin{aligned}\Delta PE &= mg(h_t - h_i) \\ \Delta PE &= 36900 \text{ kg} \times 9.81 \text{ m}\cdot\text{s}^{-2} \times (2.342 \text{ m} - 1.275 \text{ m}) \\ \Delta PE &= 386 \text{ kJ}\end{aligned}$$

h_i is half the width of the tanker (1.275 m in this case) - see Figure A4.1, step 4 of Appendix 4.

G.6.5.3 Appendix 5.3 Calculation of the impact energy per partition

It is assumed that the change in potential energy ΔPE , is the same as the kinetic energy at impact for the topple, KE_{Topple} .

$$KE_{\text{Topple}} = \Delta PE = 386 \text{ kJ} \quad (1)$$

The kinetic energy for the vertical drop case is adjusted to allow for differences between the topple motion and the vertical drop motion by applying a factor of 0.73 (explained in Appendix 4):

$$\begin{aligned}KE_{\text{Drop}} &= 0.73 \times KE_{\text{Topple}} \\ KE_{\text{Drop}} &= 0.73 \times 386 \text{ kJ} \\ KE_{\text{Drop}} &= 282 \text{ kJ}\end{aligned}$$

The impact energy per partition/surge plate is then calculated. In this example, an 8-banded tanker is assumed:

$$\begin{aligned}KE_{\text{part}} &= \frac{KE_{\text{Drop}}}{n_{\text{tot}}} \\ KE_{\text{part}} &= \frac{282 \text{ kJ}}{8} \\ KE_{\text{part}} &= 35.2 \text{ kJ}\end{aligned} \quad (2)$$

where n_{tot} is the total number of partitions/surge plates, including the ends in the complete tanker.

G.6.5.4 Appendix 5.4 Calculation of the length-adjustment factor

As the impact results in variable deformation along the length of a tanker, an additional factor is required to ensure that the impact in the drop test of a subsection represents the most heavily deformed compartments in a topple.

There are two options for making the adjustment:

Option 3a: Finite element analysis to calculate the 'length adjustment factor'

A finite element model of the complete tanker is run to represent the topple case. The rotational velocity at impact should be set to give the correct kinetic energy, KE_{Topple} , which is 386 kJ in this example (see **(1)** above).

The maximum deflections (reduction in width of the partition) for each partition/surge plate that were obtained from modelling work in this example are shown in Table A5.2, along with the average and maximum deflections for all partitions/surge plates.

Table A5.2: Deflections for each partition/surge plate in the topple model, in mm, from front to back

(front)	90	101	110	109	121	130	138	147(rear)
Average deflection:			118	mm				
Maximum deflection:			147	mm				

The length adjustment factor, f_{LA} , in this case is:

$$f_{LA} = \frac{\text{Maximum deflection}}{\text{Average deflection}}$$

$$f_{LA} = \frac{147 \text{ mm}}{118 \text{ mm}}$$

$$f_{LA} = 1.24$$

Option 3b: Use the 'standard-length adjustment factor'

As an alternative, a *standard-length-adjustment factor* of 1.25 can be used which has been obtained from the research work on this project (see Appendix 4). There is no requirement to perform finite element analysis with option 3b.

The calculations in this worked example will continue using the standard-length adjustment factor.

G.6.5.5 Appendix 5.5 Drop test: calculation of drop height and test details

In either case, the length adjustment factor is squared, as the relationship between deflection partitions and energy has been found to be to the power of 0.5 for all tanker types investigated to date.

Therefore, using the value of KE_{Part} from **(2)** above, the adjusted impact energy per partitions ($KE_{PartAdj}$) is:

$$KE_{PartAdj} = KE_{Part} \times 1.25^2$$

$$KE_{PartAdj} = 35.2 \text{ kJ} \times 1.56$$

$$KE_{PartAdj} = 55.0 \text{ kJ}$$

The total impact energy for the test is the adjusted impact energy per partition, multiplied by the number of partitions in the subsection being drop tested. As the subsection consists

of two compartments with one partition (no surge plates were present) the number of partitions, n_{sub} , (including both ends) is three. Therefore:

$$KE_{Total} = KE_{PartAdj} \times n_{sub}$$

$$KE_{Total} = 55.0 \text{ kJ} \times 3$$

$$KE_{Total} = 165 \text{ kJ}$$

This is then multiplied by the safety factor (S) of 1.1 (see step 9. Appendix 4) to give the test energy.

$$KE_{Test} = 165 \text{ kJ} \times 1.1 = 181.5 \text{ kJ}$$

In this example, the mass of the subsection test piece, m_{sub} , which consists of the rear two compartments filled with water to 95%, is 13 702 kg.

The drop height (H) can be found from

$$H = KE_{test} / (m_{sub} \times g)$$

$$H = 181.5 / (13702 \times 9.81 \text{ m/s}^2)$$

$$\text{the drop height } (H) = 1.35 \text{ m}$$

G.6.5.6 Appendix 5.6 Perform a two-compartment subsection drop test from the drop height calculated in Appendix 5.5

Ensure the filling ports are on the opposite side to the impact side of the subsection. The tanker should be filled with water to the level decided for the test (95% fill in this example).

When filling the two compartments, using a water meter on the discharge side of the pump is recommended in order to measure the water volume as the compartment fills. Leaks between the meter and the tank should be strictly controlled (leaks on the inlet side of the meter are not important). However, other methods to check fill level such as sounding rods may also be suitable.

Outline methods for carrying out the drop test are shown in Appendix 3.

Research on Performance Test Procedures for Petroleum Road Fuel Tankers



The aim of the research reported was 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 main conclusion of this research regarding the development of performance-based requirements for rollover safety is that the deflections and likelihood of major loss of containment experienced by road fuel tankers in real-world rollover scenarios can be replicated in a suitably specified, two-compartment subsection drop-test (or a full-scale physical topple test) supplemented by abrasion and penetration tests. The main outputs of this research were 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. A secondary output was the development of a better understanding of a frontal impact (tank rupture) scenario and the associated loading of the tank structure through the kingpin assembly / support structure.

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TRL

Crowthorne House, Nine Mile Ride,
Wokingham, Berkshire, RG40 3GA,
United Kingdom
T: +44 (0) 1344 773131
F: +44 (0) 1344 770356
E: enquiries@trl.co.uk
W: www.trl.co.uk

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